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THE

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B. B. BACHMAN, *President* COKER F. CLARKSON, *Secretary* C. B. WHITTELSEY, *Treasurer*

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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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The Summer Meeting

SELDOM does one find such a universal feeling of genuine satisfaction among the attendants at a national convention as that which predominated throughout the period of the Summer Meeting of the Society at White Sulphur Springs, W. Va., June 20 to 24. The attendance, though somewhat smaller than that of the past two or three years, reached 531 and all sections of the country and branches of the industry were represented. Contrary to the expectations of many, the weather conditions were ideal. The temperature never rose above 75 deg., the days were bright and clear and the evenings unusually cool. Better climatic conditions simply could not be desired. The cuisine, facilities and hospitality at the Greenbrier Hotel were as near perfection as any of our most famous hostelries can approach. The beautiful grounds with their background of picturesque mountains formed an appropriate setting for one of the most successful Summer Meetings the Society has ever held. An unusually large percentage of the members arrived early on Tuesday and remained through the entire meeting until Saturday afternoon. Every phase of the program was received with enthusiasm. The technical papers contributed much valuable engineering information and aroused a very active discussion. The sports and entertainment programs seemed to keep all busily engaged and dull moments were banished. One heard nothing but complimentary comment on the location and the facilities on every side.

Standards Committee Session

The regular session of the Standards Committee, which convened at 10:30 in the morning on Tuesday, June 20, was attended by 128 members and guests. After brief introductory remarks, and the declaration of a quorum by President B. B. Bachman, Standards Committee Chairman E. A. Johnston called for the reports of the Divisions as printed in the June issue of The Journal. There was also a report by the Lighting Division on the revision of the present S.A.E. Standard for Head-lamp Illumination, which will be printed in the August issue of The Journal.

The supplementary report on Motor-Truck Front-Axle Hubs was approved, with the supplementary table of ratings amended so that the former heading "Spindle load in lb. on solid-tire rating at ground" reads, "Assumptions on which calculations for spindle sizes were based."

The reports on Ignition-Distributor Mountings, Magneto Mountings and Starting-Motor Flange Mountings were approved as printed. The report on Breaker-Contacts was amended by omitting the reference to No. 8-40 thread size and approved.

The report on Tractor Drawbar Adjustments as presented was approved.

The report on Flywheel Housings was approved as printed. The report on Crankcase Drain-Plugs was discussed at considerable length. In view of the expressed opinion that this subject has to do with design rather than standardization for interchangeability, it was referred back to the Engine Division for further consideration toward its being discontinued. The report on Motorcycle Carbureter Flanges was withdrawn by Vice-Chairman R. J. Broege, due to criticisms that had been received. The Engine Division is to reconsider this subject.

The report on Leaf-Spring Steel was approved as printed. The report on Steel Spring Wire was amended by omitting the reference to "round, cold-drawn wire up to 3/16-in. diameter, except for some types of springs used in clutches, which are hot-rolled," and omitting the word "helical" from the caption to the table.

The reports on Head-Lamps, Electric Incandescent Lamps, Electric Incandescent Lamp Voltages and Motorboat Lighting Voltages were approved as printed. The supplementary report on Automobile Electric Head-Lamp Lighting Specifications was discussed at considerable length. A motion to endorse the complete report of the Illuminating Engineering Society on the Rules Governing the Approval for Headlighting Devices for Motor Vehicles, dated February 1922, with a reference added as to the acceptance by one State of tests approved in another State, and omitting the paragraph under "Approval," which refers to "Tilting Devices," was lost.

The principal reasons given against the recommendation of the Lighting Division were that portions of its report dealing with other than laboratory tests were non-technical, intended primarily for regulatory purposes, and were outside the function of the Society. The report of the Division, including Parts I and II, was finally

approved. As this report was not published in the June issue of THE JOURNAL, it will be given in full in the August issue.

The report on Aluminum Alloys was approved as presented. The report on Wrought Non-Ferrous Alloy Specifications Nos. 77, 78 and 82 were approved as printed, except for omitting from the captions mention of the purpose for which these specifications were formulated. It was felt that the latter information should be given in sub-captions or footnotes. Specification No. 83 was referred back to the Non-Ferrous Metals Division for joint consideration with the Electrical Equipment Division, in view of the work which is in progress in the latter on standard specifications for magnet wire. The report on White Bearing Metals was amended to include certain corrections in the percentages for Specifications Nos. 10, 10A, 11, 11A, 13 and 13A. The Specifications Nos. 13 and 13A as printed were the same as Specifications 14 and 14A through error.

The report on Flywheel Pulley Lugs, as submitted by the Stationary Engine Division, was approved.

The Parts and Fitting Division's reports on Passenger-Car Front Bumpers, Rod-Ends, Plain Steel Washers, Ball-Studs, Serrated-Shaft Fittings, Tank and Radiator Caps and Lock Washers were approved as printed. The discussion on Rod-Ends developed the suggestion that the Division consider the extension of the standard to include a series of even heavier rod-ends for truck application, it being stated that the present standard sizes provide rather small pin-bearing lengths and diameters. The Screw-Threads Division's report on Screw-Threads was approved as printed. The report on Gages and Gaging, which was proposed for general information only, was referred back to the Division for further consideration in view of the criticism that it did not deal adequately with gaging for errors in lead.

The report on Top-Irons was approved after being amended to specify a 5/8-in. length of thread, and a 1-in. length of stud.

The Springs Division reports on Spring-Eye Bushings and Frame Brackets for Springs were approved as printed. The report on Definitions was referred back to the Division because of criticism of the method of defining deflection, load height and free height.

The Lubricants Division's progress report that was presented only to secure suggestions and information for the Division's guidance followed a meeting held by the oil producing and consuming interests during the morning. A number of valuable suggestions were received and a revised tentative report will be prepared and circulated by the Division during the Summer.

A progress report was made by the Chairmen of the Passenger-Car and the Engine Divisions on their study of methods of numbering engines and frames for theft prevention, and to secure reduction of automobile theft-insurance premiums. The report was supplemented by the exhibition of a number of models and by lantern slides illustrating the application of many methods that have been considered. The progress reports on Metric Thrust Ball-Bearings, Brake-Lining, and Starting and Lighting Equipment were not given on account of the lack of time.

R. M. Hudson, of the Division of Simplified Practice of the Department of Commerce, presented a very interesting paper on the work of that Division. He explained the service that it is felt the Department of Commerce can render the automotive industry and the public through organized cooperation with the Society of Automotive Engineers and the National Automobile Chamber of Commerce. Mr. Hudson's paper will be printed in a later issue of THE JOURNAL.

The action taken by the Standards Committee on the reports submitted by the Divisions was reported to and approved by the Council and at the Business Session of the Society held Tuesday evening. The reports that were approved will be set forth in the August issue of THE JOURNAL, and submitted to a letter ballot of all the voting members of the Society.

NOMINATION OF 1923 OFFICERS

H. W. Alden was nominated to serve as President of the Society for the next calendar year by the Nominating Committee, which was completed and organized at the White Sulphur Springs Meeting. The committee reported the following other consenting nominees for the elective offices next falling vacant under the constitution, i.e., after the 1923 Annual Meeting of the Society:

First Vice-President—H. M. Crane
Second Vice-President, representing motor-car engineering—(Undecided)
Second Vice-President, representing tractor engineering—A. W. Scarratt
Second Vice-President, representing aeronautic engineering—E. P. Warner
Second Vice-President, representing marine engineering—E. J. Hall
Second Vice-President, representing stationary internal-combustion engineering—(Undecided)
Councilors (to serve during 1923 and 1924)—W. A. Chryst, F. W. Gurney and A. J. Scaife
Councilor (to serve during 1923)—H. M. Swetland
Treasurer—C. B. Whittelsey

The members of the 1922 Council who will hold over during 1923 are B. B. Bachman as past-president and Councilors C. F. Scott and L. R. Smith.

The Nominating Committee was constituted of Cornelius T. Myers (chairman), Metropolitan Section; V. G. Apple, Dayton Section; H. R. Corse, Buffalo Section; T. F. Cullen, Pennsylvania Section; L. A. Emerson, Minneapolis Section; W. S. James, Washington Section; T. J. Litle, Jr., Detroit Section; R. J. Nightingale, Cleveland Section; B. S. Pfeiffer (secretary), Mid-West Section; L. W. Rosenthal, New England Section; M. A. Smith, Indiana Section; and V. E. Clark, F. S. Duesenberg and G. E. Goddard, members-at-large. This was the annual Nominating Committee, provided for by the Society's Constitution, under which 20 or more members entitled to vote may constitute themselves a special Nominating Committee, with the same power as the annual Nominating Committee. The By-Laws of the Society provide that a special Nominating Committee, if organized, shall on or before Nov. 15 present to the Secretary of the Society the names of the candidates nominated by it for the elective offices next falling vacant, together with the written consent of each.

BUSINESS SESSION

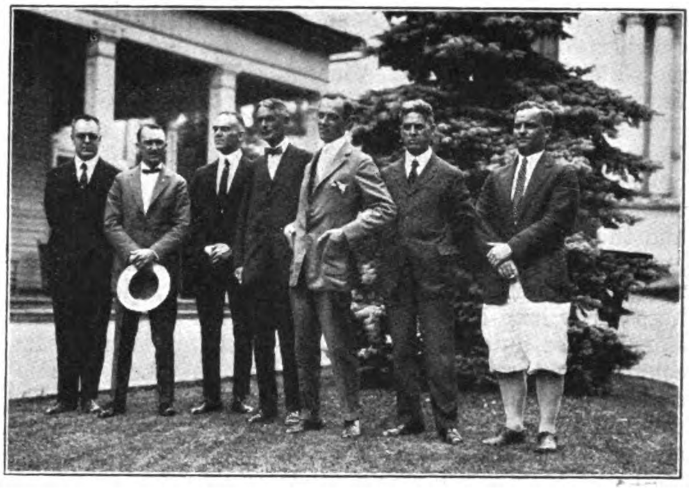
President Bachman's address, which is printed in full elsewhere in this issue of THE JOURNAL, was received very cordially at the Business Session held Tuesday evening.

It was reported that the Society's net loss for the first 8 months of the current fiscal year was $18,360.26. During the corresponding period of the last fiscal year the Society had an unexpended income of $14,504.88. The loss in income this year is due to a reduction in receipts of $27,439.30 as compared with last year. The amount of initiation fees from new members was $6,335.00 less than

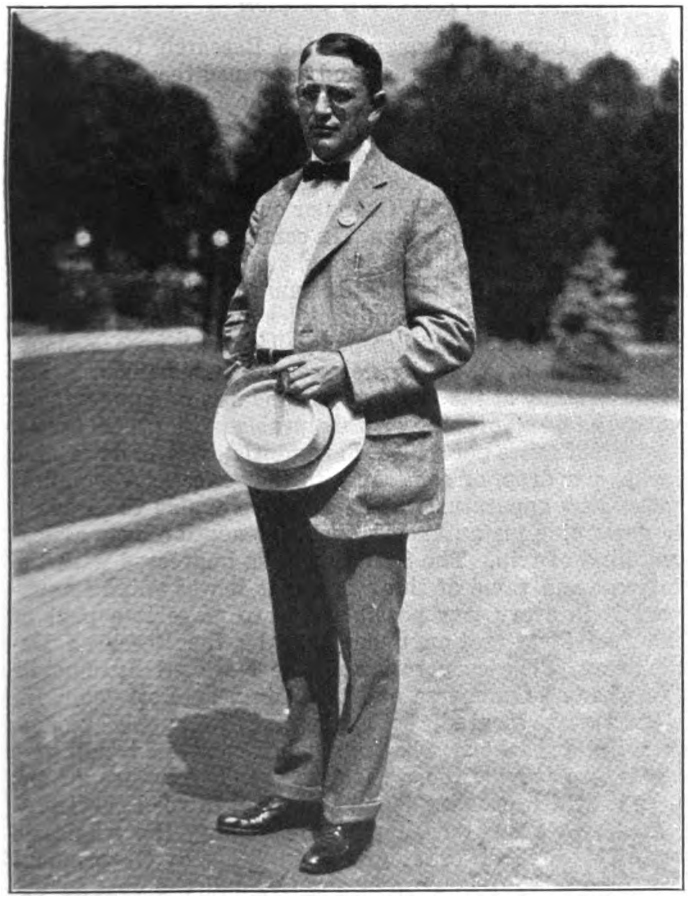
for the same 8 months of the last fiscal year. As a result of careful management the total operating expense for the period was increased only $5,425.84, notwithstanding added activities involving expenditure of approximately $10,000.

On April 30 the assets of the Society amounted to $180,127.63, these being offset by accounts payable of $9,159.29 and special reserves of $50,758.11; leaving net assets of $120,210.23, approximately $90,000 of this amount being in the form of United States Government and railroad securities.

The Meetings Committee announced that a two-day national Society meeting devoted to production engineering matters will be held in Detroit during October, this meeting including technical sessions and factory inspection trips. It was stated also that the annual Service and Tractor Meetings would be continued next year.



SOME OF THE COUNCIL



HERBERT W. ALDEN

The Membership Committee reported that the total enrollment of the Society on May 31 of this year was 132 more than on the corresponding date of last year, notwithstanding the fact that several hundred members were dropped for non-payment of dues or other causes.

The Sections Committee announced that all of the Sections are in a healthy financial condition and that many excellent papers had been presented and discussed at their meetings during last season. The committee advised strongly the practice of the Sections arranging their programs for the year in accordance with pre-determined plans. The Sections luncheon held on Thursday is reported at some length elsewhere in this issue of THE JOURNAL.

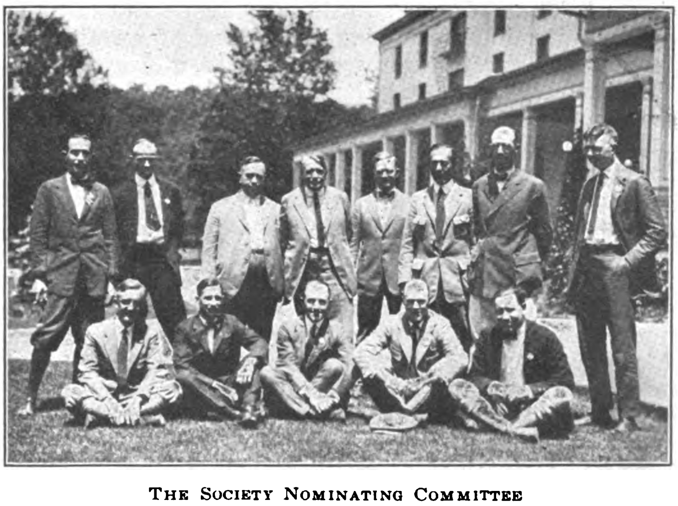
Under the item of new business, a lengthy discussion was had on the Society's affairs in general, including the matter of the grading of applicants for membership, Sections activities and the holding of local meetings in cities at which no Sections of the Society are located.

HIGHWAY MATTERS

Director W. K. Hatt, of the Advisory Board on Highway Research of the National Research Council, presented at the Friday morning technical session a valuable uptodate survey of studies on highway matters that are being made by different institutions throughout the Country. Professor Hatt later conferred with H. W. Alden, chairman, G. A. Green and Prof. W. E. Lay, of the Highways Committee of the Society, with a view to coordinating further the technical efforts of highway and automotive engineers. The Highways Committee of the Society, the Highways Committee of the National Automobile Chamber of Commerce and representatives of the Association of State Highway Officials are scheduled to hold joint sessions this month.

SIMPLIFICATION OF PRACTICE THROUGH DEPARTMENT OF COMMERCE

At the instance of the Division of Simplified Practice of the Department of Commerce, the National Automobile Chamber of Commerce and the Society of Automotive Engineers have appointed committees to make specific recommendations in furtherance of wider reduction to practice of automotive standards established and to be established. The National Automobile Chamber of Commerce Committee is constituted as follows: F. E. Mos-



THE SOCIETY NOMINATING COMMITTEE

kovics, chairman, D. C. Fenner, H. B. Harper, M. L. Pulcher and C. E. Salisbury. The personnel of the Society's committee is J. G. Vincent, chairman, C. F. Kettering, C. M. Manly, F. E. Moskovics and W. G. Wall.

These committees held sessions last month at New York City and at White Sulphur Springs during the time of the Summer Meeting of the Society. The purpose of the Department of Commerce was thoroughly endorsed and the economic value of the proposed procedure fully appreciated. It is believed that the joint action will result in wider use of standards and the reduction of unnecessarily long lists of various sizes and sets of dimensions in the automotive field. It is the opinion of the committees that the first subjects to be taken up and studied are ball and roller bearings, solid and pneumatic tires, starting and lighting batteries and their carriers, and certain spark-plug dimensions that are essential for most satisfactory vehicle maintenance by users.

The facilities of the Society will be utilized in collecting data and making studies in connection with the work of the two committees and the cooperation of all allied associations concerned in the industry will be sought.

The Aeronautic Session

Unfortunately the attendance of engineers interested in aeronautics was very small at the White Sulphur Springs meeting. Those who did attend showed great interest in the papers of Prof. E. P. Warner and Capt. G. E. A. Hallett, both of which were printed in the June JOURNAL. Capt. Hallett's paper was abstracted by E. T. Jones, the author being unable to attend the meeting due to illness. Mr. Jones supplemented the paper with entertaining motion-pictures depicting some of the engineering and flying activities at McCook Field. These were followed by authentic aviation and front-line action pictures of the recent war with Germany, which were loaned through the courtesy of the photographic section of the Signal Corps. The paper by C. L. Egtvedt was read by title only in the absence of the author.

V. E. Clark, chairman of the session, asked Professor Warner whether he had considered including some measure of the wing-curve characteristic such as $K_{y\ max.}$ in his performance formula and if this would not assure greater accuracy in the results obtained. Professor Warner stated that this addition would tend to destroy the simplicity of the suggested formula without increasing its accuracy appreciably. H. M. Crane congratulated Professor Warner on the derivation of a formula, devoid of unwieldy mathematics, that gave results sufficiently accurate for design approximations. In answer to a question, Professor Warner stated that parasite resistance is not a factor in determining the ceiling of an airplane because the greatly increased angle of attack at high altitudes results in a very large wing-drag which dwarfs the parasite resistance. However, at high speeds and a small angle of incidence, parasite resistance is a limiting factor since the drag is then relatively small. After the presentation of Captain Hallett's paper there was a general discussion commending the work of the Air Service Engineering Division at McCook Field. Professor Warner considered it inadvisable to divert all government funds to industrial plants for development work, although efforts were being made to bring about this condition. He favored leaving the work of a research character in governmental laboratories since the results would be circulated generally for the benefit of all in the art. H. M. Crane considered that Major T. H. Bane had maintained an excellent balance in expending federal funds so that his own engineers and those in the industry could be employed to the best advantage. He favored strongly the continuous testing and development of engines and planes of accepted design, believing that our most successful aircraft are the result of slow and rigorous development work. Elmer A. Sperry believed that McCook Field was of great assistance to the industrial designer because of the service it rendered as a source of research information. He strongly recommended a continuance of ample appropriations for the Field's use, feeling that the money could not be expended more efficiently. Professor Warner and Mr. Crane both argued for a closer contact between McCook Field and the airplane designer after the experimental models had been accepted and started through their flight and performance tests, this being the period when they were undergoing alterations to attain a final state of development.

Research Session

H. M. Crane, chairman of the Research Committee, called the meeting to order and explained the change that had been made in the program. The first paper on the program, that by W. S. James on Fuel Research at the Bureau of Standards, was deferred until Saturday morning, and F. C. Mock read the paper prepared by him in collaboration with M. E. Chandler on the Hot-Spot Method of Heavy-Fuel Preparation. In connection with the paper by Mr. James, Mr. Crane brought out the importance of research work on the fuel-consumption in automobiles. The question of fuel in automobiles on the road has been a matter of so many miles per gallon, with an occasional explanation of the kind of service undertaken. Mr. James's study at the Bureau of Standards was undertaken with the idea of stating this problem in other terms. Instead of miles per gallon, his ambition was to determine inches per drop, or something to that effect. A complete set of new apparatus was designed to enable the Bureau to determine instantaneous fuel-consumption over almost any kind of trip. The Society was so impressed with the interest and value of this work that it was voted to support it in every way possible, and to make it a part of the fuel investigation undertaken by the Research Department. The cooperation of the National Automobile Chamber of Commerce and the American Petroleum Institute was secured, both agreeing to supply money, material and labor. The tests, when completed, will give us a more complete idea than we ever had before, of the way in which gasoline is actually used in road service.

C. T. Coleman read a paper describing the wholesale use of fuel of different grades in fleets of Government trucks, operated in normal service, and the results obtained from the observation of mileage and other characteristics. Post-Office trucks were used, operated by an exceptionally high-class personnel, under the best economic conditions. Four fuels were used, lettered *A*, *B*, *C* and *D*. The second, which was the key fuel, represented the average standard commercial gasoline that we are buying today at the service-stations. Tests were run in Philadelphia and Pittsburgh, the first city representing smooth going with few hills and grades, and the second hilly country and cobblestone streets with considerable grades. In Philadelphia, 37 3/8-ton Fords, 83 General Motors 3/4-ton trucks, 31 General Motors 1 1/2-ton trucks and 12 3-ton Packards were used. In Pittsburgh there were used 42 White 3/4-ton trucks, 16 2-ton Whites, and 16 3-ton Rikers. Every type of truck represented a different carbureter and a different kind of induction system.

The different groups were divided equally in respect to

kind of trucks, tonnage, and service performed. One section was run on the key fuel *B* at all times, and the other section on *A*, *C* and *D* fuels, thus enabling the testing engineers to compare the two units of trucks, or the one fuel with the other. Fuels were changed every week. The mileage was noted when the gasoline was put in, and every time the fuel was changed the residue of fuel was drained out and weighed. At the end of each week or of each run, a quart sample of the crankcase oil was taken and sent to the Bureau of Standards for testing.

Taking the distillation curves of the fuels used, *A*, *B*, *C* and *D*, we might say, taking *B* as the key fuel, to represent 100-per cent production per barrel of crude oil, *A* represents 80.9-per cent production per barrel; *C* 114.4 per cent; and *D* 127.8 per cent.

Slides were shown, giving average results for different trucks run in Philadelphia and Pittsburgh, resolved into percentages of miles per gallon. *The outcome of the investigation was, there was very little difference in the percentage of miles per gallon of the different fuels, but there was a considerable difference in crankcase-oil dilution on these fuels.* In all the trucks there was a slight advantage in favor of the *A* fuel, and it was noticed that with a fuel of heavier quality or of higher end-point, the crankcase dilution tended to increase. There was more crankcase dilution on *C*, and proportionately still more on *D*. Slides were thrown on the screen giving the curve of the percentage of dilution against volatility.

Summing up, Mr. Coleman said, taking the percentage of miles run on the *B* fuel per barrel of crude as 100, we find that fuel *A* gives 86-per cent miles per barrel, and 3.4-per cent actual crankcase-oil dilution. With *B* fuel the crankcase dilution is 4.7 per cent. With *C* fuel, representing 123-per cent miles per barrel of crude, the figure for crankcase dilution is 6.8, and with the *D* fuel, averaging 136-per cent miles per barrel of crude, the crankcase dilution is as high as 9.1. It appears, therefore, that more miles are obtainable per barrel of crude on the heavier fuel, but here the problem of crankcase-oil dilution becomes more acute.

In opening the discussion, Mr. Crane urged the necessity of maintaining a reasonable degree of uniformity in the quality of the fuel, and advocated that the quality of the fuel used be adapted to the atmospheric conditions prevailing in different parts of the Country and at different seasons of the year. He commented on the fact that in Philadelphia, where the trucks were operating at part throttle, and with a fair degree of heat, the heavier fuels did not affect the mileage operation to a great extent. In Pittsburgh, however, where there was some real full-load operation, the effect of the increased end-point was much more noticeable.

P. S. Tice asked what apparatus was used to determine crankcase-oil dilution. Mr. Coleman replied that the values given were obtained in a 100-cc. Engler-flask, the apparatus used being a conventional distillation outfit with an ice bath and an ordinary Engler-flask. Mr. Tice considered that the "ordinary Engler-flask" was the weak point. In response to a query from F. C. Mock, Mr. Coleman explained that all carbureter adjustments were left to the mechanical force, since the test was carried out on an ordinary service basis. J. H. Hunt commented on the influence of cold starts and stops on crankcase dilution, and asked if any steps had been taken to correct for this. Mr. Coleman said that starts and stops were not taken into consideration because the trucks do the same thing every day, and it was simply a standard test, run under normal service conditions. Weather conditions were eliminated by the expedient adopted of running half the trucks on *B* fuel and the other half on the other grades of fuel.

Mr. Crane made some interesting observations on the importance of the human equation in tests of this kind, and added that the chief defect in laboratory and factory tests is that the mechanics are more skilled than the average. For this reason he contended that a test of this kind could afford no criterion unless it were extended over a considerable period of time. O. C. Berry argued that the use of the same fuels in the hands of the general public would not result in anywhere near the same mileage per gallon, and expressed the opinion that any test reporting miles per gallon should be preceded by a very careful series of performance tests to determine the correct carbureter setting. In thanking Professor Berry for his constructive criticisms, Mr. Crane urged that any other suggestions as to forms of test that might be considered helpful or desirable be sent to Dr. H. C. Dickinson, the research manager of the Society. Some interesting points were raised by P. J. Dasey who held that performance is the factor of chief importance in any test, while economy is secondary. He said that in his own laboratory he makes comparative tests for crankcase dilution using city illuminating gas instead of gasoline, and yet light ends may be found in the lubricating oil in the crankcase, which are not due to dilution, but to the cracking process that is constantly carried on in the cylinder. The dilution with heavy ends that is found in the crankcase are not the fault of the refiners so much as of the men who design the engines to handle the fuel. In reply to Mr. Dasey's suggestion that only one brand of oil be used in all such tests, Mr. Coleman stated that owing to the dissimilar oiling systems of the Ford and the other light ¾-ton truck, the same oil could not be used in both.

R. E. Carlson, of the Bureau of Standards, followed with a paper in which he gave a brief outline of the status of the cooperative fuel-research program that the Bureau of Standards has undertaken in cooperation with the National Automobile Chamber of Commerce and the American Petroleum Institute. He began by saying that the work may be called a continuation of that which Mr. Coleman has been doing on a somewhat larger scale, except that it is being conducted along more nearly laboratory lines. Four grades of fuel are to be used. The object of the tests is to determine the fuel-consumption under average road conditions with average cars in the hands of average drivers. These averages are rather difficult to determine but a program has been worked out, and the work is to be begun early this month. Mr. Carlson invited the cooperation of the members to determine what the program should be. The fuel is to be the same as that used by Mr. Coleman in his tests, but it is hoped that more accurate results will be obtained through the control of the various factors.

Mr. Mock presented a summary of the paper on The Hot-Spot Method of Heavy-Fuel Preparation. The authors investigated the various methods of heat application in an endeavor to produce the minimum temperature necessary for a dry mixture. It is their conviction that the sole requirement of satisfactory operation with kerosene and mixtures of the heavier oils with alcohol and benzol is the proper preparation of the fuel in the manifold.

They found that the minimum temperature varied with the method of application of the heat, and proceeded to make an analysis of the available methods on a functional rather than a structural basis. Three of these methods are discussed

(1) When the heat from the walls of the manifold is applied through the medium of the air
(2) When it is applied to the fuel alone, or partly to the fuel and partly to the air
(3) When a spray of atomized fuel and air is directed against a heated surface

A device was constructed by which the three main variables, the exhaust temperature, the exhaust flow and the area of the heating surface, could be regulated and the three remaining variables, the quantity of air, the quantity of fuel supplied and the quantity of fuel vaporized, might be controlled.

Taking into account the wide range of temperatures that the air charge and fuel supply undergo before entering the intake-manifold system, a quantitative computation of heat transfer was made, and the conclusions were drawn that only by a combination of centrifugal force, surface tension and the force of gravity could the unvaporized drops be separated from the fuel charge, and that the conditions of combustion are governed by the rate of fuel feed from the manifold to the cylinder, and not from the carbureter to the manifold. The paper was illustrated with drawings.

H. W. Alden opened the discussion by commenting on the mystery that has always surrounded the hot-spot. He called on Thomas Midgley, Jr., who described a new preparation brought out by the General Motors Research Laboratory to prevent fuel knock. He described it as tetra-ethyl lead, and offered to give a sample of it to anyone who cared to investigate its properties.

Dr. H. C. Dickinson then presented his paper, Progress of the Research Department. In it he emphasized the necessity for research, touched on the importance of the universities as bases of operation for pure research, described the resources of the Department and the special facilities that it has to offer and gave a short description of the work done in connection with the information service. He laid particular stress on the fact that the Research Department is equipped to give assistance to engineers and others in the industry and that all its resources are at their command. He concluded by describing the three major research enterprises in which the Society is engaged at the present time, namely the cooperative fuel research that is being carried on under the direction of the Bureau of Standards, the Society's fuel-research program, and the cooperative Highway Research program under the auspices of the National Research Council.

In discussing Dr. Dickinson's statement of the work done by the Research Department along fuel lines, P. S. Tice brought out the point that we have not yet arrived at a method of carburetion or handling of the fuel in the intake, that will give us the maximum fuel-economy, and suggested that we are wasting time when we try using several different fuels of known volatility in several different devices. T. J. Lítle, Jr., expressed the opinion that the most important task is to prepare for the heavier fuels that are bound to come. Chairman Crane pointed out that this work is to be accomplished by the present investigation, which will give us a curve that will indicate the utilization value of petroleum distillates of different volatility in engines of the present general type. Either the fuel must be modified to suit the needs of the present type of engine, or the present type of engine must be completely changed. He placed his confidence in the possibility of modifying the fuel to suit the engine that has been developed because of its simplicity and service, rather than in adapting the engine and complicating it to use it with some arbitrary form of fuel. There are millions of cars in service today which cannot be altered materially.

Dr. Dickinson explained that the tests, so far, had shown that the cars and trucks had actually utilized the heavy 500-deg. end-point fuel without any very marked decrease in economy. This does not take into account the question of crankcase-oil dilution. W. S. James alluded to the question now before the Federal Specifications Board as to the end-point of the most suitable gasoline. The refiners maintain that the present type of commercial gasoline is satisfactory and that the end-point or 95-per cent point can be raised without detriment. This matter is to be taken up by the refiners at a meeting to be held early this month. At the present time there are practically no data on the advantages or disadvantages in actual service of fuels with varying end-points. The few tests reported furnish at least some indication of whether the refiners' demands should be granted. He suggested that possibly the greatest gains could be expected from carbureter and car adjustments, rather than from a change in the fuel. Mr. Lítle said he believed that he expressed the view of the automotive industry when he stated that a better gasoline costing a few cents more would be eagerly welcomed. Chairman Crane endorsed this statement and added that the owner of the car is as much interested in the mileage he gets per gallon of fuel, as in the cost of the fuel per gallon. T. A. Peck stated that the petroleum industry was ready to go more than half way to meet the automotive industry, but that careful conservation of the existing supply of raw material must be the primary consideration.

Fuel and Engine Session

Chairman O. C. Berry opened the session by calling on W. S. James, of the Bureau of Standards, to present his paper on Fuel-Volatility Research at the Bureau of Standards, with demonstration of test-car equipment. Mr. James's talk was profusely illustrated with lantern slides which showed the various devices that have been adopted by the Bureau in connection with its fuel-research program. Among the devices illustrated and described briefly by Mr. James were the accelerometer, the velocity-recording instrument, the oil and carbureter tester, draw-bar dynamometer, and a multiple recorder.

M. C. Horine inquired what data Mr. James had on the effect of friction of the springs. Mr. James replied that in driving at a uniform speed along level stretches of road it is possible to get a zero independent of spring action. Chairman Berry discussed calibration on level roads, and noted that inertia effect is eliminated largely because there is very little movement of the mercury itself in the accelerometer. The effect of temperature on the accelerometer can be practically eliminated by making the free ends wide.

Mr. Chase was inclined to think that inaccuracies would creep in owing to the instability of the pens due to irregularities in the road. Mr. James replied that the friction of the pens made very little difference owing to the motion of the paper and the vibration of the car. The pens do not stick, thus avoiding the resulting reduction of sensitivity.

Thomas Midgley, Jr., presented the paper, prepared in collaboration with T. A. Boyd, on Detonation Characteristics of Some Blended Motor-Fuels. The authors have measured the effects of admixtures of various percentages of alcohol and alcohol-benzene mixtures for reducing the detonating tendency of paraffin hydrocarbons. These results represent an extension of previous work in which similar determinations were made for benzene

and other aromatic hydrocarbons. The bouncing-pin apparatus was used for making the determinations. The data obtained by its use are considered to be remarkably accurate.

In order that the effects of the blending materials might be measured through as wide a range as practicable, they were blended with kerosene for making the majority of the determinations. This made it possible to ascertain the characteristics of the materials up to a concentration of 80 per cent of benzene or 50 per cent of alcohol without introducing the difficulties due to excessively high engine-compression. Because xylidine has the property of exerting a powerful suppressing action on detonation when present in a fuel in percentages that are relatively very small, the standard used as a basis of comparison in the tests was composed of small percentages of xylidine in the paraffin fuel. Tables and curves are appended that show the results of the tests in detail.

Mr. Bachman, in discussing the paper, asked Mr. Midgely if the kerosene dilution with benzol had proved more effective than high-test gasoline dilution with benzol. Mr. Midgely replied that his observations had borne out this surmise. Mr. Chase asked what procedure had been followed in setting the spark for measuring detonation. Mr. Midgely said that the spark had been set for the maximum power.

G. A. Round read an abstract of his paper on Oil-Pumping. He defined oil-pumping and mentioned its results. The influence of various operating conditions was brought out, particular reference being made to passenger-car service. The factors that control the rate of oil consumption were brought out, and some unusual conditions reported. Various features of piston grooving and piston-ring design were mentioned and the effect of changes illustrated. The relative advantages of the splash and force-feed systems as affecting the development of oil-pumping troubles were set forth and improvements suggested. A new device for reducing oil-pumping and dilution trouble was described and illustrated.

F. F. Kishline, discussing Mr. Round's paper, asked if the tests referred to were made on a dynamometer or on the road. Mr. Round stated that the tests were made on the road. In discussing methods adopted for dealing with an excessive consumption of oil, Mr. Round stated that oil has to be controlled at the source and cannot be regulated by a piston.

A. A. Bull gave his paper on Oil Consumption, in which he considered fundamental factors. No attempt was made to determine the differences between lubricating systems. Beyond the fact that different oils apparently affect the oil consumption and that there is a definite relation between the viscosity and the oil consumption,

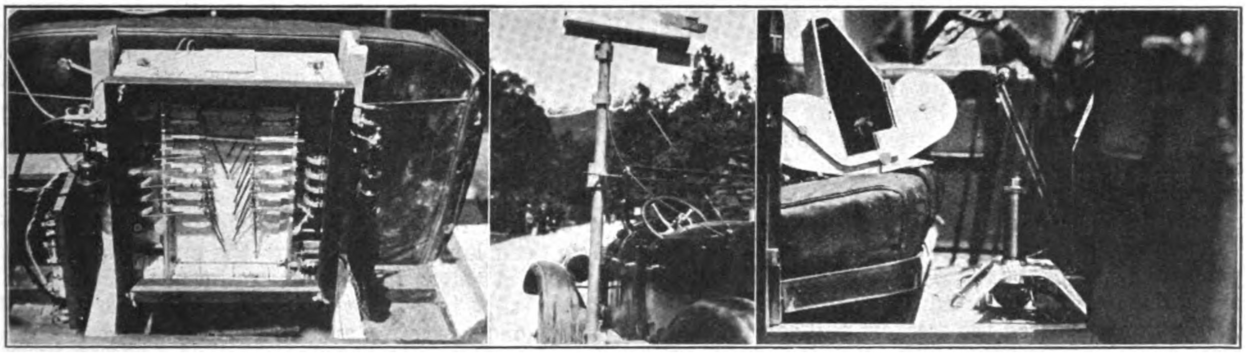
A Portion of the Golf Course

the effect of the physical characteristics, or the quality of the oil, did not receive particular attention.

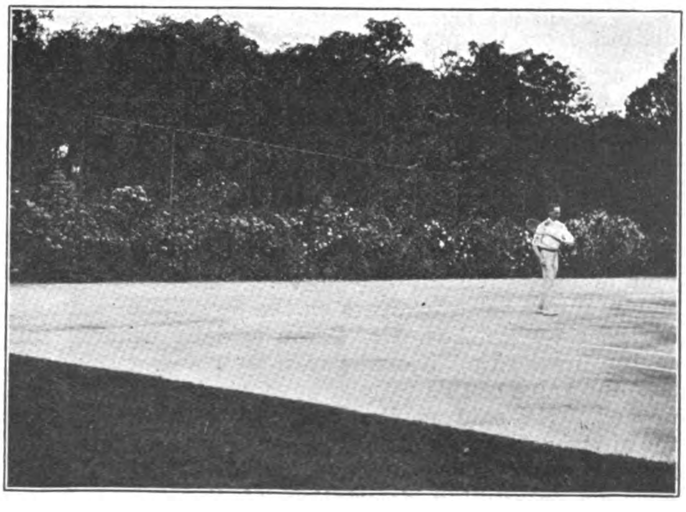
The paper describes the methods of testing and the subject is divided into

(1) The controlling influence of the pistons, rings and cylinders
(2) The controlling influence of the source from which the oil is delivered to the cylinder wall

The subject is treated under headings that include the piston-ring; the effects of oil-return holes, side-clearance and ring-motion; thin rings; influence of piston-fit; efficiency of the scraper-ring; ring and cylinder contact; carbonization and spark-plug fouling; oil-supply control;



Some of the Apparatus Developed by the Bureau of Standards for Testing the Performance of Engines on the Road



President Bachman in Action on the Tennis Courts

influence of oil viscosity; effects of dilution; external oil-leaks and breather discharge; and influence of controlling lubrication in proportion to throttle opening.

Motorbus Session

President Bachman opened the Motorbus Session by introducing G. A. Green, vice-president and general manager of the Fifth Avenue Coach Co., who then read his paper entitled, Principles of Motorbus Design and Operation. The paper dealt with the fundamental characteristics of buses, the principles on which their design and operation should be based, fares, service requiremens and the unwisdom of overloading. The subject was treated impersonally, except for specific references that were made from time to time to the practice followed by the Fifth Avenue Coach Co. in its equipment.

The factors controlling bus design are said to be

(1) Safety
(2) Comfort and convenience of the public
(3) Minimum operating cost

The various subdivisions of each were commented on in some detail and numerous illustrations and tabular data supplemented the text. The conclusions reached are that trucks or automobiles, either modified or unmodified, are incapable of rendering satisfactory service as buses, and that if the Society would concentrate its standardization work on the motorbus, much good could be accomplished.

In opening the discussion, President Bachman emphasized three points as being of particular interest, namely, powerplant construction, control and stability.

R. E. Plimpton then read his paper on Some Fundamental Characteristics of Present-Day Buses. He enumerated the distinctive features of buses designed for city, inter-city and rural services, illustrating his remarks by practical examples. Steam and electric motive-power were discussed and considerable attention was given to chassis components requisite for bus service. The general types of bus body were treated, together with the influence of climate and local preferences.

Mr. Plimpton discussed the comfort and convenience of passengers and problems of heating, lighting and ventilation and methods of fare collection. State and local regulations were referred to in connection with their effect upon bus operation. The paper was accompanied by illustrations and a table showing condensed specifications.

President Bachman opened the discussion by remarking that Mr. Plimpton's paper represented the technical newspaperman's point of view on the bus problem, as distinct from the preceding paper that treated the subject from the engineer's standpoint. Mr. Green agreed with Mr. Plimpton that one of the greatest problems from the viewpoint of both designers and operators of motor buses is the elimination of all unnecessary parts. Aside from the matter of weight and first cost, there is the question of repairs and renewals, also possible failures and consequent delays and service interferences. For normal bus service he advocated a four-cylinder engine of approximately 300-cu. in. capacity. Referring to Mr. Plimpton's comments on the possible advantages of the six-wheel bus, Mr. Green said he believed this design involves additional parts, weight and cost, and that the value of this form of construction had not been proved. He said emphatically that in his opinion the six-wheel vehicle should be considered as being still in the early experimental stages. H. L. Howell gave an interesting description of the types of bus operated by the London General Omnibus Co. Before the war the company used the B-type bus, with a 34-passenger seating-capacity. This was succeeded during the war, when the buses were used for transport service on the Western Front, by the K-type, 46-passenger bus, which in its turn has been succeeded by the S-type 54-passenger bus. He added that the cost of operating the S-type 54-passenger bus is not much greater than that of the B-type 34-passenger bus.

Mr. Green brought out an interesting point in discussing the types of equipment necessary for bus service. He stated he believed that in the majority of instances, initial operation ought to be commenced with single-deck vehicles; then after the service is built up, double-deck buses should be added; and that practically all operations require the two distinctive types. After the service has reached a point where double-deck vehicles are required, the single-deck type will be found extremely valuable to aid in the natural process of development for operation during cold and wet weather and for all kinds of special service.

Several members inquired as to what extent automobile companies are specializing on the production of motorbuses, since it is evident that the motorbus requires special parts and specifications. Mr. Plimpton said that several companies are now specializing on such construction. It is hoped that in time the building of motorbuses will be so simplified and standardized that it will be possible for any operator to order a fleet of buses that will be standard for the type of service he has in mind. Here again, the need for standardization work by the Society was strongly emphasized.

Passenger-Car Session

J. V. Whitbeck, chairman, introduced P. M. Heldt, who read his paper on Overhead-Camshaft Passenger-Car Engines. The paper illustrated the steady increase in the use of overhead valves in passenger-car engines from 6 per cent in 1914 to 31 per cent in 1922. This increase has been largely effected by the successful operation of overhead valves in aircraft engines and by the publicity given this type of valve through its almost universal adoption on racing machines. Mr. Heldt described the methods of operating valves in the cylinder-head; the advantages of the valve-in-head construction as regards the form of combustion space, engine cooling and high-speed operation; the reason for using an overhead camshaft to operate the valves on racing engines, the question of noisy operation and the possibility of having an overhead-camshaft engine operate as quietly as one in which the camshaft is enclosed in the crankcase; the feasibility of silent operation with a rear drive; the use of the various types of gear for the camshaft drive, with an estimate of the advantages and disadvantages of each, illustrated by practical examples; and some radical designs of overhead-camshaft drive and valve-actuating mechanism developed abroad.

Mr. Crane opened the discussion by commenting on the causes of noise, which he attributed to the seating of the valve. He stated that the valve opening is almost noiseless. In comparing the relative merits of the different types of engine, he stated that the overhead-camshaft engine lacks accessibility and the drive tends to be noisy. For a quiet high-speed engine he considered the L-head most suitable. For a moderate-speed engine he preferred the overhead-valve construction, which can be made almost noiseless.

(*Concluded on page* 119)

Presidential Address of B. B. Bachman

THE activities of the Society during the period since we met in New York City have been numerous. I would like to lay a few of them before you in the way of a report, covering in a general way the problems that have come before the Council. This can be accomplished best by following the work as divided among the administrative committees.

THE MEETINGS COMMITTEE

The results of the activities of the Meetings Committee during this period need little comment as the material evidences are before you, and will unfold during the session in which we are now participating. On the basis of the work that I know they have each done individually, I believe that I can anticipate your unanimous approval, and extend to the Committee your appreciation as well as that of the Council for the sacrifice of time and effort that they have made.

For the future, I know the Committee is interested in the activities of the Sections, of which I will speak more in detail later, and will welcome the opportunity for more active cooperation with the Sections in establishing a rounded-out and harmonious program of meetings for the year. As a large part of the value of the Society to its members lies in the meetings that are held, there has been some discussion as to whether it would not be wise to hold annually more than the two meetings for which the Constitution provides, and very definite thought is being given to this subject.

SOCIETY MEMBERSHIP

The Membership Committee has been dealing with a very serious problem in the affairs of the Society. The growing activities and the new fields of work that are opening around us on every side require the addition to our ranks of all who are equipped to assist us in our work or who can be benefited by it. While the scope of our organization is large, it is in a certain sense, limited. We must keep in mind that restricting our membership to the field of active designers of automotive vehicles and their more important component parts would be a very narrow policy. On the other hand, we must recognize that opening our doors for the general admission of all without regard to the service we can render them or they can render us would be unwise. Between these two extremes, the Committee and your Council believe, there is a sufficiently large field for the Society to draw from in permitting a rational and satisfactory growth.

This field, it seems to me, will cover, first, the designer of automotive apparatus and their important component parts; the instructor in arts and sciences relating thereto; the specialist, or the man versed in the design, construction and efficient operation of the production agencies for materials and completed structures; and, finally, but by no means least, the man who is skilled and is competent by training and experience in the maintenance and operation of the apparatus that the industry manufactures. This view may be criticized as being too broad, but I am firmly convinced that unless we recognize the valuable assistance that these other classes can render in developing the field of internal-combustion power application, and are willing to receive them on the basis of equality to which their ability and dignity entitles them, we will be retarding our attainment to that position of consideration to which we are entitled.

In the class of those who, while not engineers in the broad sense outlined above, are nevertheless interested in the work that we are doing, and directly or indirectly are benfited by it, there should be included with the man engaged in marketing our products, the man who has the responsibility of purchasing materials that we use in our processes of manufacture. Both these classes can in many instances give a wider perspective to our vision and can be materially benefited by our activities.

While the continuing growth and activity of the Society require careful consideration of our financial resources, and while the fees that are received from our members are a very important source of revenue, we must guard against any plan of membership increase that has only revenue in mind. The work that the Society is doing and the service that we can render to each other as members cannot be measured by the cost, nor should we knowingly solicit membership for mere financial support from anyone.

Another problem that the Membership Committee has actively in mind is the question of retaining the active interest of those who are already members. The recent industrial conditions have brought about the reduction of organizations which has resulted in personal difficulty for some of our members. There are others who possibly have been persuaded to join on the wrong basis, and have therefore never really been members.

In any event, whatever the reason may be, it is probably only natural that there should be a percentage who are delinquent in their financial obligations and thereby, as a result of our constitutional provisions, deprived of some of the benefits of membership. Extraordinary efforts have been made during the past months to increase the value and effectiveness of our employment service for the assistance and benefit of those who are numbered among those who have suffered personal loss through the recent unsettled conditions. As for the others, it is difficult to say just what steps should be taken to stimulate their interest and to retain their active association with us, but several methods are being actively canvassed toward this end by various committees and the Council.

THE SECTIONS

The work of the Sections Committee in the phase of the Society's activities that it represents is undoubtedly of prime importance. As was to be expected, the proper organization of these activities has presented for a number of years some very intricate problems which are far from being settled, even today. Fundamentally, an organization of the character of ours is dependent in a large measure upon two things for its success; first, the character and value of its meetings; and second, the character and value of its publications.

It is manifestly impossible with a widespread membership that more than a relatively small percentage can attend meetings such as this and at the same time the multiplication of meetings on a national scale at more frequent intervals, while desirable in some ways, will not in itself fulfill all the possible functions of an active local Section. The theory upon which we have conducted the affairs of the Sections to date has been to give them a very large measure of indpendence, allowing them to direct and regulate their affairs in accordance with their local needs and under officers of their own selection.

They have been financed in large part by the contribu-

tions of the members of the Society who have joined these Sections, the Society contributing to the support of the Sections in addition. The reason that the present system has been adopted is that it has been felt that only a percentage of the membership of the Society would be so located geographically as to benefit by participating in Section activities, except to the extent that the multiplication of Section meetings with the presentation of valuable papers would furnish subject matter for THE JOURNAL. In view of this, it was felt that, even if it were possible, it would be wrong to appropriate from the funds of the Society the necessary amount to finance completely the Sections whose activities would be of direct benefit to only a portion of the membership. At the same time, on the basis of the increased contributions to THE JOURNAL, it was felt that some appropriation was justifiable.

Against this theory of the present method, we have the view that those members who wish to participate in Section activities should not have to pay further dues than those to which they obligate themselves in joining the Society; and it is urged that this additional taxation is in a large measure responsible for difficulty in bringing up the membership of the Sections to what it should be. There is much to be said for both of these viewpoints, but fundamentally I do not believe that in either one of them lies the secret of success or the reason for failure in the Section activities.

It has been suggested that a very large percentage of our members do not care to attend Section meetings, and unfortunately this is probably only too true; but I believe that one of the reasons for this lack of willingness to attend is that a proper survey of the need of the members has not been made and the programs that have been put forward have not been consistently of the caliber or kind to attract a consistent attendance. This statement is not made with any intention of criticizing any of the past or present Section administrations. It is merely a fact that I believe we must face thoroughly, and a problem that we must solve before we can put the Section work on the high plane where it should be.

In a large degree the Section problem is similar to the Society problem. While we recognize the sacrifice of time and the effort and thought contributed by committee members, we surely recognize that if it were not for our headquarters organization we would be lost. I know of no individual who has the ability for organization and the time to devote to the detail work of properly conducting a Section. The result is lack of continuity of effort and policy.

SOCIETY FINANCES

During the last year, the question of finance has caused your officers considerable thought and anxiety. For the first time in many years, our current revenues have been insufficient to meet our expenses and afford a margin to be transferred to surplus. This is due to several things: first, to a loss in income from our advertising; and, second, to a reduction in the number of new members. Both of these conditions, it is believed, are temporary and will show improvement with a recovery of normal business conditions. We should nevertheless recognize that growth in usefulness and numbers will require not only continuation but expansion in our services, which will require thought and careful planning to balance the budget. It would have been possible this year to meet this emergency by a reduction in our activities, and it would be easy to recommend raising dues to prepare for the future. It is, however, the feeling of your officers that it is in times of commercial difficulty that organizations of the character of ours should increase rather than decrease their activities, for the reason that it is during times of this kind that the members individually and the industry as a whole need the greatest stimulus.

Regarding the raising of membership dues, there are of course many who could meet an increase with little difficulty, but on the other hand there are a number who, while they are vitally interested in the work of the Society, have found it impossible to meet the current dues. It has been my privilege to conduct a rather extensive correspondence with this smaller group. This has been, of course, in many instances of a confidential nature, but I am not violating that confidence in telling you that the opinion expressed above has resulted from this contact.

Nevertheless, we had to meet this problem. A survey indicated that we were spending a considerable amount of money annually in publications. While it is recognized that this is a legitimate and important function of the Society, conditions have changed; and the changes had not been reflected in the publications' policies. When the Society was first organized, it, in common with other engineering organizations, published its proceedings in the form of an annual or semi-annual volume containing complete papers and discussion of them. So long as there were no other or better avenues for the distribution of information to the members, this was very well. However, in later years, we have, through the activities of the Standards Committee, published the S.A.E. HAND-BOOK; later the *Bulletin*, which has developed into THE JOURNAL, was brought into existence. In THE JOURNAL we have a means of presenting to the members at a much earlier date than was possible in the TRANSACTIONS a complete record of the proceedings of the Society. Therefore, it seems that it would be highly inefficient for the Society to continue indefinitely to distribute the complete proceedings in THE JOURNAL and then at a later period duplicate this information in the form of a bound volume, without charge to the members in addition to dues.

There are many questions connected with this problem which it would be impossible to cover except in an inexcusably lengthy manner. Suffice it to say that, after long and careful consideration, the Council has finally decided that the TRANSACTIONS for the years 1921 and 1922 shall be sent to only those members who indicate that they wish to receive them. There will be no additional charge for these. After that time, it is the recommendation of your present Council, that the TRANSACTIONS be sold to the members at a nominal price which will partially cover the cost of production. This will permit several things: first, it will enable us to concentrate in greater degree on making THE JOURNAL more uptodate and complete in its record; and, second, it will relieve the finances of the Society of a burden that in a large degree under the old order was imposed for a service that was of little or no benefit to a large proportion of our membership. That this viewpoint is correct we believe is demonstrated by the fact that there were orders for only about 1200 copies of the last issue of the TRANSACTIONS.

I can appreciate thoroughly that a change of this nature will seem radical to some. I also appreciate the powerful influence of precedent and the fact that the receipt of bound volumes of TRANSACTIONS has long been a perquisite of members of engineering societies. On the other hand, this Society has in a large degree established itself and justified its existence as an organization on the basis of a disregard for precedent; and I

believe the other arguments that I have outlined herein are ample justification for the step that your Council has taken, and trust that the development of the plan will recommend itself to those of our loyal and interested members who have felt inclined to question the wisdom of the step.

The Standards Work

With regard to the Standards Committee, I think you will recognize that, in view of my long association with this phase of our work, I am most vitally interested in what is being done. We were fortunate this year in being able to get a complete working organization of the Standards Committee going very promptly. A considerable amount of work has been done, as evidenced by the reports that were presented to the whole Committee by the Divisions this morning. This part of the work speaks for itself, and I will not do more than make this reference to it.

There are, however, certain other phases of the Standards work to which I wish to call your attention. At the risk of being tiresome, I would repeat what has been said so often before, that the Standards work is one of the most important activities of the Society. It has been suggested that the direct benefits of this work have reacted in favor of the industry as a whole, rather than of the individuals who hold membership in the Society and from whom we obtain financial support in large measure, and that, in view of this fact, it would be well if means could be found that would place the financial burden for the support of this work on the shoulders of those who most largely benefit from it. While there is no doubt as to the soundness of these suggestions from the viewpoint of placing the burden of expense on the shoulders of those to whom the benefits accrue, there are other considerations which should have our thoughtful attention. I am placing before you herein what are largely my own opinions, and hope that you will recognize them as such.

I believe that the fundamental strength of the Society resides in its being an association of individuals, and that the strength of our position as an impartial agency for the conduct of many of our activities would be jeopardized were we to make provision for corporate memberships the main purpose of which would be to obtain revenue. We have been fortunate in having our work recognized by several trade organizations which have indicated their approval and support by making financial contributions. The degree in which these have come to us, and representing as they do, not individuals but groups, I believe is therefore the most practical solution for this problem. It would be of considerable assistance and I believe perfectly proper if this form of recognition were extended. However, whether it shall be or not, we should exercise all our ability and energy to proceed in a rational way to extend and continue the Standards work that was inaugurated about 12 years ago, and has been carried on continually since. Cooperative endeavor of this sort, bringing together as it does the individual members of the Divisions, is of the greatest benefit in promoting the development of the individual and the building-up of the spirit of service that is vitally essential to the health and growth of such an organization as ours.

I wish that we could find some way of still further impressing upon the industry the importance of this work; that we could find a successful method of definitely determining the degree to which the S.A.E. Standards are used and a more definite measure of the economies that their use brings about. While it is necessary for the purposes of efficient organization that the Divisions be not too large, I would like to see the time arrive when the Division meetings should partake of the nature of technical sessions to which not only the Division members but all interested members would feel free to come and would desire to come. One particular reason for this feeling is my belief that as we grow and continue this work we must be more and more particular that the subjects proposed for standardization are properly considered and thoroughly analyzed in view of the broadest possible experience and opinion, so that assurance may be had that all interests have been properly represented. I wish that we could individually make it our plan and purpose to sell the idea that it is good business and money well invested for organizations to give their engineers the time and to assume their expenses in the attendance at these meetings as well as those of a more general nature.

As we proceed with the work of standardization, we will encounter more and more difference of opinion as to how far it should be carried. There are those among us whose breadth of vision carries them far in the list of subjects that they believe can rationally be standardized. There are others who feel that these suggestions if followed would be unwise, and would result in harmful restriction of initiative in the design and development of our apparatus. I hope that these two views will always be in evidence, but that they will be brought into contact in the work of the Committee, so that they may temper each other and produce a rational result. I believe that any student cannot but recognize that it is in the reaction of the extremes of opinion in their contact with each other that sound and conservative policies are formulated.

In connection with the work of the Standards Committee, it is well for us to recognize the increasing recognition of the importance of such work in every quarter. In the January issue of the *Automobile Engineer* Basil H. Joy outlined the work of seven committees, operating under the British Engineering Standards Association, having to do with the general subject of automobiles. These subcommittees are dealing with nomenclature, steel, small fittings, electrical fittings, wheels, rims and tires, and cast iron. You will recognize that for practically all of these our Standards Committee has already brought into existence valuable standards. The Department of Commerce, under Secretary Hoover, in the Division of Simplified Practice is taking a very active interest, as the name of the Division suggests, in the simplification that can be obtained by the adoption and use of standards. You have had the opportunity of hearing today from Mr. Hudson of the Division exactly what its aims and ideals are, and we hope in cooperation with the representatives of the National Automobile Chamber of Commerce to be able to further this work.

Research

With regard to the work of the Research Committee, I feel that it would be presumptuous for me to attempt to make an extensive statement. We are devoting to this subject a session which, in conjunction with the report of Mr. Crane, chairman, and Dr. Dickinson, manager of the Department, will go farther than it would be possible for me to go, in outlining what has been done and what it is proposed to do.

My remarks on the work of the Standards Committee bear with equal force on the work of the Research Committee. The work in itself is largely of a character that will produce benefits that, in a considerable degree at least, will permit their being secured by others than the

individuals who comprise the membership of the Society. This viewpoint should not, however, blind us to the potential benefits that can accrue to the members. I say "potential" for the reason that the benefits will not be secured except insofar as the membership participates in the work and what each member will get out of it will, in a large degree, depend upon what he puts into it.

The results may not be startling in their scope at the present time, but I am firm in my conviction that the foundations that have been laid, if built upon with patience and with the consistent support of the membership, will in the very near future justify the inclusion of this work as a part of our regular program.

New Developments

After this more or less hurried summary of the affairs of the Society, I would direct your attention to a more general survey, with a view of determining along what lines our activities as engineers and as an engineering society should be directed in the immediate future.

The period of industrial depression through which we have gone should be productive of some lessons to which it would be well for us to give thought. Naturally, those that appeal to me most forcibly and which I feel most competent to discuss are those having to do with the truck rather than the passenger vehicle. There have been three outstanding developments during recent months, the appearance of which may be due in part to conditions resulting from the depression. They are: the speed-wagon, the motorbus and the motor rail car. That there is a fertile field of usefulness for all three of these types can probably be accepted without question. That they each present features of design requirements which are distinctive and possibly not yet fairly appreciated in general is, I believe, also true.

We held in January and will hold at this meeting a session dealing in a degree with the problem of bus transportation. There have been sessions held by the Metropolitan and the Indiana Sections that had to do with the matter of the motor rail car. The problem of the speed-wagon may be more commercial than technical, but I believe that it deserves consideration. I am mentioning these points with the hope that our Sections will find some suggestions for their development for meeting topics.

Highways

The question of highways is one that has been given considerable attention in the past in our discussions and should receive continuing attention. The ability and the efficiency of the vehicles that we construct are dependent in a large degree upon the character of the roads upon which they are operated. While it is true that the invention and development of the automobile has increased the demand for improved roads, it is also true that the growth of improved roads has increased the demand for and use of the motor vehicle, and future limitation in road construction will act as a limitation on the vehicle market.

It appears to me to be particularly unfortunate that there should be any controversy between the railroads and the users and builders of motor vehicles, instead of complete harmony and cooperation. Except in the most isolated cases, competition between these two forms of transportation is most unlikely. I think this is almost universally true with regard to transportation of goods; and in the transportation of passengers it is almost equally true if we stretch our imagination to embrace what must be the development of the future. I recognize the fact that there is a large amount of capital invested in street-railway transportation, but I am also impressed more and more daily with the fact that the streets of our cities are becoming less able to accommodate the burden of traffic that they are called upon to bear. It seems to me not at all improbable that this condition will make it imperative in the not very distant future to replace track vehicles with a more flexible form of vehicle for short hauls and where frequent stops are necessary.

This problem of highway capacity as evidenced by our city streets deserves the most careful study on the part of every automotive engineer, particularly as to what its probable effect will be on future design requirements as affecting the size of the vehicle, the control with respect to steering, turning-radius, acceleration and braking. In many of our cities very stringent regulations with regard to parking have been put into force. It is useless to spend our time in railing against these provisions, for in some measure at least they represent the legitimate effort to distribute the use of the streets in a fair way among all citizens. The problem presented is of the most complex nature and deserves careful study and analysis.

Another result of the increasing traffic-density is the lowering of the efficiency of motor vehicles as a means of saving time. As the cost of operation of motor vehicles has been reduced, and the possibility of use thereby increased, this new factor of limitation of speed, due to congestion, becomes increasingly important.

In the broader aspect of transportation in rural and suburban communities there should be practically no question of conflict between the railroad and the motor vehicle. We have in this Country a sufficiently close-up picture of the development of transportation facilities to be able to get a very comprehensive and intelligent view of the relation between various means of transportation and the establishment and development of communities.

The early settlements were along the seaboard and the more navigable streams, and this condition of affairs continued up to the time of the development of the railroad, which resulted in the unlocking of the vast inland empire and the linking-up of the Pacific coast with the Atlantic, which would have been practically impossible without this new means of transportation. The development of electricity and its application to high speed interurban lines was the next step in bringing high-speed transportation into closer contact with the small community and individual. It is obvious, however, that the operation of rail lines calls for a virtual monopoly of territory in the form of a franchise, and limits the operation of vehicles over any given track to one centralized authority, and calls for fixed schedules of operation.

The advent of the automobile has resulted in placing into the hands of the individual a smaller and more flexible unit with practically the equivalent speed-capacity of the railroad. This vehicle, capable of being operated over the road, can be made more truly competitive and infinitely more flexible and independent of fixed schedules. The growing use of the automobile and the truck, coincident with the development of and as an auxiliary to the railway system, has resulted in extensive suburban and rural development which would probably have been as impossible without the automobile as the development of the inland cities of this Country would have been without the railroad.

While this development has resulted, and the increase in realty value is recognized and acknowledged, the in-

(Concluded on page 26)

Principles of Motorbus Design and Operation

By G. A. GREEN[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* PHOTOGRAPHS AND DRAWINGS

IN the paper an attempt is made to answer the broader phases of the questions: What constitutes a bus? and In what respects does a bus differ from other classes of automotive equipment? by establishing the principles on which the design and operation of motorbuses should be based. The treatment of the subject is in the main impersonal, although specific references to the practice of the Fifth Avenue Coach Co. and illustrations of its equipment are made to emphasize the points brought out. The questions of the unwisdom of overloading, rates of fare and the service requirements are discussed briefly as a preface to the paper proper.

The factors controlling bus design are stated to be (*a*) safety, (*b*) comfort and convenience of the public and (*c*) minimum operating cost. The various subdivisions of each are commented on in some detail, and numerous illustrations and tabular data supplement the text. The conclusions reached are that trucks or automobiles, either modified or unmodified, are absolutely incapable of rendering satisfactory and economical service as buses; such failures of buses as have occurred were due to the combination of extemporized equipment, indiscriminate operation, overloading and lack of experience; and, if the Society would concentrate its standardization work on the motorbus, much good could be accomplished.

THE questions that builders and intending operators are asking today are, What constitutes a bus? and In what respects does a bus differ from other classes of automotive equipment? There seems to be a general agreement that a properly designed bus has special requirements; that it differs materially from equipment such as trucks and automobiles.

I have been requested to give the Fifth Avenue Coach Co.'s views on this subject. It is, of course, possible to deal with only the broader phases. No attempt will be made to discuss detail design, but merely to establish the principles on which it is thought such design should be based. We believe that with problems of this character, it is principles that really count, that once having clearly established them, the rest is comparatively easy. Actually, there is no real mystery in motorbus design. It is purely an engineering problem and there is available ample engineering talent to afford its solution, but the principles must first be established.

In the preparation of this paper the underlying thought has been to treat the subject in an impersonal manner. Illustrations and specific reference have been made to our practices only when this has appeared to be the simplest and most direct method of approach.

THE UNWISDOM OF OVERLOADING

We believe this question is of paramount importance, not only to the automotive industry but to all who are contemplating bus operation in any form. Our policy is predicated on a seat for every passenger. At the inception of our business this was our slogan. We have never departed from it and we never expect to do so. We are convinced that this policy has been, perhaps more than anything else, a factor in the building up of our enterprise.

It is, of course, possible to carry a certain percentage of standees in a vehicle, the spring-suspension of which has been correctly designed to carry properly a seated load. In our judgment, however, this figure should not exceed 30 per cent. But even this is unsatisfactory, for once standees are permitted, their limitation is most difficult.

Obviously, the problems requiring solution from the standpoint of spring-suspension are much less numerous with vehicles operating on rails than is the case with rubber-tired equipment running over roads. With the former, overloading has no immediate serious consequences—at least from the standpoint of the rolling stock. The spring-suspension with a bus must of necessity be a compromise between minimum and maximum loads. If the range is too wide, bad riding conditions must obtain during by far the greater percentage of the total time, for the packed loads will, generally speaking, occur only during the rush periods. This means that 90 per cent of the time there will be a state of discomfort. This will have an extremely bad effect on both the vehicle and its occupants. Another vital point to consider is that a bus is not kept in a comparatively straight and rigid course by steel rails. The advantageous flexibility of a bus in steering its course at will has its disadvantages if standees are permitted, for the shifting of the weight of the standees when the bus swerves tends to make it unsafe, throwing the passengers about inside the vehicle and rendering the operator liable to heavy damage and accident suits.

We are unqualifiedly behind any movement that will aid the bus to come into and remain in the field that is peculiarly its own. We are positive that the short road is the seated load and if builders will bear this in mind from the standpoint of design and warranty, the automotive industry will assuredly find ample repayment.

We earnestly hope that the automotive industry will read the writing that is so plain to see and that it will profit by what has occurred with the street railways, in regard to the matter of overloading. For it must be remembered that the bus has its limitations and that it is not the cure-all for every ill that transportation is heir to.

THE MATTER OF FARES

Strictly speaking, there is no actual relationship between the design of a bus and the fares charged to passengers. Obviously, however, the better the design, the lower will be the operating cost. Naturally, this will make for lower fares. We believe that in the present state of the art no real success can be attained with less

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than a 10-cent fare. We are, of course, assuming operation based on seated loads and ample service during both the light and the heavy hours. But with character service, properly designed and maintained equipment, the people are quite willing to pay a 10-cent fare. There is ample evidence of this in New York City, Detroit, Chicago, Toronto, and other cities.

The necessity for a 10-cent fare does not rest with only the bus. Many electric railways need a 10-cent fare in order to be put on a paying basis. The last available tabulation shows that 140 electric railways in the United States are receiving a 10-cent fare, and that over 95 per cent of the electric railways in the cities of the United States have received varying increases in fare during the last few years. Some cities have a first fare of only 6 or 7 cents, but to this must be added a charge for transfers. Many cities have been placed on the zone system that works out in some cases as high as 3½ cents per mile. Even with an increased fare, the last available figures show that about 10 per cent of the electric railways in the United States are in the hands of receivers.

It is not the purpose of this paper to enter into a lengthy discussion of operating costs, for unless this matter is treated in considerable detail, accurate deductions are almost impossible. Obviously, a correct comparison of operating expenditures can be made only on the assumption that similar detail classifications are employed in conjunction with a similar accounting system. Here the difficulties begin, for as yet few companies operating buses use the same accounting methods.

No doubt there are many who, while not desirous of making a minute survey of details of operating costs, would be interested in knowing something about this rather complicated matter other than mere expressions of opinion. For this reason there is shown in Table 1 not the customary detail cost statement, but what might be described as an income analysis. Actually it represents a distribution of the dime as received from each of those who rode on our buses during the year 1921.

TABLE 1—DISTRIBUTION OF EACH FARE RECEIVED

	Cents
Total Operating Expenses	6.50
Total Taxes	1.16
Reserved for Injury and Damage Claims	0.17
Reserved for Depreciation	0.29
Interest on Capital Investment	0.39
Net Income	1.49
Total	10.00

From these figures it is abundantly clear that we should have made a very bad showing with a fare of less than 10 cents. Here is emphasized very clearly the fact that the success of failure from the standpoint of an undertaking such as our own depends absolutely on the addition or subtraction of what at first sight appear to be insignificant amounts. To emphasize this point, during 1921 we carried a total of 52,216,946 passengers, so the net income from this source at 1.49 cents per passenger works out at $778,032.50. To permit of a comparison being made between the conditions confronting us and those faced by others, it should be noted that we operate a total of 25 miles of one-way route, that our longest run is 10.2 miles and our average haul 5.0 miles.

The Bus and Its Service Requirements

Before discussing the bus from a design standpoint, something may be gained by outlining the character of service that must be expected, for it is here that the average engineer underestimates the difficulties to be encountered. First, let us consider the cumulative result of a year's performance of the physical limitations that are primarily responsible for wear-and-tear. For the sake of argument it may be assumed that these data are applicable to any bus operated by any public utility. The figures are presented in Table 2.

TABLE 2—DATA ON BUS OPERATION IN NEW YORK CITY

Yearly Mileage	30,000 to 60.000
Stops and Starts	180,000 to 360,000
Change-Speed Applications	360,000 to 720,000
Clutch Applications	360,000 to 720,000
Different Drivers	1,095 to 2,190
Brake Applications	200,000 to 400,000

Assuming the same general plan of upkeep as employed by the Fifth Avenue Coach Co., each bus would be thoroughly inspected after every 2000 miles of operation and rebuilt and repainted yearly. A vehicle would be expected to require no incidental repairs between inspectional periods and no major repairs between either inspections or yearly overhauls. The inspectional periods would occur approximately every 14 days. The maximum inspectional allowance is 8 hr. The allowance for yearly overhaul is 7 days. Roughly, it may be said that under these conditions, each bus is scheduled for service 358 days out of 365.

The statistics quoted as to mileage, stops and starts, and the like, speak for themselves. Those who have never had control of a public utility operating buses cannot possibly picture the sum total of the abuse the average bus must suffer. More than anything else, frequent changes in drivers result in increased service difficulties. It may be safely said that if one could with a bus have the same driver daily, at least 50 per cent of the service troubles would disappear. This, however, is quite impractical, since the loss in earnings would many times offset the decreased service cost. Even with an operation of moderate size, the bus must of necessity lose its identity. It becomes merely a transportation unit. There must be changes in drivers daily, many of whom will feel scarcely any pride of ownership. All they are concerned with is being on schedule time. This means that the bus will be subject to extraordinary abuse. The mechanisms of the bus must be capable of treatment of the most brutal nature; otherwise constant failures will occur.

Before one can proceed very far from a design standpoint, there must be some fairly clear conception of the vehicle life that is to be expected. In this connection it is necessary to lay stress on the fact that motorbus design is still in its initial stages. Five to 7 years is about the maximum life of the most modern type. It is not a matter of wear-and-tear, for a vehicle may be so well cared for that there is no limit to its life. Obsolescence is the real issue. The ideal conception is to carry out the design so that the various units which when assembled comprise the complete structure, have as nearly as possible an equal life.

Controlling Design Factors

In its broadest sense we believe the controlling design factors from the standpoint of the motorbus, in the order of their importance, are

(1) Safety
(2) Comfort and convenience of the public
(3) Minimum operating cost

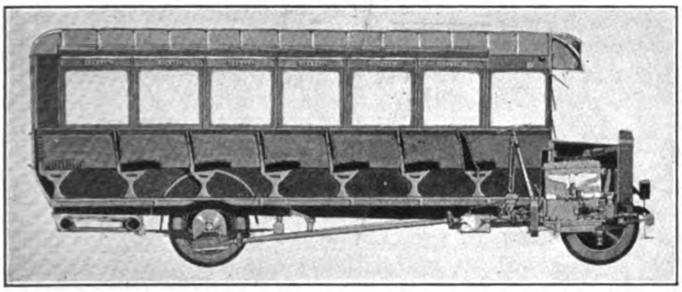


Fig. 1—Sectional View of the Type-J Bus

Safety easily heads the list and a very large proportion of the engineering development work must be concentrated under this heading. It is generally agreed that a truck carrying freight should be in all respects safe, and that every reasonable precaution should be taken to render automobiles transporting from 1 to 7 passengers safe; so how much more important is it that a vehicle carrying 50 or more passengers should be free from every sort of hazard! It must be remembered that much of the mileage of the bus is through congested thoroughfares. This is not the case with the average automobile or truck. Again, the average individual makes some effort to get out of the way of a truck or automobile, but the bus, with its acknowledged flexibility, is supposed to move out of the paths of both vehicles and pedestrians.

The design of a motorbus from a safety standpoint includes certain basic features which must be incorporated in the general constructional plan. There are also other detail features which must be included. The latter are dictated by human considerations. Reference is now being made to providing the driver with reasonable comfort and convenience so that no undue hardship will be inflicted upon him as a result of the performance of his duties. First, let us consider the former. These are

(1) Low center of gravity
(2) Wide frame, track and spring centers and general dimensions
(3) Effective brakes
(4) Short turning-radius

Low Center of Gravity

Beyond doubt, the future bus will be low hung. The inherent danger in connection with any other form of

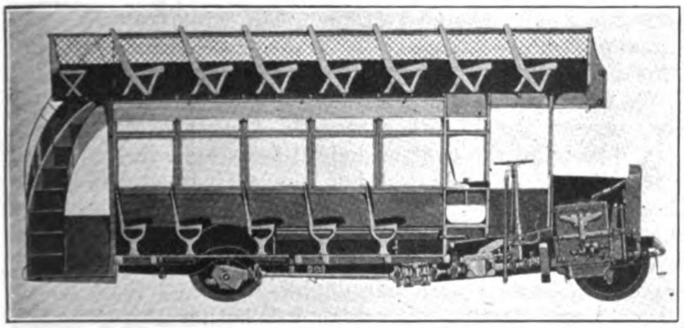


Fig. 2—Sectional View of the Type-L Bus

construction is the possibility of overturning. Under conditions of proper operation, the hazard may be nonexistent, but we have always before us the possibility of human failure. Actually the danger is much more real than apparent. The controlling element governing overturning is centrifugal force. Vehicles seldom if ever overturn as a result of high speed and sudden impacts or brake applications. Overturns are almost invariably due to a combination of speed and turning-radius. The only reliable guarantee against this class of accident is a low center of gravity.

In many cities there are overhead wires and various other obstructions. The low bus is often a necessity to pass under such obstructions. Certainly, the lower the vehicle, the less the hazard. These remarks apply particularly to double-deck vehicles. With the single-deck vehicle, the higher speed is a factor that must be fully taken into account. Entirely apart from the matter of safety, a low-hung vehicle has a more graceful appearance. There is less time lost in boarding and alighting, there are fewer boarding and alighting accidents, and the schedule speed can be faster. Lastly, assuming proper

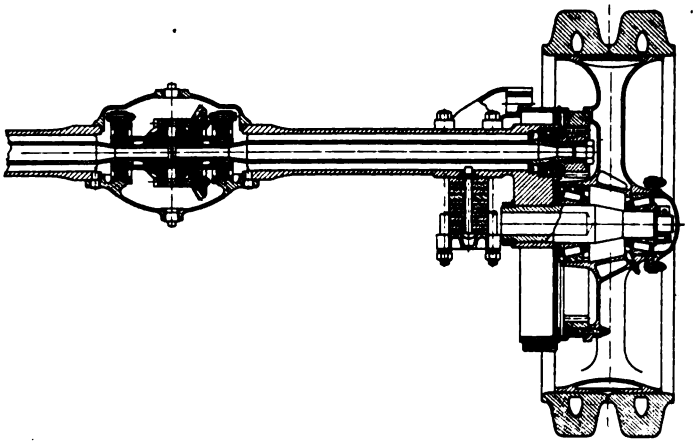


Fig. 3—Sectional Drawing of the Type-L Axle

design, a low center of gravity results in improved riding properties.

We have found that a safe and practical height of the frame from the ground for a single-deck bus is 25 in. and for double deck bus, 18 in. The center of gravity of our type-L double-deck vehicles, with a full complement of passengers on both decks is 52 in. from the ground. With our type-J single-deck bus, this dimension is 38 in. It is interesting to note that when rounding corners, even at a high rate of speed, skidding will occur due to centrifugal force and overturning is scarcely possible. Furthermore, rolling or sidesway is practically eliminated. The sectional views of our J and L-type buses reproduced in Figs. 1 and 2 indicate clearly how this condition has been reached. With type L it will be seen that the frame and rear-axle construction is somewhat unconventional. The rear axle is of the internal-gear type. The spiral bevel-gear and differential assembly is in unit form and can be entirely assembled and adjusted on the bench. The carrying member is a heat-treated forged job.

From the sectional drawing shown in Fig. 3 the general construction of the type-L axle will be clear. It

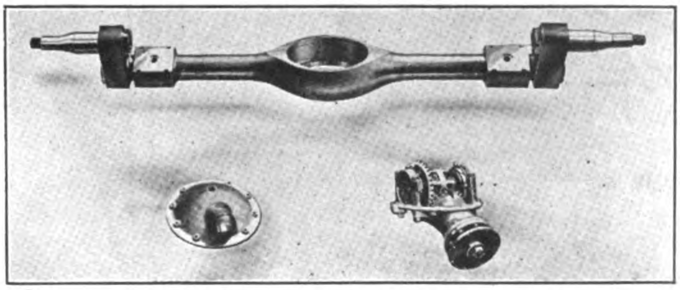


Fig. 4—The Type-L Rear Axle

will be seen that the ends of the carrying member are cranked, the wheel spindles being above the drive-shaft center-line. It is in this manner that the low-level feature has been accomplished. The photograph showing the carrying member and driving-gear assembly which is reproduced in Fig. 4 at once emphasizes the general simplicity and accessibility of construction. Due to the fact that the drive-shaft pinions are in the vertical plane, a special form of tooth has been developed for the internal gear to provide adequate clearance and at the same time permit of maximum silence even after a certain amount of wear has occurred.

We do not employ this special form of axle construction for the type-J bus. This class of vehicle will have a much wider use; therefore, the matter of road clearances must be taken into account. In many cases single-deck vehicles will be operated over very bad roads. The double-deck vehicle is essentially a city job where the streets are, generally speaking, in fair condition. Again, with the single-deck vehicle, the floor-level requirements are not so exacting. There is no top deck to take care of, and the entrance can therefore be located at the front end of the bus; but with the double-deck vehicle, conventional practice is to have the passengers enter at the rear, so in passing to the interior they are obliged to cross the rear axle which must be of special design to have the floor level within easy stepping distance of the ground. In the case of the single-deck bus it is not desirable to have a step 18 in. high. Therefore, the best plan appears to be to employ an orthodox rear-axle design. Even assuming the use of our type-L rear axle, it would not be practical to produce a stepless vehicle. The

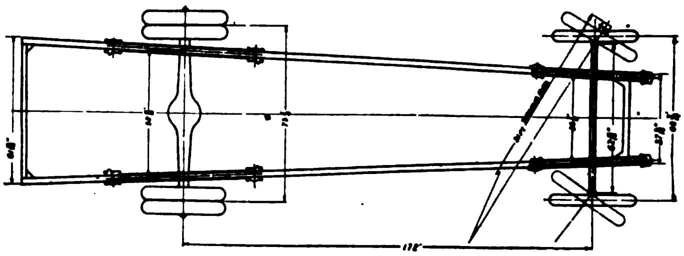


Fig. 6—Chassis Dimensions of the 29-Passenger Single-Deck Bus

appearance would be completely spoiled and, as explained above, the ground clearance would be cut to a point where the vehicle would be unsuitable for use in many localities. Of course, a stepless single-deck vehicle can be produced, but its practical value for general utility purposes is debatable.

Among the constructional difficulties in connection with the production of low-level equipment, one of the problems is to obtain a flat floor. There is a natural tendency for the components to project above the frame and therefore through the floor. To avoid this, special design is required. The effect of a flat floor is very pleasing to the eye. Its structural strength is greater. It is less costly to keep in repair and there is less possibility of accidents due to the passengers' feet coming into contact with the obstructions during the boarding and alighting processes. The view through the door of our type-J vehicle, Fig. 5, brings out this point to advantage.

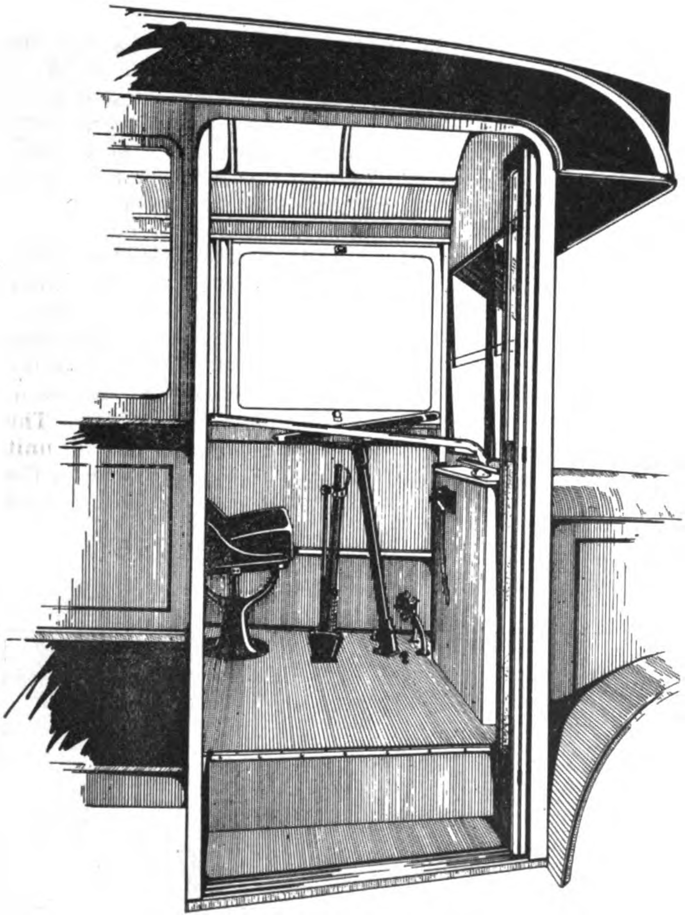


Fig. 5—View through the Door of the Type-J Bus

Wide Frame, Track and Spring Centers

These features are necessary to provide for adequate vehicular stability and, in conjunction with a low center of gravity, make for maximum safety. The necessity of providing proper stability applies equally to single and double-deck vehicles. It may be said that the added risk due to the top-deck load with the latter is more than equalled by the faster speed of the single-deck unit.

Apart from the matter of safety, a wide frame is necessary in connection with the body construction. Obviously it is desirable to support the body as far out as possible, for in all cases the seating arrangement is such that the passengers are grouped about the outer edges. Then, the wide frame admits of the lightest possible form of body under-frame. The wide frame also is a factor from the standpoint of the passengers' comfort. This point will be referred to later.

We believe that the overall length of a motorbus for city service should not exceed 26 ft.; the total width, 7 ft. 6 in.; and the over-all height for single-deck vehicle, 9 ft. With the double-deck bus, the last-named dimension should be such that a person standing on the top deck can clear a 14-ft. structure. With these dimensions we have found it possible to accommodate comfortably 51 seated passengers with our double-deck, and from 25 to 29 with our single-deck vehicle. Whether this practice is economically correct for all localities, we cannot say. We have, however, up to the present found that this arrangement works out very well both in our own service and in the service of those who have purchased our equipment.

Next, there is the question of important dimensions other than those over-all, such as the wheelbase which naturally affects the axle load distribution, the turning-radius and the general comfort and balance of the vehicle. For the class of vehicle now under discussion, we believe

that this dimension should not be less than 168 nor more than 180 in.

The front track should be ample in width and not less than 67 in., for to turn a 'bus within the intersection of the average city street, it is necessary to move the front wheels through an angle of not less than 35 deg. This determines the distance between the front-axle pivots and the springs. The spacing of the front springs should not be less than 36 in., since they are responsible to a large extent for the stabilization of the vehicle when turning a corner.

Regarding the rear track, we believe that the outer edge of the tires should closely correspond to the extreme over-all width of the body and that the rear springs should be as close to the tires as is practical. For buses as above described, the rear track should not be less than 72 in. This will bring the distance between the springs to approximately 52 in. Having decided the approximate distance between the vehicle springs, it naturally follows that the best design is to arrange the frame dimensions so that they connect with the springs in the closest and most practical manner. Our practices in regard to these matters may be readily followed from the diagrammatic sketch of the type-J chassis as shown in Fig. 6.

Effective Brakes

Perhaps the most difficult problem that engineers must face is the brake question. Even now it has not as yet been solved entirely satisfactorily, at least insofar as our knowledge goes. With the bus, the number of applications is in excess of that of the average truck or automobile, and the brakes of a bus must be sufficiently powerful to lock the wheels at any moment. Yet the effort required for average application must not be such that a driver may become exhausted as a result of the work imposed upon him.

Particular attention must be paid to the location of hand-brake lever. It should be positioned so that it can be grasped firmly without moving the body out of the normal seated state. We believe the best practice is to have the lever arranged for a push and not a pull-on. Time can thus be saved, and a fraction of a second is often the determining factor from an accident-prevention standpoint.

The brakes of a bus must be free from undue noises such as squeals or rattles. This means, among other matters, the use of special brake-drum material. The conventional soft pressed steel is practically useless. The best plan is to employ treated steel forgings or, failing in this, steel castings with a high carbon-content.

The friction surfaces must have long life, and the adjustment be such that no tools or special skill are necessary. We attach considerable importance to the matter of foolproof adjustment. The J system as illustrated in Fig. 7 shows our method. It will be seen that there are two vise-like levers. The outside controls the hand, the inside the foot brake. One turn is usually sufficient. If by any chance the levers are not returned to the vertical, they will automatically reach this position by force of gravity.

The braking action must not be too abrupt. It must be positive yet not sudden and violent, for such a condition is exceedingly severe on the driving members, tires and body. It is also a frequent source of accidents from which serious claims may result. Brakes must be sufficiently good, yet not too good. Excessively efficient brakes have a most marked influence on tire wear. It may be said that tire wear is almost directly proportionate to the effectiveness of the brakes.

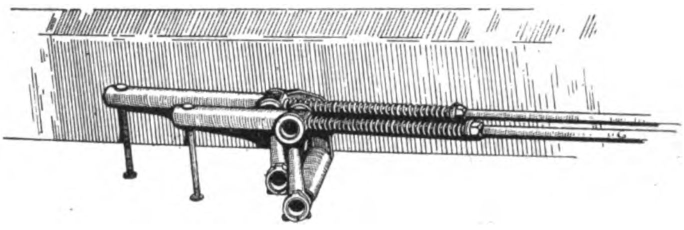


Fig. 7—Outside Brake Adjustment

In bus operation it is desirable from every point of view to cover the route as quickly as safety will permit. In this manner the maximum number of passengers can be carried daily. With a fixed maximum-speed, this means fast deceleration and acceleration. Expressed in another way, the problem is to move from a stop in one location to a stop in another in the least time. In our own service this must be done without exceeding a speed of 15 m.p.h., or accelerating or decelerating faster than 2 m.p.h. per sec. A still more rapid rate of deceleration is, of course, available for emergency, but it will be uncomfortable and unsafe, especially for standees.

The acceleration and deceleration graph as reproduced in Fig. 8 shows how closely the present type of equipment approaches this conception. To make the test, one of our double-deck buses was selected at random.

Short Turning-Radius

One of the great advantages of a bus over any other form of transportation unit is its flexibility. A bus can be switched around at any point, and it is highly desirable that it should be able to make a complete turn in the average thoroughfare without backing, for the latter practice if followed in congested areas merely adds to both confusion and congestion. There is also a marked possibility of an increased number of accidents.

A short turning-radius is dependent on the interference of the tires with the drag-link, front springs or frame,

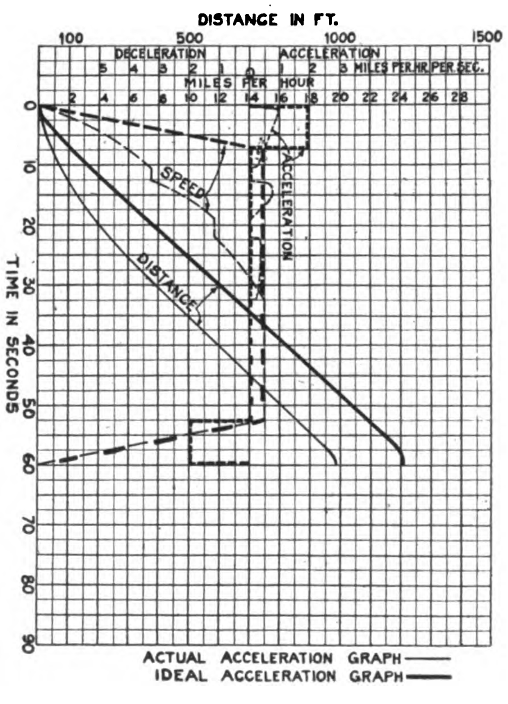
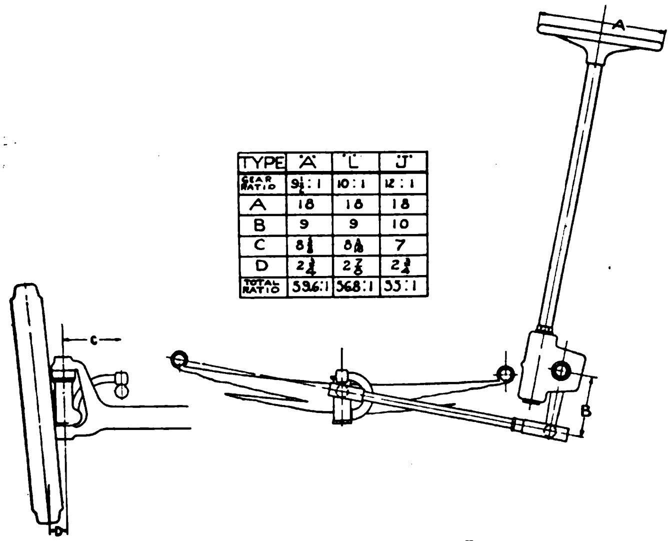


Fig. 8—Curves Showing the Actual and Ideal Acceleration



TYPE	"A"	"L"	"J"
GEAR RATIO	9½ : 1	10 : 1	12 : 1
A	18	18	18
B	9	9	10
C	8⅜	8 5/16	7
D	2¾	2⅞	2¾
TOTAL RATIO	39.6 : 1	36.8 : 1	33 : 1

Fig. 9—Diagram of the Steering Leverages

when the wheels are turned at the maximum angle. The controlling elements are wheel-spring tracks and wheelbase. As the radius of the steering angle equals the wheelbase divided by the sine of the front-wheel lock, it can be seen that a wheelbase of reasonable length is important to secure a short turning-radius.

From the viewpoint of safety, the design features dictated by human considerations are

(1) Easy steering
(2) Clear vision for driver
(3) Comfort and convenience for driver

Easy Steering

The steering of a bus should be at least as easy as that of the average automobile. To operate a stiff steering-gear is a hardship that certainly should not be inflicted upon the driver of a public-service vehicle. A driver's energy and effort must be concentrated on his regular duties, and if he becomes fatigued through the expenditure of unnecessary effort, faulty operation is bound to result. This means possible accidents. Tests have convinced us that the actual physical labor imposed on the driver of a bus in connection with the manipulation of a steering-wheel represents by far the greater proportion of the sum total of his work.

Ease of steering is controlled by the total ratios between the hand and road wheels. Naturally frictional losses in the steering-gear box and steering-knuckles are of importance. Minimum losses in these respects are dependent upon the use of properly lubricated anti-friction bearings. Another very important matter is that the steering-knuckle pins should lie in the vertical plane; otherwise there will always be a tendency to lift the front end of the bus when turning the steering-wheel. An angle in either the longitudinal or transverse plane will cause lifting at the expense of effort on the part of the driver.

Fig. 10—A Type-J 25-Passenger Single-Deck Bus

It is highly desirable that there should be an absence of shocks at the steering-wheel. This is largely controlled by the total ratio, but also by the distance between the point of contact of the wheel and the road and the intersection of the knuckle center-line and the road. Every effort should be made to keep this distance small. With the J type the length of the lever arm is about 2¾ in.; and an increase of only 1 in. would decrease the total ratio some 36 per cent. This is the only point in the steering linkage where a change increasing the total reduction does not result in increased steering-wheel travel for a given lock. A short drag-link or the incorrect alignment of the drag-link with the front springs will also result in shocks at the steering-wheel when passing over rough roads.

Minimum steering-wheel travel is important as it makes a change of hand position unnecessary for ordinary driving. It also decreases the apparent back-lash, which is present in all steering mechanisms. The steering-wheel travel is roughly inversely proportional to the total ratio, which is kept as low as possible for this reason. Our practice so far as the important dimensions referred to above are concerned may readily be followed from an examination of the diagram of steering leverages as illustrated in Fig. 9.

Clear Vision for Driver

This very important feature can be accomplished only as a result of joint chassis and body design. The driver should be located close to the left-hand side. This permits him to observe and also to signal his intentions to oncoming traffic. There should be absolutely nothing obstructing his view. He should face clear glass. It should also be mentioned that with single-deck vehicles the placing of the driver well over on the left-hand side provides for the very necessary boarding and alighting space for passengers and adequate room for operation of door.

Briefly, a driver's vision should be such that when seated, even back of a closed windshield, he will have nothing on which he can readily concentrate, no vertical posts or obstructions of any kind. He should just naturally sense that he is in the open. The illustrations of the front end of our type-J bus reproduced in Figs. 10 and 11 bring out this point with marked clearness.

Comfort and Convenience for Driver

This is largely a question of seat formation in conjunction with the correct positions for brake, change-speed levers, pedals, accelerator, etc. Obviously, it is not a practical matter to give the driver of a bus as much room as with a touring car; therefore, much care and thought must be paid to the placement of pedals and levers. The conventional cowl as used in automobile practice is almost out of the question, for anything that tends to increase the over-all length of the vehicle is distinctly undesirable, particularly if such increases add nothing to the passengers' seat or pay-load space.

The driver should be comfortably seated at all times.

He should be able to reach his change-speed or brake levers without body movement. He should have ample leg-room and not be obliged to cramp his limbs when his feet are either on or off the pedals. To some extent this point is brought out in Fig. 5. The value of the flat floor from the standpoints of both passengers and driver, is apparent; also the side control without which there is of necessity a considerable loss of most valuable space.

Comfort and Convenience of the Public

The American public is automotively inclined and the percentage of those owning cars is so large that when riding in any self-propelled vehicle, there is a natural tendency to compare its behavior with that of an automobile. In designing a bus this factor must under no circumstances be lost sight of. The success of any public utility depends on the good will of the public. It has been correctly stated that the permanence of any business depends upon the good will of those it serves and

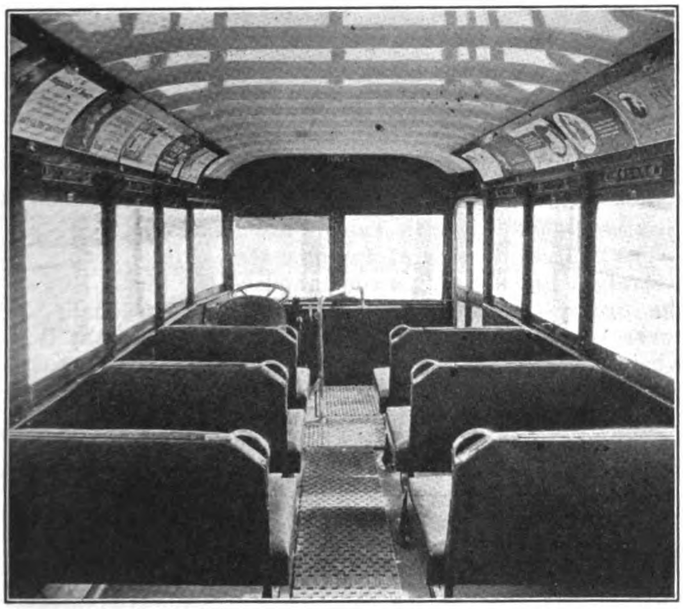


Fig. 11—View from the Rear End of a Type-J Bus

that no business can achieve permanent success that does not give in exchange for its earnings at least an even measure of helpful service. This applies especially to public utilities, and the truth has been abundantly proved in connection with the operation of our enterprise.

From the viewpoint of design, it is essential that consideration be paid to the attitude of the public as a whole. It is not enough to consider only the attitude of the actual riders; regarding the matter of comfort from these somewhat different angles, it is necessary that attention be given to

(1) Riding ability
(2) Reliability
(3) Silence of operation
(4) Smoothness of starting and stopping

Riding Ability

Broadly, this is a matter of proper spring-design. There are, however, other important influences; the wide frame, track and spring-centers bear materially upon this question, for the nearer the wheels are to the outer edge of the body, the less will be the movement to which passengers must be subjected when obstacles are passed over. Again, with the wider track, many of the ruts and

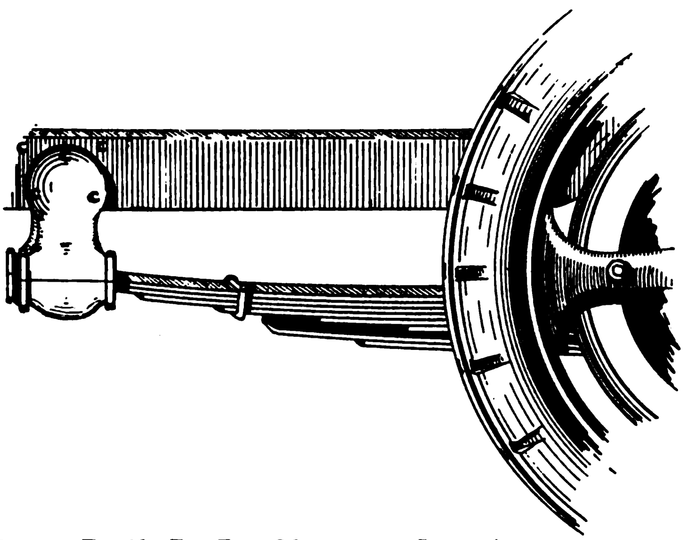


Fig. 12—The Type-J Progressive Spring Arrangement

depressions created by vehicles of narrower gage, will be passed by. Incidentally, this is quite an important matter from the standpoint of road wear. The wide track also diminishes the wheel-pocket projection inside of body. The modern tendency is to employ cross seats and with the narrow-gage vehicle the wheel pockets are a source of much discomfort to those seated upon the inside immediately over them. A rigid frame, correct axle-load distribution and minimum overhang are all factors that make for better riding performance.

Apart from the points briefly touched upon above, the controlling factor from the standpoint of riding ability is, of course, the design of the suspension itself. Obviously, the difficulty is to obtain good riding under all conditions of load. Spring design is always a compromise; a spring must be able to withstand maximum load, yet vehicles are expected to ride reasonably well when light. As a matter of fact, they seldom, if ever, do so. In general, more damage is done to vehicles when running light than heavy because the riding properties under these circumstances are at their worst and the speed too often is high. Under conditions of heavy load, springs function best, and at the same time there is less likelihood of excess speed.

We believe that the answer will be found largely in

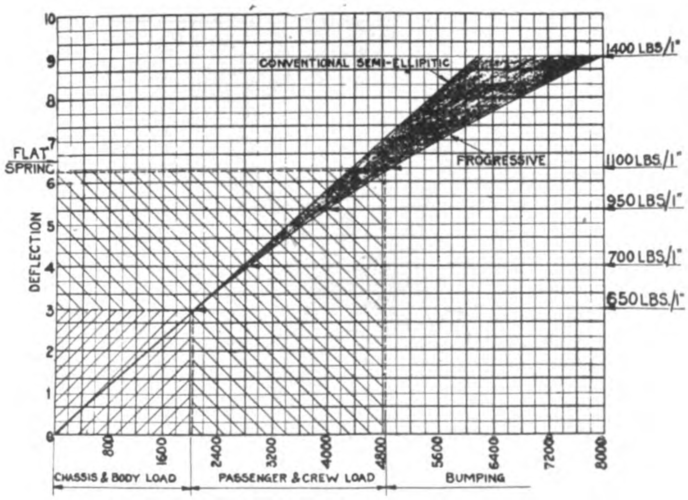


Fig. 13—Curve Showing the Comparative Deflections of the Progressive and Conventional Semi-Elliptic Rear Springs

the employment of what we term the progressive spring as illustrated in Fig. 12. It will be seen that spring is split into two parts. The top half takes the weight of vehicle, body and a certain proportion of load. The bottom part or helper, comes into action progressively. The top part must make a rolling contact with the bottom. One of the great advantages of this system is the fact that for no additional cost or weight, a marked improvement in performance is possible. The theory behind our choice of the progressive spring and the advantages that may be derived from its employment can readily be seen from an examination of the rear-spring deflection curve for both the progressive and the conventional semi-elliptic designs reproduced in Fig. 13. No doubt it will be appreciated that to secure comfortable riding with a small number of passengers, it is necessary to have a spring of not over 670-lb. per in. deflection. But a spring having these characteristics is not a practical arrangement, for the result would be too great a difference in

TABLE 3—DEFLECTION FOR PASSENGER LOAD

	Conventional Semi-Elliptic Spring	Progressive Spring
Full Passenger-Load	4¼	3¼
Maximum Bumping-Load	8½	6¼

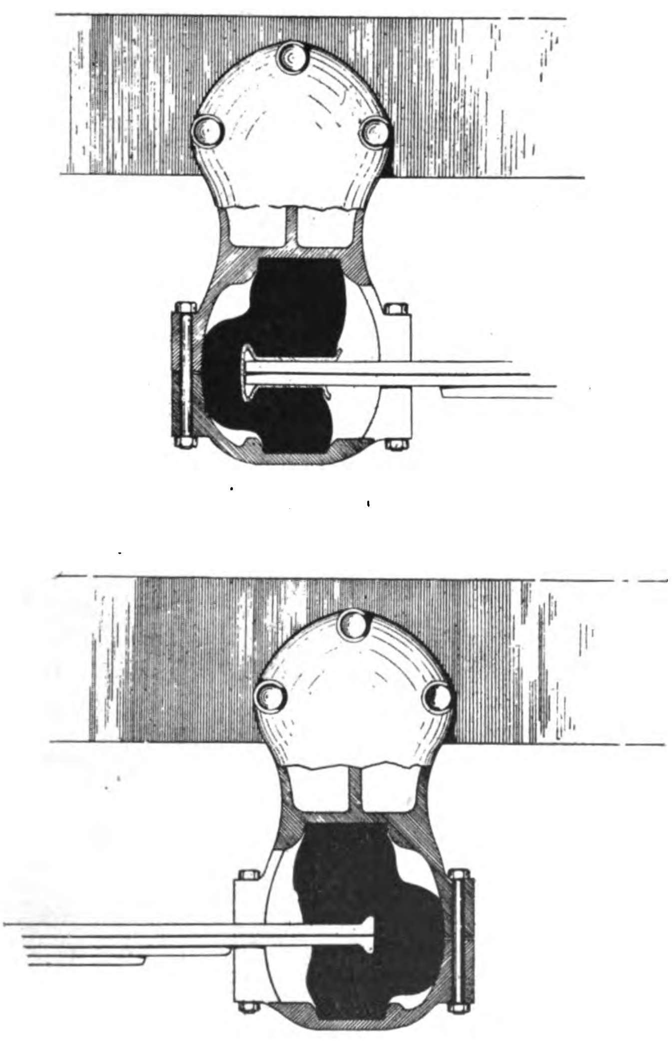


FIG. 14—RUBBER SPRING-SHACKLES USED ON THE TYPE-J SPRING SUSPENSION

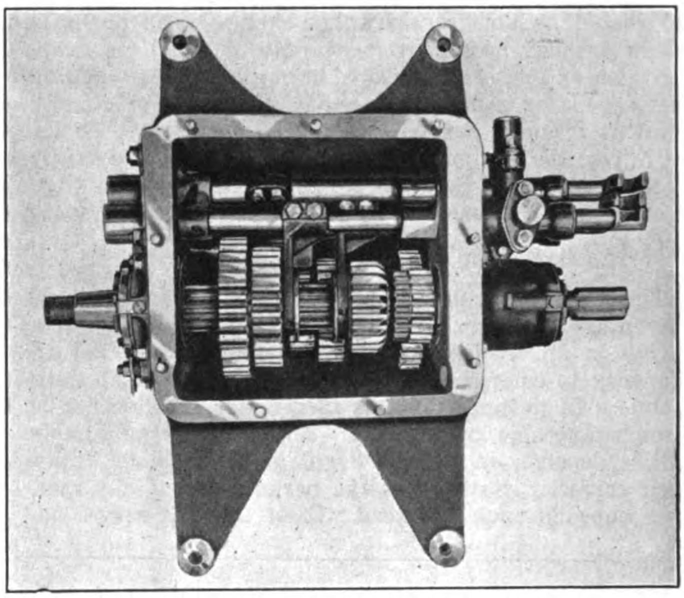


FIG. 15—THE FOUR-SPEED GEAR TRANSMISSION

body and step height between the minimum and maximum number of passengers. This point is clearly shown in the graph where the proportion of the 51-passenger load equals 2800 lb. per rear spring, from which the comparative figures given in Table 3 are deduced.

The deflection curve of a simple semi-elliptic spring is a straight line showing a constant load per inch. But as the progressive element comes into play gradually, a curve is apparent. The departure from a straight line which is shown shaded represents the load carried by the progressive element which can be designed to come into action at any desired point. It has been found most satisfactory to design this spring so that the stiffened action begins very gradually, that is to say, after a limited number of passengers have been taken on. Obviously, as the progressive element comes into action, there is a gain in the stability of the vehicle.

From the graph above referred to it is exceedingly interesting to note the change in rate of progression as a result of a variation in passenger load. The figures based on increments of 10 passengers given in Table 4 bring this point out in a striking manner.

TABLE 4—CHANGE IN RATE OF PROGRESSION FOR VARIATIONS IN LOAD

No. of Passengers	Load per 1-In. Deflection, lb.	Increased Stiffness, per cent
0	670	0.0
10	780	16.4
20	810	20.9
30	850	26.9
40	900	34.4
50	1,080	61.3

For our single-deck equipment we have standardized the Mack type of rubber shock-insulator which is illustrated in Fig. 14. This is by special arrangement with the Mack company. We are experimenting with this device for our double-deck vehicle but as yet are not prepared to state the results. This arrangement, in conjunction with our progressive system, markedly improves the riding conditions. It also avoids the necessity for lubrication and for replacement of shackles, shackle-pins and bushes; also, no spring-eyes are required. Experi-

ence up to the present shows that we may expect a very satisfactory life from rubber blocks.

Silence of Operation

It is a problem to produce a silent vehicle. It is doubly a problem to retain this state throughout the life of the vehicle. Silence necessitates freedom from engine vibration, quiet transmission gears, evenly stepped gears, a quiet rear end, and generally the elimination of all rattles

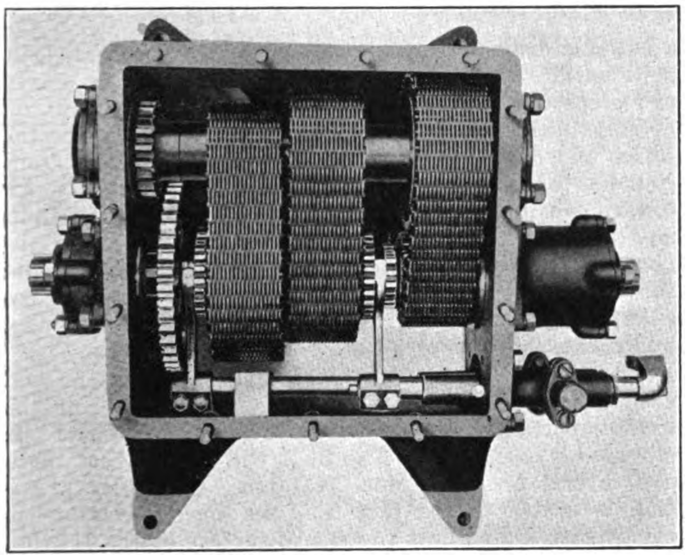


Fig. 16—The Three-Speed Chain Transmission

and squeaks from both body and chassis. To attain this, every detail of design must receive the most minute care. Silent operation is necessary in crowded thoroughfares, and certainly the people demand this condition in the residential areas, particularly at night when the streets are comparatively empty and noises become automatically emphasized. As a rule, noises are tolerated simply because such things are nearly always with us, but in the quiet of the evening sounds that ordinarily pass unnoticed become startlingly evident. In connection with the general question of noise it is interesting to consider

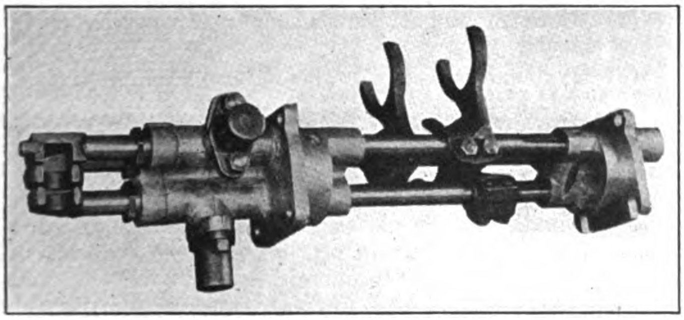


Fig. 17—The Shift-Rod Assembly of the Four-Speed Transmission

for a moment conditions on Fifth Avenue in the rush period during which we operate 180 buses per hr. in each direction. If this vehicular volume were not reasonably quiet, we should soon be ordered off the streets as a public nuisance and a menace to health.

From the standpoint of silence, our greatest difficulty has been and still is the matter of transmission gears. We employ a four-speed gear and three-speed chain transmission, shown in Figs. 15 and 16 respectively, de-

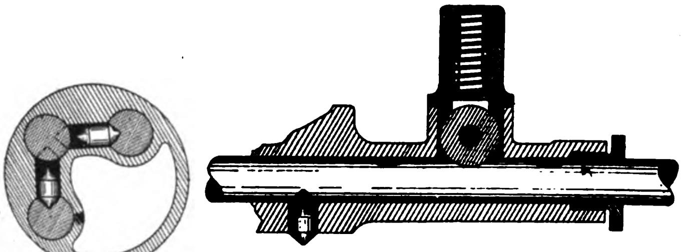


Fig. 18—The Locking Mechanism Used on the Shift Rod of the Four-Speed Gear Transmission

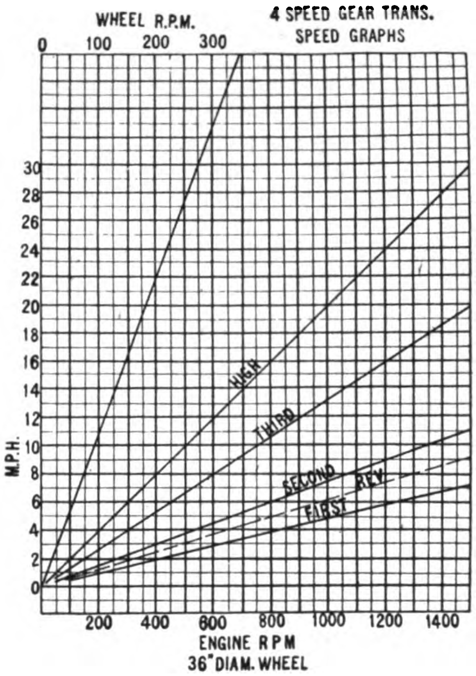


Fig. 19—Speed Curves of the Four-Speed Gear Transmission

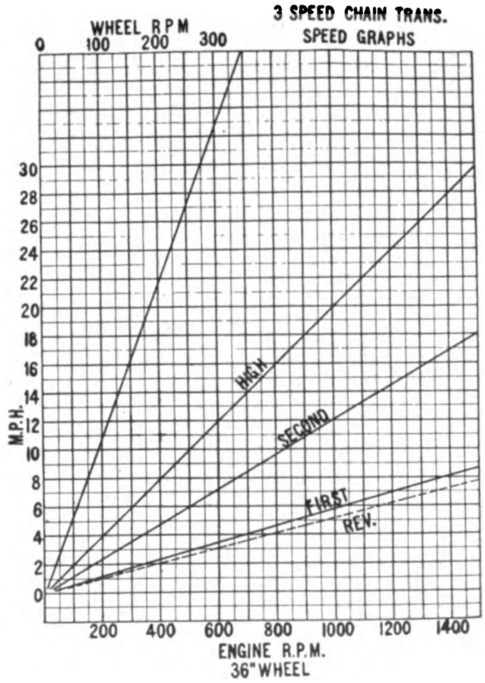


Fig. 20—Speed Curves of the Three-Speed Chain Transmission

pending upon the class of service and general operating conditions. It will be seen from an inspection of Figs. 17 and 18 that the shift-rods, their bearings and the lock mechanism are of substantial proportions.

Illustrations of the speed curves are presented in Figs. 19 and 20. It is worth while noting that the ratios of the four-speed transmission are almost exactly in geometrical progression. The three-speed transmission is not so satisfactory in this respect but here a compromise is of course necessary. This remark applies to all three-speed jobs. Where grades are severe, four speeds are highly desirable, to cut down ability losses to the minimum. But where roads are practically flat, the advantages of a four-speed transmission are not nearly so marked.

The silent-chain transmission is particularly useful for city service where there are frequent stops and starts, and where the percentage of direct-gear operation is relatively small. Substantially it is similar to a constant-mesh gear transmission but chains are used in place of gears. The shift is extremely short and very easy to effect. Such transmissions remain quiet throughout their useful life, and from our observation one can expect at least a year's service from the chains, which are cheaper to replace than gears. Chain transmissions are standard practice for London bus service.

Reliability

The word "reliability" with a bus attains an entirely new meaning. The entire design must be predicated on ability to give uninterrupted service between clearly defined periods, preferably based on mileage. The ability of a bus to fulfill this requirement with particular reference to the duration of period will at once determine the utility of the design. The public will not long tolerate an unreliable service. Failures with an automobile cause confusion enough but the number of persons involved as compared with a bus is relatively insignificant.

One point it is especially desired to bring home is that under average conditions, drivers cannot be expected to make any attempt whatever to spare their equipment. All they are concerned with is stopping for passengers, avoiding accidents, and keeping in their places on the road in accordance with their schedule. Everything must be subordinated to these three things, and in cases where vehicles cannot stand up under such conditions, either the required changes must be made to enable them to do so or they should be scrapped, for assuredly they have no place in the operation of a public utility.

Smoothness of Starting and Stopping

Smoothness of starting is primarily a clutch function, but of course the driver is a factor. Correct gear-ratios,

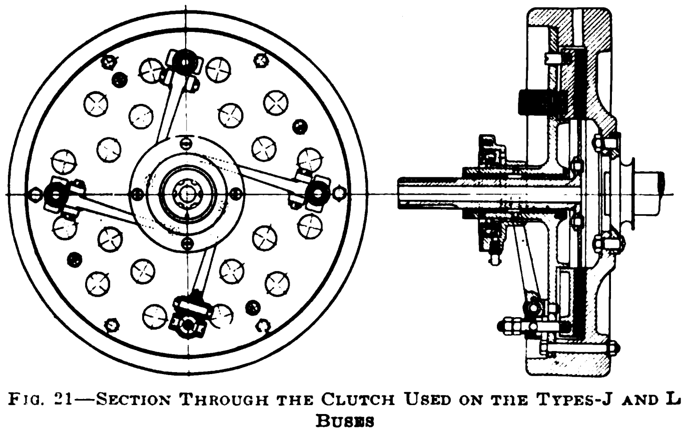


Fig. 21—Section Through the Clutch Used on the Types-J and L Buses

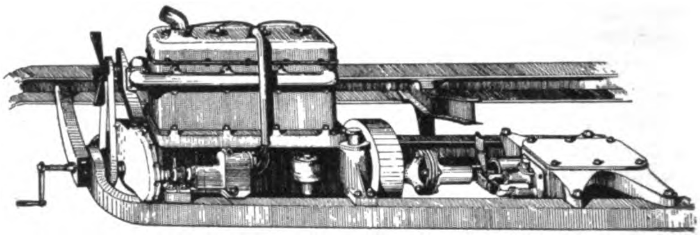


Fig. 22—Sub-Frame Mounting of the Type-J Bus

a satisfactorily performing engine and proper axle-load distribution are contributing influences. Quick starts and stops are highly dangerous from the viewpoint of possible accidents. Some of the heaviest claims for injuries and damages result in this manner. Apart from injuries to passengers, quick starts and stops do more toward causing damage to the chassis and the bodies than anything else. All driving members are subject to abnormal stresses with the former. With the latter, the fore-and-aft or lateral movement, which of necessity results, causes a loosening up of post joints, panelling, etc., and consequently a very high rate of depreciation.

Of the various features that make for efficient and economical operation, the clutch is perhaps one of the most important. We employ exclusively a clutch of the single-disc type. From Fig. 21 it will be seen that there are several unconventional features. Particular attention is drawn to the fact that the spring pressure is evenly distributed over the entire surface of the friction members by 20 small springs, the levers are balanced against centrifugal force and the disc is exceedingly light, thus simplifying the changing of gears. Incidentally, a clutch-stop has been found unnecessary. The removal of the clutch body is an extremely simple operation, as is also the adjustment of the levers. Smoothness of stopping is discussed under the heading of Brakes.

Minimum operating cost demands:

(1) Maximum accessibility
(2) Minimum consumption of labor and material. This of course means excellence of both materials and workmanship
(3) Minimum consumption of fuel
(4) Minimum weight, particularly that which is unsprung
(5) Maximum safe speed. This naturally comprehends rapid acceleration
(6) Maximum tire-mileage

Maximum Accessibility

It is fundamentally necessary that the design of a motorbus be such that inspection and repairs can be carried out quickly and economically. We believe it is imperative that separate unitary construction be followed. For instance, engines, carbureters, all electrical equipment, fans, clutch couplings, transmissions, control levers, axles, wheels and propeller-shafts should all be entities unto themselves, so that the repair of any one of these assemblies will not necessitate the removal of any other.

As a practical illustration, take the orthodox unit powerplant and assume it is necessary to renew the clutch friction linings. The propeller-shaft, transmission and complete control system must first be taken down, possibly even the engine moved forward. In all probability the vehicle must lose a complete day's service. Compare

this for a moment with the relatively simple operation where the separate-unit form of construction is employed, such as with our J or L types. Here we need only remove a few bolts from the clutch coupling and housing. The clutch can then be taken out as a complete unit and the linings replaced within a period of 20 or 30 min. To picture this condition, there is illustrated in Fig. 22 our form of subframe mounting.

The unitary system, if properly carried out, guarantees minimum loss of bus-hours, minimum operating cost, and minimum difficulties from the standpoint of training employes. Obviously, less skill is required on the part of mechanics where they are constantly performing the same operation; here it is simply a question of specialization. But where the construction is such that multi-repair operations are required, the situation is much more complicated. Summing up, to be obliged to remove several units before a faulty unit can be inspected, repaired or replaced, is a condition not to be considered for a moment. Such practice would be ruinous from a public utility standpoint.

It must be remembered that the general conditions surrounding repair work are seldom ideal. There is the matter of wet floors, dirt surrounding the various units, often lack of light. Garage repair forces must work Saturdays and Sundays, which is not particularly attractive. In actual practice it is exceedingly difficult to find men who are willing to work nights. Taken as a whole, the conditions surrounding the work of the repair-men seldom bear favorable comparison with modern high-class factory practice. Here again we wish to emphasize the desirability of unit construction, for the theory is to remove the defective unit and take this to a central repair plant having all the advantages of the modern factory, so that the repairs can be promptly executed by skilled men working under the best possible surroundings.

In connection with the matter of accessibility, it should be remembered that repairs and adjustments must be occasionally carried out at night, sometimes under most unfavorable conditions. Again, assuming the use of low-level equipment, the design should be such that inspections, repairs and renewals can in practically all instances be undertaken from the sides or underneath the vehicles. This means the use of pits. The practice of providing trap-doors inside buses is not desirable. Trap-doors weaken the bodies, are a possible source of accidents, cannot be kept tight in place, permit exhaust gases to leak through, and create undue noise. Experience has shown that it is highly unsatisfactory to carry out chassis repairs from the inside of the body. If this practice is indulged in, claims are bound to result from passengers due to their clothes coming into contact with grease or dirt. Mechanics are sometimes careless and this results in unnecessary damage to the interior fittings, particularly the seat cushions.

Minimum Consumption of Labor and Material

From a financial viewpoint, the success or failure of a utility operating buses depends upon the cumulative additions or subtractions of small amounts expended on either labor or material. Sometimes the items may appear insignificant but, taken as a whole and over lengthy periods, the story is entirely different. When working, a bus is a heavy consumer of both labor and material. The consumption is perhaps much greater than is generally supposed. To afford a practical illustration, Table 5 representing the actual consumption by our company of some of the major elements for the year 1921,

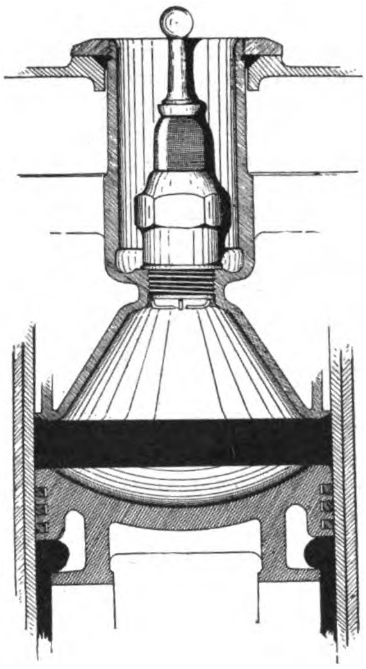


Fig. 23—Sectional Elevation Through the Combustion Chamber of the Engine Used on the Types-J and L Buses

may be of interest. These figures are based on the average of all buses.

TABLE 5—FIFTH AVENUE COACH CO.'S COST PER BUS FOR 1921

Gasoline			$1,125.94
Lubrication			109.42
Tires			284.34
Repairs to Chassis	Labor	$676.97	1,436.78
	Material	759.81	
Repairs to Bodies	Labor	359.00	521.44
	Material	162.44	
Drivers			3,071.71
Conductors			2,692.48
Total			$9,242.11

From a casual study of these data it will be seen that a relatively small percentage of saving, if applied to any of the items and then multiplied by a large number of

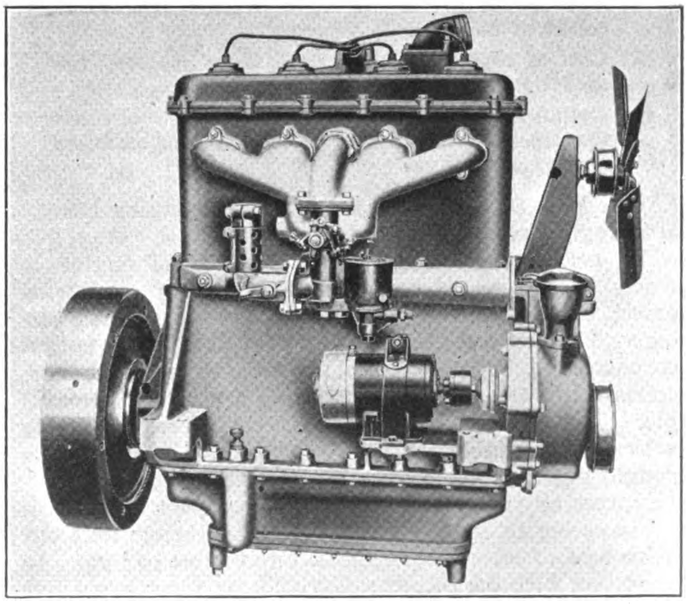


Fig. 24—The Engine Used on the Types-J and L Buses

vehicles, must total a vast sum annually. If one assumes that the equipment in question is of good design and that its maintenance is economically undertaken, then how much more important does this issue become when the reverse is true.

Perhaps it will not be out of place here to point out that the profit of the average utility expressed percentagewise, usually does not run beyond one figure, and that there are a vast number of utilities where the figure is in red. To change the color and to exceed the single-figure basis, requires all that is best in design, material, workmanship and operating care.

Minimum Consumption of Fuel

Aside from the human elements which have been covered in a previous paper, Motor-Bus Transportation,[3] presented at the 1920 Semi-Annual Meeting, the major issue, of course, is the engine. We employ exclusively the sleeve-valve type. From our viewpoint this type possesses certain basic advantages which make for economy of operation.

First, taking the question of fuel, high gasoline-economy is possible due to

(1) Absence of valve pockets and the spherically shaped combustion-chamber. Incidentally, this permits of high compression being employed. The illustration of the combustion-chamber (Fig. 23) brings out this point very clearly
(2) Positive action of valves at all speeds
(3) Extraordinarily low friction-horsepower
(4) Ideal location of the spark-plug

Next, there is the question of service. In this respect we believe the sleeve-valve engine has the following advantages:

(1) The performance remains reasonably constant throughout the useful life. It is not necessary to make adjustments constantly to permit of satisfactory and uniform behavior
(2) Throughout the useful life the performance tends to improve
(3) Practically no adjustments can be made since there is nothing to adjust. This alone represents a considerable saving in the garage force
(4) Throughout useful life there is little, if any, increase of noise due to wear
(5) Cost of repairs is small since there are very few operations requiring skill
(6) Cylinders never require reboring. This obviates the necessity of carrying in stock second-standard pistons and rings

From Fig. 24 it will be seen that the engine has an exceedingly clean appearance.

The performance of a correctly designed engine is largely a function of its carbureter; therefore a wide variety of results is always obtainable with varied settings. From the graph showing fuel and power output reproduced in Fig. 25 it will be noticed that the characteristics of the sleeve-valve engine are rather remarkable. The setting in question is considered as being particularly suitable for type-J equipment. The points brought out in Table 6 are of special interest.

Expressing the results obtained in another manner, it is interesting to reflect on the fact that during 1921 our entire fleet of buses averaged 50.7 ton-miles per gal. In connection with the rather remarkable performance which

[3] See Transactions, vol. 15, part 2, p. 143.

TABLE 6—HORSEPOWER AND TORQUE DATA FOR TYPE-J BUS

Power Developed at 1,000 R.P.M., hp.	36.20
Power Developed per Cubic Inch of Displacement, hp.	0.12
Weight of Vehicle per Horsepower, lb.	301.00
Weight of Vehicle per Cubic Inch of Displacement, lb.	36.20
Maximum Torque, lb-ft.	194.00
Speed for Maximum Torque, r.p.m.	800.00
Decrease in Torque at 400 R.P.M., per cent	5.10
Decrease in Torque at 1,400 R.P.M., per cent	11.90
Speed for Maximum Torque with a 5.4 to 1 Rear-Axle Ratio, m.p.h.	16.10
Minimum Fuel-Consumption, lb. per b. hp-hr.	0.55

this type of engine delivers in our service, particularly from the standpoint of fuel economy, mention should be made of the carbureter, which is of the Zenith type. From Fig. 26 it will be seen that there is no exterior adjustment. The throttle spindle is 7/16 in. in diameter, hardened and ground. There is a total of 4 in. spindle bearing-area. There is a gland with a suitable packing at the front end and a blank nut at the other. It is interesting to compare the arrangement with conventional designs that in many instances have throttle spindles resembling closely wire nails. With the bus there is an abnormal amount of throttle movement, and unless this factor is taken into consideration from the standpoint of design, rapid spindle and bearing wear will take place. It will also be seen that the design is rugged throughout. All screws, nuts, plugs or unions are of ample size. The butterfly is exceedingly well fitted and provision is made for a simple throttle-stop adjustment. These points are clearly brought out by Fig. 26.

Minimum Weight

It seems scarcely necessary here to argue as to the desirability of light weight. These remarks particularly apply to the matter of unsprung weight. Assuming good design, obviously minimum weight means minimum fuel-consumption, maximum acceleration and speed and minimum costs for repairs and renewals. These are the controlling elements. Henry Ford started out with this idea firmly imbedded in his mind and, as far as we know, he has had no cause to change his views.

Clearly, the lighter the vehicle, the easier the solution of our problems. Heavy vehicle-weight means unnecessarily large tires, stronger axles and frame, larger brakes, slower gear-ratios and, last but not least, more engine power. The entire theory of design should be based on the highest safe vehicle-speed for the smallest throttle-opening, and consequently the minimum number of engine revolutions. Of course, this is out of the question if we start off with an unnecessarily heavy unit.

From our experience in operating 21 different types of buses in the past 14 years, we believe that the weights and percentages of axle-load distribution given in Table 7 make for safe and efficient practice.

Maximum Safe Speed

The greatest single factor from the standpoint of economical operation is speed. This point is perhaps not sufficiently recognized. The following facts in connection with our operation may make the matter somewhat clearer. During 1921 we spent in platform payment, drivers' and conductors' wages, in round figures, $1,625,-000. So, for each 1-per cent economy in speed there is a yearly potential saving of more than $16,000. Looking at the situation another way, the ratio of expenditure

between our platform payment and all money expended in connection with repairs and renewals to chassis and bodies, is approximately 5 to 1.

From this it is clear that, while there are always opportunities to effect a saving in connection with maintenance methods generally, the real solution is to employ the fastest possible safe speed and to drive the vehicles up to the limit of their endurance. This, of course, necessitates all that is best from the standpoint of design. Naturally, to maintain a high average rate of speed, rapid acceleration is essential. But in connection with this matter it is well to bear in mind that there is nothing gained and much lost if the engine power is in excess of actual requirements, for it is bound to be abused. A very real problem is to ascertain with each operation the exact amount of power required, then to adopt a standard carbureter-setting with a view to its proper control. Obviously, the questions of acceleration, deceleration and maximum safe speed are closely allied. Reference has been made to deceleration under the heading Effective Brakes.

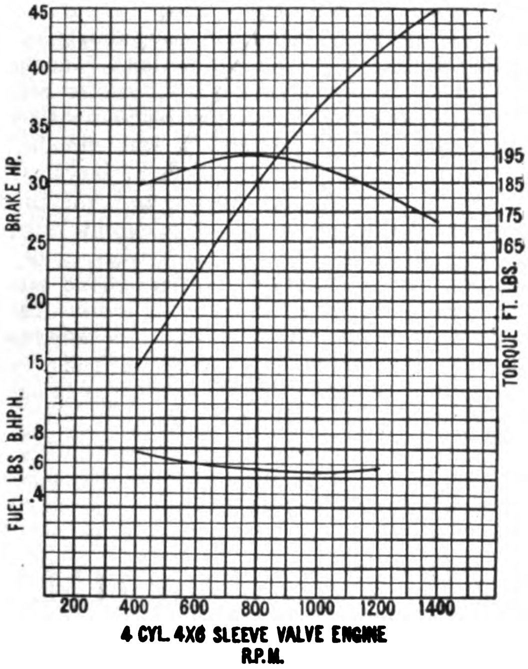


FIG. 25—CURVES SHOWING ENGINE PERFORMANCE

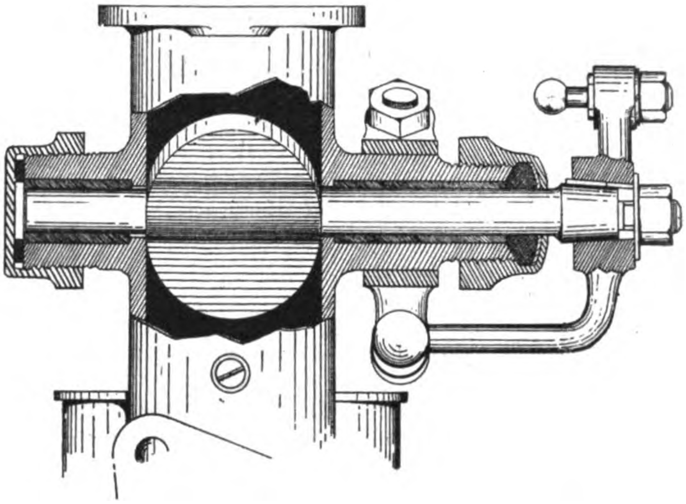


FIG. 26—SECTIONAL VIEW THROUGH THE CARBURETER THROTTLE OF THE TYPES-J AND L ENGINE

MAXIMUM TIRE-MILEAGE

In the earlier days of bus operation, the tire question was one of our chief anxieties. Today the situation is very different, for wonderful improvements have been made in tire manufacturing methods. Of course, there is no sense in decreasing tire expenditures at the cost of the equipment generally. Resilient tires are essential and

TABLE 7—DISTRIBUTION OF WEIGHT IN 51-PASSENGER DOUBLE-DECK AND 25-PASSENGER SINGLE-DECK BUSES

Description	Double-Deck Bus					Single-Deck Bus				
	Total, Lb.	Front		Rear		Total, Lb.	Front		Rear	
		Lb.	Per Cent	Lb.	Per Cent		Lb.	Per Cent	Lb.	Per Cent
Chassis, Gasoline, Oil, Water and Foot-Boards	6,000	3,000	50	3,000	50	5,000	2,600	52	2,400	48
Body, Sign, Battery and Heaters	4,000	880	22	3,120	78	2,000	500	25	1,500	75
Chassis and Body	10,000	3,880	39	6,120	61	7,000	3,100	44	3,900	56
Passengers and Crew at 150 Lb. Each	7,950	2,385	30	5,565	70	3,900	430	11	3,470	89
Grand Total of Fully Loaded Bus	17,950	6,265	35	11,685	65	10,900	3,530	33	7,370	67
Load on Each Tire	....	3,133[a]	....	5,842[b]	....	....	1,765[a]	....	3,685[c]	....
Unsprung Weight, Springs to Tires	....	700	....	2,000	....	....	650	....	1,550	....
Chassis Load on Each Spring	....	1,150	....	500	....	....	975	....	425	....
Body Load on Each Spring	....	440	....	1,560	....	....	250	....	750	....
Passenger and Crew Load on Each Spring	....	1,193	....	2,782	....	....	215	....	1,735	....
Total Load on Each Spring	....	2,783	....	4,842	....	....	1,440	....	2,910	....

[a] Size 34 x 4 in. single.
[b] Size 34 x 5 in. dual. Figure is for each set.
[c] Size 34 x 6 in. single.

too great a wear must not be permitted. It is our regular practice to remove a tire immediately the rubber has worn to within 7/8 in. of the hard base.

In looking back over our records, it is extremely interesting to note that in 1911 our cost per mile for tires was 4.93 cents. From that date on, a steady reduction has been effected. The figure for 1921 was 0.87 cents per mile, and this, of course, includes the use of six tires. From our viewpoint the factors which have permitted this condition to be reached are, in the order of their importance

(1) Better tire manufacturing methods
(2) Improved vehicle design. This includes decreased weight, particularly unsprung weight, the substitution of metal for wood wheels, etc.
(3) Closer supervision from an operating standpoint
(4) Closer supervision from a maintenance standpoint

Conclusion

As the result of long experience in connection with the design, construction and operation of buses, we are convinced more than ever that trucks or automobiles, modified or unmodified, are absolutely incapable of giving satisfactory and economical service if operated as buses. The tendency today is to employ trucks or automobile chassis as buses, or to attempt to modify their construction, then to re-christen them. This is a dangerous policy from the standpoint of both the builder and the user, and eventually it must surely result in dissatisfaction and disillusionment.

There is another and very important matter: We must not lose sight of the fact that the bus has not made good in some of the localities where it has been tried out. We are constantly confronted with failures such as those at Des Moines, Toledo, Kansas City, and other cities. Such failures, when analyzed, invariably point to the fact that the combination of extemporized equipment, indiscriminate operation, overloading and lack of experience is responsible. But these failures can be avoided, and the automotive industry in its own interest should do all that is possible to guard against such occurrences.

It seems scarcely necessary here to comment upon the splendid achievements of the Society in connection with standardization work in general. Certainly, this has been a controlling influence in the development of the automotive industry. We believe much would be gained if it should now concentrate upon the motorbus. What we have in mind is the standardization of certain of the main dimensions; for example, front and rear-axle tracks, spring center-to-center distances, frame width, dimension between dash and wheel pocket, seat dimensions, aisle widths, etc., for the various classes of service.

The main object of this paper is to bring to the attention of interested parties in a clearcut, vigorous and interesting manner, the fact that to produce motorbus chassis that can be operated efficiently and economically, a very close study must be made of the entire situation. It is also desired to destroy as far as possible the illusion that a bus chassis is merely a modified truck. If in these things, even a moderate degree of success is achieved, we shall feel amply repaid for our efforts.

The matter of body design has not been touched upon since this is a subject that, because of its magnitude, must receive separate treatment.

PRESIDENTIAL ADDRESS OF B. B. BACHMAN

(Concluded from page 12)

creasing traffic, particularly over main routes, will bring a reaction unless we are peculiarly alert to study and suppress in design all objectionable characteristics of our vehicles to the greatest possible degree. I appreciate that the control of all these features is not in the hands of the engineer or builder, but he should be thoroughly posted as to what they are and be prepared to cooperate intelligently with regulatory bodies to assure that rational measures for the protection of the public, which do not impose unreasonable restriction on road transportation, are enforced.

Originally road construction was in the hands of individuals or corporations that operated them for profit in the collection of tolls. While we have rejected, as a Nation, the idea of public ownership of the railroads, so also have we rejected the idea of private ownership of the highways. I believe both these ideas are proper. In the railroad we require concentration of authority and responsibility in operation over any one given line. This can be obtained most efficiently by private ownership and operation under reasonable government regulation. The highway, on the other hand, is primarily for the use of the individual according to his needs and desires, with as little restriction as possible consistent with public safety, which can be obtained best by public ownership and complete government control through one of its departments.

Much of the discussion on the question as to who should bear the burden of the cost of construction and maintenance of our highway systems, or whether the motor-vehicle operator is receiving a public subsidy that is not shared by the railroad, etc., appears to be beside the point. The cost of transportation of passengers and freight, by railroad, water or highway, is borne by the whole community and shared by every citizen in proportion to his requirements for transportation. I believe this to be so, whether the cost of transportation is included in the cost of the commodity or it appears partially in the form of taxes. The big fundamental problem is to determine the economic field for each medium of transportation and the relation each should bear to the other for maximum efficiency, and the most satisfactory means of proportioning the expense to the individual.

I have endeavored to the best of my ability to give you a brief and yet comprehensive view of the problems that are confronting us and should receive our active individual and collective attention. I hope the result may be to stimulate interest in the affairs of the Society and enlargement of the horizon in our view of the future activity of, and the service that can be rendered by, each of us individually and as an organization.

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The Hot-Spot Method of Heavy-Fuel Preparation

By F. C. Mock[1] and M. E. Chandler[2]

Semi-Annual Meeting Paper *Illustrated with* Drawings

THE development of intake-manifolds in the past has been confined mainly to modifications of constructional details. Believing that the increased use of automotive equipment will lead to a demand for fuel that will result in the higher cost and lower quality of the fuel, and being convinced that the sole requirement of satisfactory operation with kerosene and mixtures of the heavier oils with alcohol and benzol is the proper preparation of the fuel in the manifold, the authors have investigated the various methods of heat application in the endeavor to produce the minimum temperature necessary for a dry mixture.

Finding that this minimum temperature varied with the method of application of the heat, an analysis was made of the available methods on a functional rather than a structural basis. Three of these are discussed: (*a*) When the heat from the walls of the manifold is applied through the medium of the air; (*b*) when it is applied to the fuel alone, or partly to the fuel and partly to the air; and (*c*) when a spray of atomized fuel and air is directed against a heated surface. A device was constructed by which the three main variables, the exhaust temperature, the exhaust flow and the area of the heating surface, might be regulated and the three remaining variables, the quantity of air, the quantity of fuel supplied and the quantity of fuel vaporized, might be controlled.

Taking into account the wide range of temperatures that the air charge and fuel supply undergo before entering the intake-manifold system, a quantitative computation of heat transfer was made and the conclusions were drawn that only by a combination of centrifugal force, surface tension and the force of gravity could the unvaporized drops be separated from the fuel charge and that the conditions of combustion are governed by the rate of fuel feed from the manifold to the cylinder and not from the carbureter to the manifold.

RECENT years have witnessed increasing attention to the design of intake-manifolds and to the varied methods of handling automotive fuels in preparing them for introduction into the combustion-chamber. The resulting development, however, has been limited in direction, being confined usually to slight modifications of the construction that has been followed ever since automotive engines began to have more than one cylinder. The improvement in economic conditions that all authorities agree is approaching will certainly result in considerably increased use of automotive equipment, and it is not impossible that the demand for motor fuel may bring about a condition of higher cost and lower quality. As it might require three or four years to develop a change of design to meet such a change of fuel, it would seem that now is a fitting time to make a survey of the problems involved in the preparation of our present fuels and of heavier ones and to make a fresh analysis of the situation, entirely apart from and unhampered by the conditions of previous practice.

It is true that many 1922-model cars have operated satisfactorily with the motor fuels at present in use, both in summer and in winter, but many have not. We are convinced that it is possible to operate on mixtures of gasoline, kerosene and some heavier oils, combined with alcohol, benzol or other anti-knock component, as well or better than a number of cars today operate on gasoline, by the use of improved methods of fuel preparation in the intake-manifold.

Specific Requirements of Fuel Preparation

The requirements of proper fuel preparation are

(1) A thoroughly and continuously homogeneous mixture of fuel and air with no drops or liquid-film wall-flow to the valve ports

(2) The charge temperature should be the minimum possible while complying with requirement (1)

(3) The provision for a prompt change in the rate of action under changes of load and speed

A cylinder charge of fuel is only a medium-sized drop. Any one who has observed through glass manifold sections, the storm of drops that is usually present, can easily appreciate the importance of this point. All the oil dilution in the crankcase is due, of course, to the introduction into the combustion-chamber of fuel that is not burned later. We believe that a large part of the rapid carbon formation, characteristic of engines having poor distribution, is due to the cracking, without burning, of the drops of excess fuel that occasionally enter the cylinders. The first requirement would include the prevention of liquid gasoline from reaching the cylinders after the use of a primer or choke means of starting. As starting is really an increase of the load from zero, devices for starting and quickly warming-up come under the last requirement.

Methods of Heat Application

If we consider as our objective a minimum temperature of the dry mixture, that is, a mixture of transparent fuel-vapor and air, it is immaterial, in theory, whether the heat is applied first to the fuel or to the air. If, however, we accept what our experiments have apparently demonstrated, that is, that a fog mixture of condensed vapor and air is satisfactory, provided the cylinder temperatures are such as to change this fog to a vapor before the end of the compression stroke, we shall find, both in theory and in practice, that the minimum temperature that can be used will vary with different methods of heat application. The theoretical considerations involved are, we hope, clearly shown by an analysis of the known and available methods of heat application. These have been classified as follows:

Case No. 1. Heat imparted to the mixture through the medium of the air, by the communication of

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heat from the manifold walls to the air and to such part of the fuel as has been deposited on the manifold walls. This is considered to involve only the production of a dry-vapor mixture. An interesting variation of this is shown as Case No. 1b, where part of the preheating of the air is accomplished by subtracting heat from the air and the vapor mixture already formed, thus giving a fog mixture

Case No. 2. Application of heat first to the fuel alone, with resulting condensation of the vapor when it joins the main air-column; this results in a fog mixture

Case No. 2a. Heating the fuel and a part of the air to generate a rich dry-vapor mixture, which is then condensed as it enters the stream of the remaining air-supply. This gives a fog mixture

Case No. 3. Directing a spray of atomized fuel and air against a heated surface. One result obtained with this construction is the breaking-up of the spray drops into even smaller drops in the so-called "spheroidal" condition; the mixture thus formed can scarcely be properly designated as a fog mixture

This classification has been made on a functional rather than a structural basis. Most of the hot-spot constructions in actual use employ two, and sometimes three, of these heating methods, but for analysis the distinction we have made seemed necessary. Consideration of the direct application of heat to the fuel has been purposely limited to designs in which the fuel has been previously metered in a liquid state, as doing so after heating has not thus far been demonstrated as practicable.

The computation of the mixture-temperatures is based upon the methods used and determinations made by Professor R. E. Wilson and described in THE JOURNAL.[3] The gasoline values used are those of the high end-point gasoline referred to in Professor Wilson's discussion in THE JOURNAL[4] for October 1921. This gasoline, by the way, is apparently quite similar to the "D" gasoline of the fuel research consumption test recently concluded.

[3] See THE JOURNAL, November 1921, p. 313, and January 1922, p. 65.
[4] See THE JOURNAL, October 1921, p. 265.

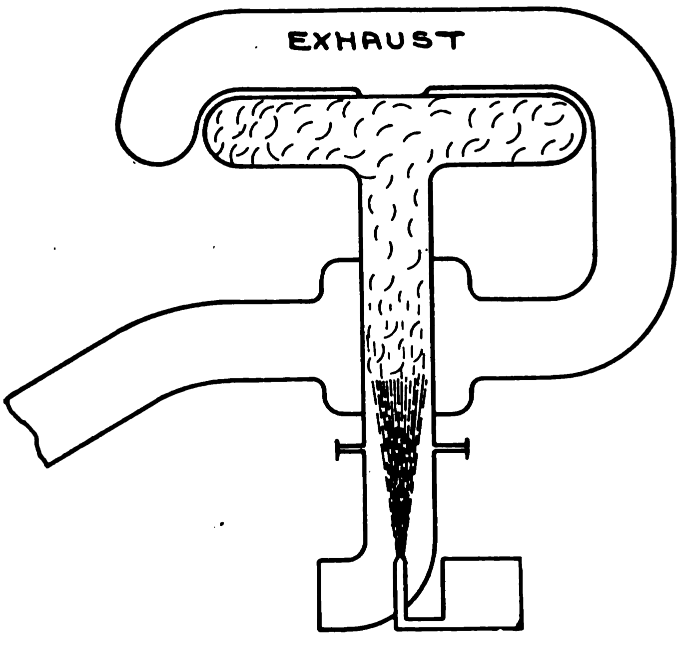

FIG. 1—IN THIS CONSTRUCTION THE HEAT IS APPLIED DIRECTLY TO THE AIR PORTION OF THE CYLINDER CHARGE

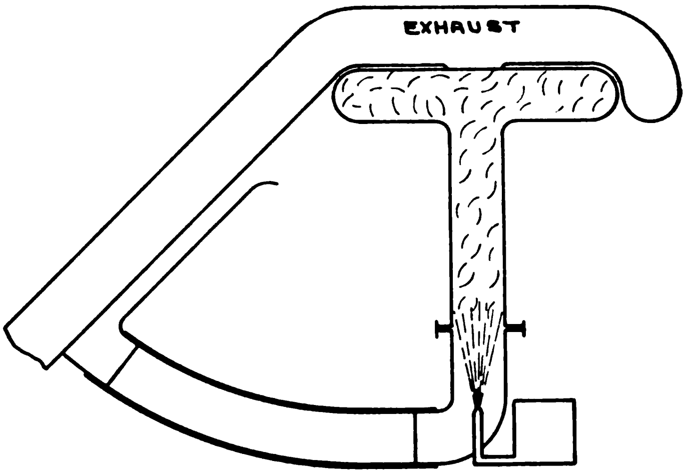

FIG. 2—MANIFOLD CONSTRUCTION TO SUPPLY HEAT TO THE AIR CHARGE BEFORE IT ENTERS THE CARBURETER

HEAT APPLIED DIRECTLY TO THE AIR

In Case No. 1 the air charge receives a heat supply such that after the latent heat of evaporation has been supplied to the fuel, the resulting mixture will have the minimum temperature of a dry vapor. A typical construction is shown in Fig. 1. Its practical equivalent, the application of heat to the air charge before it enters the carbureter, is shown in Fig. 2. With the complete evaporation of a 15-to-1 mixture of the high end-point gasoline measured by Professor Wilson, and ignoring the heating of a small amount of fuel on the walls, this would involve air entering the carbureter at 181 deg. fahr. with a resulting mixture-temperature of 135 deg. fahr. For the kerosene measured by Professor Wilson, the air would have to enter the carbureter at 283 deg. fahr. with a final mixture-temperature of 230 deg. fahr.

In practice, however, dry mixtures are not realized at such low temperatures, for the reason that only part of the hot air comes into contact with the fuel. Within a short distance from the carbureter jet the tiny droplets of fuel spray take up a velocity and direction identical with that of the air which bears them and thenceforth, until they strike a wall, they generally are surrounded by a miniature atmosphere of vapor at the dew-point. Fuel that travels along the walls comes into actual contact with only a thin film of air. We have endeavored by various means to create a turbulence that would accelerate and decelerate the spray droplets in the air medium that carries them, but every effort of this kind has resulted in increased deposition of fuel on the manifold wall and has made conditions worse than before. The temperatures actually existing in practice are more nearly those that would result if the fuel came into heat-conducting contact with but one-half to one-third the air charge during the travel through the intake-manifold. On such a basis the average temperature of the mixture is considerably higher, for instance, with a 15-to-1 dry mixture, and, if the fuel receives heat from one-half the air, the final average temperature will approximate 175 deg. fahr. with gasoline and 276 deg. fahr. with kerosene, as is brought out in Table 1.

On account of the high heat-capacity of dry-mixture charges formed in this way, there being no cooling from any further evaporation of the fuel during the compression-stroke, the tendency toward detonation should be, and apparently is, greater with this method of fuel preparation than with most others. Due to the relatively slow heat-transfer, more than the customary difficulty is expe-

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rienced during changes of engine speed and load. The proper functioning of a device of this kind is contingent upon the maintenance of adequate temperatures; but in actual practice such temperature regulation is disturbed by a number of factors, depending upon seasonal and climatic conditions, as will be explained later. Since the mixture-temperature depends upon that of the air entering the carbureter, which in most cars depends in turn upon the temperature of the cooling water and of the whole mass of metal under the hood, there is a long duration of "warming-up" which can be taken care of only by elaborate thermostatic devices. A factor of safety, to provide for the occasional use of fuels heavier than the average, can be obtained only by raising still farther the temperature of the fuel charge of normal operation.

TABLE 1 — FINAL AVERAGE MIXTURE-TEMPERATURE WITH ENTERING AIR AND FUEL AT 75 DEG. FAHR.

Fuel	High End-Point Gasoline		Kerosene	
Air-Fuel Ratio	12 to 1	15 to 1	12 to 1	15 to 1
When Heat Is Transferred from All the Air to the Fuel (Fig. 2)	145	135	240	230
When Heat Is Transferred from One-Half the Air to the Fuel (Fig. 2)	195	175	299	276
When Heat Is Transferred from One-Half the Air to the Fuel (Fig. 3) Followed by Cooling of the Charge by the Intake Air	140	128	196	176
When Heat Is Applied Directly to the Fuel (Fig. 4)	143	132	174	159
When the Fuel Only Is Heated to Temperature of Required Vapor Density (Fig. 6)	98	92	124	116

More important is the fact that there is nothing to prevent raw gasoline entering the cylinders during the starting and warming-up period and probably also during normal running.

In Case No. 1b, Fig. 3, the fuel vapor is formed as in Case No. 1, but a smaller exhaust air-heater is used. The air entering the intake system, before it reaches the exhaust heater, is used to cool and condense to a fog the dry mixture coming from the carbureter. The temperatures of the air entering the carbureter and of the mixture leaving the carbureter are the same as in Case No. 1, but the final mixture-temperature in the intake-manifold, if a complete heat-transfer could be established, would be considerably lower than in Case No. 1; for instance, 128 deg. fahr. with gasoline as against 175, and 176 deg. fahr. with kerosene as against 276. But we do not believe that in practice the addition of this condensing device would be of value. If made elaborately enough to accomplish the desired heat-transfer, it would probably increase the amount of fuel on the walls and require a still higher temperature of the air entering the carbureter. It would also increase the difficulties of acceleration and the "loading" in the intake-manifold while the engine is cold.

Heat Applied Directly to the Fuel

In this method, which is shown in Fig. 4, the fuel, after being metered is discharged into a heating cham-

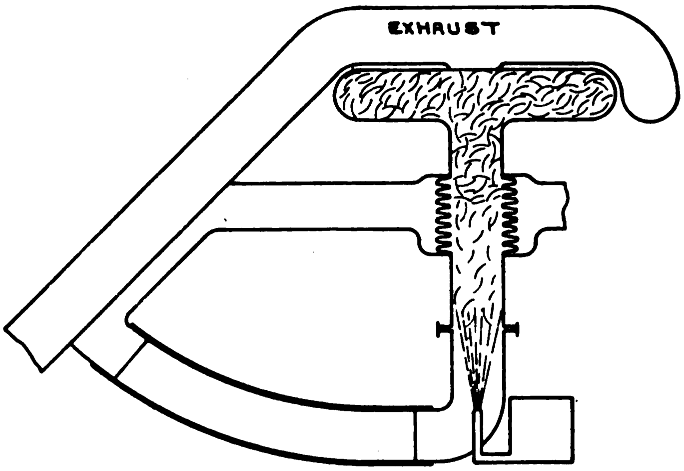

Fig. 3—In This Construction Part of the Initial Heat Application Is Obtained by Cooling the Vapor Mixture to a Fog

ber that the air charge does not enter; the vapor formed here is then mixed with the unheated air-charge to form a true fog-mixture. At first thought this system seems to be promising, but actually it has serious inherent disadvantages, for the reason that the delivery of vapor depends upon the temperature being kept above a certain minimum.

A homely illustration of the difficulty of evaporation with this type of heater is afforded by the example of a covered kettle or pot of water maintained at a temperature slightly below the boiling point, say 208 deg. fahr. As any housewife knows, a kettle can be heated in this manner for a long time without losing much water, the reason being that, although the evaporation from the surface of the water is rapid for a while, until the space above the water and beneath the lid becomes filled with vapor, there is no difference in pressure between the vapor and the outside air and no marked escape of water vapor from the spout. It is only when the temperature is

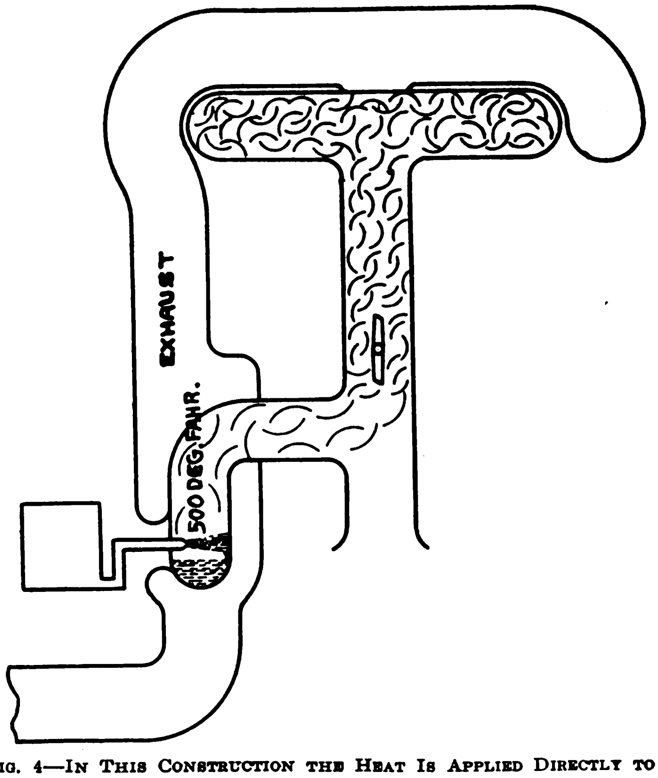

Fig. 4—In This Construction the Heat Is Applied Directly to the Fuel Portion of the Mixture Charge

raised to the boiling-point that the vapor pressure is able to rise above that of the atmosphere and create a continuous outflow of steam. An open chamber will evaporate liquid below the boiling-point much more quickly than a closed one, the difference being due solely to the more rapid escape of the vapor from the open chamber. In the design illustrated a normal flow of vapor from the heating chamber should take place only when the vapor temperature is raised to the final boiling-point of the fuel; that is, the vapor must be between 400 and 500 deg. fahr., which is much higher than the temperature needed with any other construction shown. The final temperature of the mixture may, however, be quite low because of the fact that very little more heat need be added to the system than is necessary to vaporize the fuel. Also, the

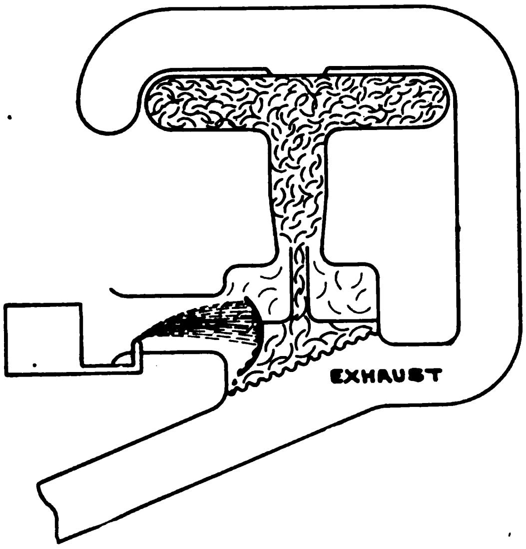

FIG. 5—IN THIS MANIFOLD ARRANGEMENT THE HEAT IS APPLIED DIRECTLY TO THE MAIN PORTION OF THE FUEL AND A SMALL PORTION OF THE AIR

"factor of safety" in heating capacity may be large without raising the final temperature of the mixture in proportion.

This arrangement might be hard to start and would possibly be slow on acceleration. With heavy fuels there would be a tendency for the heavy elements to collect in the bottom of the heating chamber during idling, when the exhaust temperature is lower than the boiling-point of the fuel. Upon a sudden increase of the exhaust temperature this pool of heavy elements is apt to coke. In fact, we have known of a number of instances where a pocket for the collection and heating of the fuel would fill with greasy tar or coke. This trouble was particularly marked with high end-point gasolines.

This construction has the additional advantage, when properly designed, of permitting no liquid fuel to reach the valve ports; on this account, as also with Case No. 2a, it will give a homogeneous fuel charge, or "good distribution," as we call it, with any shape of intake-manifold and any convenient location of the carbureter.

Case No. 2a, Fig. 5, is a sort of compromise between Cases Nos. 1, 2 and 3, which seems to possess all the advantages of Case No. 3 and fewer disadvantages. The mixture spray from the carbureter is thrown against a deflecting surface, that may be heated, and the fuel not vaporized is thrown down into a heating chamber as in Case No. 2. An opportunity is afforded for the fuel to evaporate in and mingle with the air, before the separation of the liquid and the vaporized portion. This reduces the fuel lag on acceleration and also reduces the amount of fuel that must be taken into the heating chamber. An air circulation is maintained through the heating chamber, which helps to carry the vapor away as fast as it is formed; the action in the heating chamber then can be *evaporation* rather than *boiling* as in Case No. 2. This distinction is important because *boiling* implies the maintenance of temperature above a certain point, at all engine speeds and at a constant pressure, while evaporation can take place at any temperature and, fortunately, under a change of engine speed, the decrease of the exhaust temperature is accompanied by a reduction of the fuel feed and the rate of evaporation required.

The air taken through the heating chamber is, of course, highly heated, so that, as compared with Case No. 2, we have a small part of the fuel and of the air heated, to be cooled by the remainder of the air charge and a certain part of the fuel charge. The temperature balance would, of course, depend on the percentages of the fuel initially vaporized and of the air passed through the heating chamber.

This arrangement possesses the advantage of Case No. 2, in allowing a large reserve capacity for warming-up without excessive heating of the mixture under normal operation, and also of preventing liquid fuel from going into the engine cylinders. A device of this sort, though of design entirely different from Fig. 5, has been used in the actual driving of a passenger car with a six-cylinder engine and gave as good a demonstration on kerosene, with a benzol component to avoid detonation, also alcohol at a mixture-temperature of 120 to 140 deg. fahr., as with gasoline. It was also found possible to use heavier fuel combinations which resulted in perhaps better operation than that shown by the average car in the hands of its owner: One of these mixtures was one-third benzol, one-half kerosene and one-sixth Mobile B lubricating oil; another one-fifth alcohol and four-fifths 38 to 40-deg. Baumé distillate, a light oil that cannot be ignited by itself with a match in the atmosphere at ordinary temperatures and which will burn slowly from a wick with a very smoky flame. With these latter mixtures the mileage per pound was not as good as with gasoline or kerosene, and there was a perceptible carbon-deposit; also a slight slowness, but not hesitation, on acceleration. The operation of the car in general was so good that it would easily satisfy the average car-owner, were it not for the necessity of starting on a different and lighter fuel. Starting on gasoline in very cold weather was not more difficult than with the ordinary carbureter and intake-manifold arrangement. In fact, no difficulty was ever experienced in starting; the starter was always strong enough to turn the engine over, and closing the choke would always effect a start. On gasoline the warming-up was very good. In weather 10 deg. fahr. above zero, it was necessary only to use the dash mixture-control device for about ½ min. or less after starting, after which it was possible to set all the controls in the normal driving position and drive away. This usually synchronized with the development of a mixture-temperature of about 90 deg. fahr. With gasoline the fuel-consumption was but slightly lower on a gallon test than with a good carbureter on a conventional type of hot-spot intake-manifold, but the engine would run smoothly on very lean mixtures and the weekly mileage, particularly in winter, was better. The smoothness and the absence of carbon, crankcase-oil dilution and ignition trouble were marked. We found also improved operation at low speed on hills. The engine would pull smoothly and without apparent

effort and maintain this smooth low-speed pulling indefinitely.

Air and Fuel Charge Projected Against the Hot Surface

As illustrated in Fig. 6, this includes a condition aimed at, and more or less realized, in many hot-spots in use today. It is the general belief, perhaps, that the fuel spray strikes the heated surface, vaporizes, and then condenses in the airstream. More recent observations lead us to believe that very little of the fuel vaporizes on the heated surface. It seems rather that the sudden application of heat to one side of the drops of spray, as they strike the heated surface, relieves the surface tension that holds them in globular form and causes them to burst; meanwhile, if an air-draft is present, the "spheroidal condition" keeps them from adhering to the heated surface. This belief was first suggested by the observation that large drops come off such a hot-spot in a coarse spray, while small drops come off in a finer spray.

There is one interesting hypothesis of action under these conditions, the realization of which would give a fog mixture at very low temperatures with a very simple structure. If the heating surface were of exactly the size and location to be wholly covered by the liquid of the fuel spray; if its heating capacity were such that it could vaporize all the fuel that strikes it; and if the scouring action of the air-draft across the heating surface were sufficient to carry away the vapor as fast as it was formed, it would be possible to produce the vapor at the relatively low temperature corresponding to a density of one-fifteenth to one-twelfth that of air; also, there should be little, if any, heat transmitted directly to the air from the heating surface. Under such conditions, which we believe can be realized only in theory, the mixture temperatures would be the minimum among all the systems suggested for producing a fog mixture by external application of heat energy.

Fig. 7 is an effort to show the nature of such action, assuming complete evaporation at the surface. There is, first, near the surface, a film of liquid, or a layer of liquid

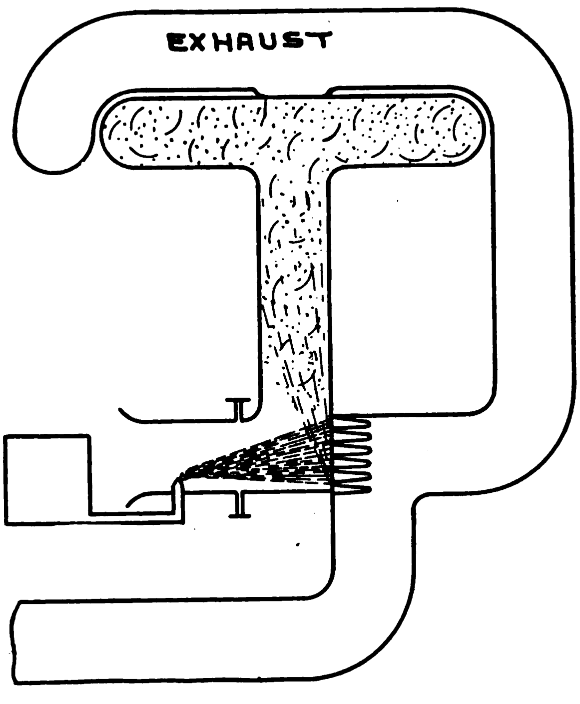

Fig. 6—In This Manifold the Air and Fuel Charge Is Projected Against a Heated Surface

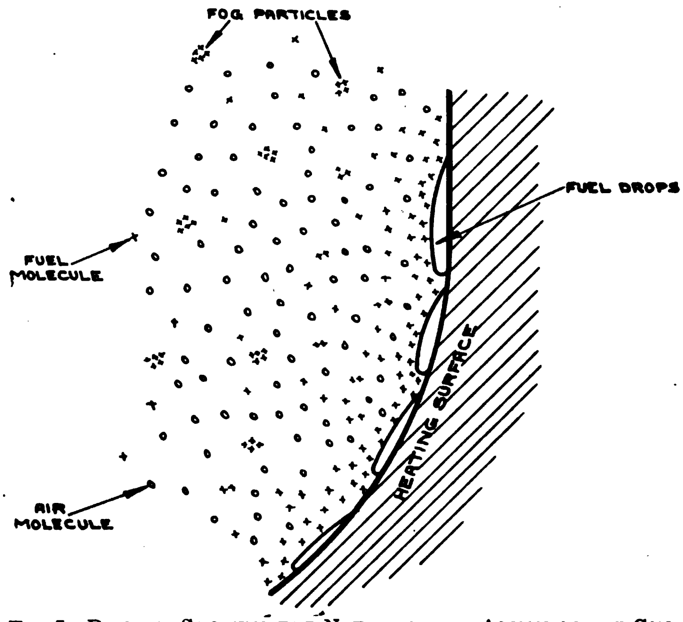

Fig. 7—Diagram Showing the Nature of the Action of the Construction Illustrated in Fig. 6

drops. Just off the liquid film the greatest vapor-density occurs, but as the distance increases and the air begins to lower the temperature, the molecules will begin to gather in droplets, in the action that we term condensation. It is obvious that it would be impossible to bring *all the air* into such contact with the liquid film that the vapor would be swept away, and uniformly diffused, within a few molecule paths of the liquid film; and it is only under such a condition that the temperature balances of Fig. 6 could be obtained. But it also is clear that the more completely we can direct and diffuse the air charge on the heating surface in the conventional hot-spot design, the lower the temperature and density can be next the liquid film, the lower can be the temperature of the liquid film and the wall itself and the lower the final temperature of the charge.

Regardless of the correctness of the theory of operation of this type of hot-spot, there are several advantages and disadvantages in practice that should be pointed out. As already outlined, reserve capacity can be obtained only by making the surface larger. Also, there is no inherent characteristic of this arrangement that would prevent liquid fuel from going into the engine. The heat capacity of the wall of any structure that could be used would be sufficient to prevent any lag in acceleration, provided the carbureter were made to give a charge of slightly increased richness, with a fuel of graduated volatility.

Experimental Determination of Heating Action

The foregoing analysis indicated the great importance of several considerations not previously investigated in the problem of properly preparing the fuel. We undertook, therefore, to build an experimental device that would allow us to regulate the three main variables governing the heat input; (*a*) the exhaust temperature, (*b*) the exhaust flow and (*c*) the area of heating surface; and to control the remaining variables affecting heat absorption; (*d*) the quantity of air, (*e*) the quantity of fuel, and, so far as possible, (*f*) the vapor density. The device used is shown diagrammatically in Fig. 8. The heating element was an iron plate, 3½ in. wide and about 8 in. long, exposed to the exhaust on the ribbed lower side, and receiving fuel from a series of jets placed across

the width of the plate at the air-intake end of its top surface. A slab of magnesia cement encased in a thin metal cover was used as a movable heat-insulating shield to give any desired area of heating surface up to the maximum. Thick layers of asbestos gasket were used to insulate the air-chamber from heat contact with the heating-plate. The fuel flow took place under gravity head and was regulated by hand at each speed to give the desired mixture-strength, the air supply being metered by the orifice-plate method. The fuel-jets could be set to give either a fine or a coarse spray and could be made to discharge in any desired direction. One side and both ends of the device were fitted with glass windows so that the liquid and fog conditions existing within could be easily observed.

It will be noted that by closing air entrance *a* and directing the fuel spray along the length of the plate, the conditions involved in Case No. 3 of our analysis could be reproduced. By using air entrance *a*, the conditions of Case No. 2 or No. 2a would be given, according to whether air entrance *b* was entirely or partially closed. And if the fuel spray were directed along the passage, instead of at the hot plate, with the air supply taken through entrance *b*, the conditions of Case No. 1 would exist.

The first test-runs with this device indicated that a broader range of investigation than initially planned was advisable. Unfortunately it was impossible to complete the program in time to include its results in the preprint of this paper, but it is hoped that at the Summer Meeting a resume of the observations and measurements of several different fuels can be given, the tests to include measurements of the mixture-temperatures, the heating surface necessary, the variation of temperature with increased heating surface and "volumetric efficiency" under the above variations.

Natural Variation of Air Temperature Under the Hood

In the quantitative computation of heat transfer, we first must take into account the very wide variations of temperature that the air charge and fuel supply undergo before they enter the intake-manifold system, on account of the large range of variation of hood temperature. Fig. 9 is given as a rough indication of the various changes in temperature that a molecule of air undergoes in getting from the external atmosphere to the cylinder port, without purposeful application of heat to the intake charge, other than the commonly used hot-air stove around the exhaust-pipe. Starting from atmospheric temperature, the temperature of the air is raised between 30 and 60 deg. in passing through the radiator. It has been our observation that there is a greater difference between the motometer temperature and that of the external air in summer than in winter. Perhaps some of the radiator engineers can tell why this is so. In the summer there is sometimes an additional rise of temperature under the hood due to the radiation of heat from the engine. The rise of temperature from the hot-air stove is presumably about the same in the summer and in the winter but on many cars an appreciable portion of the heat added by the stove in the winter is lost before it gets to the carbureter, because of the cooling effect of the fan-blast of relatively cold air on a long length of flexible tubing. The temperature-drop in the carbureter and the manifold due to vaporization is indeterminate, dependent upon the fuel, the temperature and the vacuum. In many cars the intake-manifold is so close to the exhaust that under full load the temperature is raised considerably by the cross radiation. We have sometimes gained 3 to 4 hp. in a maximum of about 70, by cutting off this radiation with asbestos board.

Fig. 9 will give an idea of the range of natural temperature-variation with which our intake systems have had to deal. Between the temperature of the air entering the intake system just after starting in the winter and that during a long run in the summer, there easily may be a difference of 120 deg. fahr. Very few current applications of heat to the intake charge, by either hot air or hot-spots, affect the temperature one half this amount. Any effort to attain minimum charge-temperatures in actual practice must include means for dealing with the natural temperature-variation under the hood.

Means of Temperature Control

With heating methods that approximate Fig. 1, the heating surface should perhaps be in two sections, one of which is in action at all times, and the other of which may be thrown open to the exhaust, either by a seasonally regulated valve, or by the dash mixture-control of the carbureter.

Arrangements such as that shown in Fig. 2 can be controlled within certain limits of temperature by using a hot-air stove on the exhaust line that has at least three times the heat capacity of those in common use today, with a valve adapted to cut-off part of this hot air and admit cold air as the engine warms-up. The regulation should preferably be automatic.

In the methods of Case No. 2, Figs. 4 and 5, no particular regulation for variation of atmospheric temperature is necessary. The heating action is almost independent of the outside temperature. With this type of construction, I have always recommended making the heater large enough so that cool air from outside the hood can be taken into the carbureter in the summer time.

Fig. 6, like Fig. 1, would perhaps be taken care of best by a regulable variation in hot-spot area. Difficulty is experienced in practice in confining the heat to the region where it is desired. In warm weather the heat from the warm hood atmosphere tends to conduct across the flange junctions and through the walls of the heating chamber. In future designs we may find thick heat-insulating material, or spacers of refractory tile, used to separate the hot from the cool portions of the heater.

Homogeneous Mixture Quality

As has been brought out in the foregoing, a homogeneous mixture requires a fine spray from the carbureter issuing directly into the heating region. If the fuel is allowed to condense or gather on the walls, it will reach the hot surface in waves and irregularly timed splashes, under which conditions the carbureter setting must always be somewhat rich, and many details of engine operation will suffer. Acceleration is always more difficult when there is a fuel lag between the carbureter and the heating surface.

The arrangement, common in many heavy-duty engines, of locating the governor between the carbureter and the hot-spot, is very bad. Everything indicates that the carbon deposit will be reduced to the minimum and crankcase-oil dilution eliminated only when this custom is disregarded and the carbureter is placed close to the hot-spot.

Several 1922 engines that have the property of operating very smoothly on extremely lean mixtures, have intake-manifolds that are characterized by a hot-spot at the

(Concluded on page 48)

Progress of British Aeronautical Research[1]

By Brig-General R. K. Bagnall-Wild

RESEARCH work for the Air Service comprises specific researches at establishments under the control of the Air Ministry and an important series of studies at the universities by arrangement with the Ministry. Much research work is being conducted at the universities independently of the Government, the results of which are of great importance to the future of flying. It is at least as important for the director of research to be in touch with cognate researches in the universities and other research organizations, at home and abroad, as it is for him to be cognizant of the developments aided by State finance, through the Air Ministry. The chief research establishments under his direct control are the Royal Aircraft Establishment at Farnborough, the National Physical Laboratory and the Air Ministry Laboratory. The practical applications of the results of research are tried out at the air-stations at Martlesham Heath, the Isle of Grain, Biggin Hill and, by courtesy of the Royal Air Force Coastal and Inland Areas, at certain other air-stations.

The directorate of research is, however, an engineering as well as a scientific organization. It acts as the engineering department of the Ministry and probably as much as four-fifths of its work relates to experimentation with specific appliances, efforts to develop such appliances along channels useful to aeronautics, and test performances of approved air-service material. The remainder is research, pure and applied.

Long experience has taught the universities the most suitable organization by which research as distinguished from ordinary technical work can be undertaken. This requires for its success a much greater personal freedom of work than had previously been the rule in Government organizations. In the last Wilbur Wright lecture before the Royal Aeronautical Society Major G. I. Taylor pointed out the great difficulty of organizing scientific research on the man-hour principle, adding that in his opinion research work is so difficult and exacting that a man can do his best work only when he is free to go where his researches lead, free to choose his own time for work, free from other duties which would divert his mind and lastly, free to sit down and produce no visible result for an indefinite period. In certain forms of applied research the problem is little different, and this is one phase of the work with which the directorate of research has to deal. The other, larger, but fortunately simpler, portion of the work consists in the design, development and ultimately the production of apparatus or methods which the work of the research organization shows to be capable of serving some definite purpose. In theory it may be possible to draw a sharp dividing line between these two classes of work, but in practice they tend to shade into each other in a bewildering way, and such boundary as may develop is liable to vary its position in accordance with the subject under investigation. With limited funds it is sometimes necessary for the research worker who may have started a new development, to see it through its infant difficulties, and mother it until it is in suitable form for the production stage.

It may, I think, be stated fairly that the biggest technical problem affecting civil aviation is the developing and perfecting of the airplane engine; the nature of the ground organization, the question of flying over or under clouds, the attractiveness of air travel to possible passengers and numerous other questions turn on the fundamental issue of engine performance.

Aeronautic Engine Research

The ultimate aim of aeronautic engine research is to produce an engine as free from breakdown as the average engine in a motor vehicle. It must be recollected, however, that the engine in a motor vehicle runs usually at about one-third its full brake-horsepower, whereas until recently it was customary for aircraft engines to run at from 80 to 100 per cent of their full-load capacity; it is only now, with the Napier Lion engine, that on the cross-Channel service it is possible to run with but 60 per cent of the maximum load. This, in my opinion, is the fundamental difficulty. This is, of course, much increased by the need for reducing the weight per horsepower to a minimum.

Many pin their faith to the almost magic properties of specific fuel-mixtures, but the work of Tizard and Pye and of the Ricardo Laboratory has shown that the efficiency and horsepower that are obtained from any mixture of the various volatile hydrocarbon fuels are within 2 or 3 per cent the same as from any others, provided the compression-ratio is not altered. Any advantage to be gained from a specific fuel-mixture must lie in its relative freedom from detonation, and therefore in its suitability for employment at a higher compression-ratio, with the resulting higher fuel-economy.

Experiments are being made to determine whether it is better to replace the carbureter system by some method of direct injection.

The experimental flight carried out by my predecessor, from Egypt to Mesopotamia and back, brought out with much emphasis the need for economizing in water on such flights. If water could be dispensed with entirely in favor of air-cooling a great step forward would be made. From a fighting point of view also this would be of great advantage since it would make the engine much less vulnerable to hostile machine-gun fire. The staff at Farnborough have long been of the opinion that this is a possible development, and we now have the Bristol Company's Jupiter engine of 380 b. hp., a nine-cylinder radial engine of which much is expected. There is also the Siddeley Jaguar, of 350 hp.; while in some other tests a single air-cooled cylinder has given as much as 100 hp., with a brake mean effective pressure of 134 lb. per sq. in. Some enthusiasts consider that an air-cooled engine of 1000 hp. should not be impossible of attainment; and that with the necessary cooling equipment an aeronautic engine of 2400 hp. at 750 r.p.m. may be realized. An advantage of large engines is that the lower number of

[1]From an abstract of a paper read at the 1922 Air Conference held at London. The author is director of research for the Royal Air Service.

revolutions per minute avoids the use of reduction-gearing to the propeller and reduces the weight about ½ lb. per b. hp.

When comparisons are made between the weights of air-cooled and water-cooled engines, it is sometimes forgotten that, although in the former case the weight of radiator and water is saved, it is necessary to include on the other side of the ledger an item for the additional weight of the air-cooled cylinders. In a comparison made recently it was found that although the gain due to the elimination of radiator, water, pipes and pump was about 0.70 lb. per b. hp., against this was an excess in cylinder and piston weight of 0.46 lb. per b. hp., so that the net gain was only 0.24 lb. If the fuel and oil consumption of the water-cooled engine be put at 0.50 lb. per b. hp-hr. and that of the air-cooled engine at 0.58 lb., the two become equal as regards overall weight when a 3-hr. supply of fuel and oil is included.

Navigation

The most difficult of all problems in connection with "navigation" is the provision of means to enable an aircraft pilot to locate the airdrome for which he is bound in foggy or misty weather and to make a successful landing. Unquestionably it is this difficulty that causes pilots on long routes to prefer to fly under, rather than over, clouds.

When in 1912 use was made of the precessional movement of a gyrostat to measure the velocity of roll on one of His Majesty's ships, it was not contemplated that by far the most successful application of the apparatus would be for air travel; it is known as the gyro-turning indicator. Ample experience has shown that it is more sensitive and much more rapid than any other method of indicating the turns of an airplane. Unlike a constant-azimuth *gyro*, this apparatus does not need to be delicately balanced as regards the position of its center of gravity and is therefore remarkably foolproof. Airplanes flying in a fog frequently get into turns without knowing it. When they do so, the magnetic compass, particularly when of an old pattern, is reasonably sure to indicate a turn in the wrong direction, and to thoroughly mislead the pilot.

Machines

The question of stability is being continually studied. Stability may be either inherent or automatic. Automatic stability is attained through the operation of some more or less complicated auxiliary mechanism, while inherent stability is due directly to the nature of the design of the aerodynamic surfaces, the disposition of weights and similar factors. If inherent stability could be attained there would be small need for automatic stability. The problem of longitudinal stability is to a large extent solved, but that of lateral stability will require much work.

Attention has been drawn to the interesting arrangement of slotted wings proposed by Handley Page. A large monoplane wing with a single long slot has also short slots in front of each aileron. Two other new Handley Page designs have appeared. Earlier trials showed remarkable results. A D.H.9 airplane fitted with a Puma engine was found, when equipped with a Handley Page wing, to have both greater aerodynamic efficiency and greater climbing capacity than with the standard wing. The climbing slope in the one case was 1 in 7.2, and in the other 1 in 10.4. This is important when clearing obstacles around an airdrome. For getting off and onto decks, it was found that the launching run with the Handley Page wing was less than half that necessary with the standard wing under similar conditions.

The ability to fly from or land on the deck of a ship is becoming increasingly important. Amphibians as well as airplanes should be capable of doing so. Descents onto water cannot always be avoided and "landings" of this kind with airplanes are liable to be expensive. Experiments are being made to determine the sea-keeping abilities of the large type N-4 multi-engine flying boats. These weigh about 30,000 lb. and the power developed by its engines totals 2600 hp.

It is difficult to know what to say of the helicopter. Efforts are proceeding and we shall, I hope, be rewarded for our foresight. Mr. Brennan's helicopter has flown to the extent of lifting the pilot and 250 lb. of useful load, an encouraging preliminary flight.

Gasoline Piping

Perhaps the most troublesome material in present-day aircraft is the rubber tubing used for conveying gasoline. It has to be of "gasoline-resisting" quality. The "petroflex" piping, devised by Blaisdell, affords a very useful means of avoiding both the rapid aging of the rubber and the stiffness of the copper. Petroflex is composed of about 10 layers, glued together, of the intestines of Chinese hogs. Around these layers are layers of canvas, fireproofed. Outside these in turn, as a final protection, is a spiral of wire, which in the case of land machines is of aluminum. This piping appears to be totally unaffected by gasoline, although care has to be taken to keep water away from the inside. Experiments have shown that the piping will stand an internal pressure of 200 lb. per sq. in.

ETCHING REAGENTS FOR ALLOY-STEELS

SEVERAL series of "sequence etchings" have been tried out at the Bureau of Standards on a specimen of high-speed steel that was in the "as received from the mill" condition. Sequence etching means the etching with two or three reagents in successive order without any repolishing of the section between the etchings, and the taking of micrographs after each etching at the same spot in the microsection to note any changes in the microstructure produced by the last etching as compared with that developed by the preceding etching reagent. The reagents used in various combinations were: (*a*) Dilute NH_4OH solution together with a weak electric current, the specimen being made positive pole; (*b*) a 2-per cent alcoholic solution of nitric acid; (*c*) boiling sodium picrate; and (*d*) Murakami's reagent, which is a solution of potassium ferricyanide and sodium hydroxide at boiling temperature. The results, as judged from the behavior toward the various etching reagents tried, appear to show that there are at least three different constituents present among the imbedded globules or particles in the specimen of high-speed steel, in the "as received from the mill condition," though any attempt to state the nature of these constituents would, at present, involve speculation.

To obtain tungsten carbide in sufficient bulk for making various etching tests thereon, plans are now under way to try carburizing metallic tungsten, ½-in. diameter bar stock, by the "cementation" method.

The Crankcase Oil Dilution Problem and Its Solution

By WILLIAM F. PARISH

DETROIT SECTION PAPER *Illustrated with* CHARTS AND PHOTOGRAPH

STARTING with the premise that while the present automotive type of internal-combustion engine was designed to use the volatile fuels, the increased production of these engines has made it necessary to raise the end-point of the fuel with each succeeding year, the author points out that this practice has resulted in the use of a fuel that is not completely vaporized and leaks into the crankcase in a liquid state and dilutes the lubricating oil. The effect upon the viscosity of the lubricant of this fuel dilution is commented on and the viscosity limits of lubricating oils are established. The relation between the viscosity of the lubricant and the efficiency of the engine in which it is used as determined experimentally by C. W. Stratford several years ago is brought out graphically and also in tabular form. How this dilution has forced the refiners specializing in internal-combustion engine lubricants to increase the body of their various grades of oil in response to the demands of the motoring public, the viscosity having increased in some cases almost 95 per cent between the years 1913 and 1921, is pointed out. Tables are presented to substantiate this claim and also to show the differences that existed in 1920 between the lubrication recommendations of three leading oil companies and the actual sales of the three different classes of their product.

The carbon-forming properties of oils and the effect of heavy oil upon engine efficiency are discussed. Although the dilution of the lubricant should be a minimum with one engine operating at full load on the dynamometer stand, results of engine tests extending over the last 10 years show an increase in such dilution of from 2 to 12 per cent. A series of service tests conducted at Chicago slightly more than a year ago showing a dilution of from 15 to 41 per cent in less than 100 miles of running is mentioned as well as other series of tests on passenger cars, tractors and trucks, in which the percentage of dilution increased with the length of the time run, the viscosity in one case decreasing almost 87 per cent after the engine had run 563 miles in a single month. The effects of the diluted lubricant upon the oil-consumption, friction of the engine and the wear are all emphasized. The history of the development of the lubrication of various forms of prime-mover is traced and commented upon at considerable length.

As a solution of the problem of crankcase dilution the author offers a system for crankcase oil regeneration consisting of four main elements by which the fuel and water dilution are automatically removed and the sediment composed of carbon particles, sand and minute pieces of metal filtered out. The operation of this system, which it is claimed will not interfere with the present lubricating system of the engine and functions equally well with the splash and forced-feed systems is described and illustrated. Charts showing the results obtained from comparative test-runs with and without the oil refining system in use are also presented. These indicate that the use of the system practically eliminates crankcase dilution.

[1] M.S.A.E.—Consulting lubrication engineer, Chicago.

THE present automotive type of internal-combustion engine was designed to use the volatile and dry fuels. The automotive industry has been producing so many of these engines that the oil industry has been unable to keep pace with the fuel demand without cutting further into the crudes, and by various and sundry methods has produced fuels with end-points

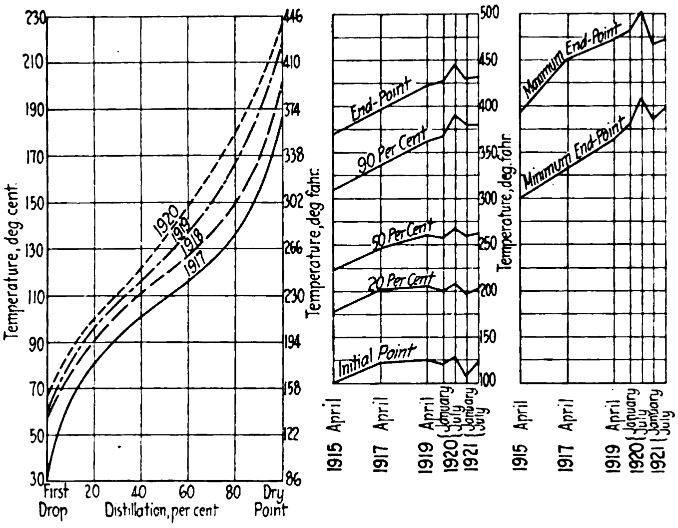

FIG. 1—CHARTS SHOWING HOW THE END-POINT OF THE FUEL SUPPLIED FOR USE IN INTERNAL-COMBUSTION ENGINES HAS RISEN FROM 1915 TO 1921

increasing year by year, as is shown by the charts reproduced in Fig. 1. The result is the unbalanced condition of the original engine, made for dry fuel, apparently using a fuel that passes into the cylinders in a more or less liquid state, where it leaks into the crankcase, combining with the lubricating oil and causing the present serious dilution problem. It is possible to reduce materially the amount of liquid fuel entering the cylinders, by the application of considerable heat and, in this connection, the best results will be secured through using the greatest heat. Heating the mixture to secure better vaporization makes it necessary to lower the compression, which is reflected in lower mileage to the gallon of fuel. It is also possible to catch the flow of raw gasoline in the manifold to some extent, and reatomize it sufficiently to secure better running engines on less fuel. This is being done by many manufacturers and by a number of appliances now on the market. To the extent to which these appliances are used, there is a reduction of the raw gasoline entering the cylinders and draining to the crankcase. Dilution from direct raw-fuel leakages is due to the inability of the carburetion and manifold systems to handle the heavy fuel properly. Any improvements in this direction will naturally tend to alleviate dilution from this cause.

Should the direct leakage of raw gasoline be entirely

overcome, and an acceptable remedy for this condition must be found, there will still be the leakage of the charge in the cylinders during the compression stroke. At this time, while the charge is compressed to a high degree, the mixture is forced by the rings. This mixture contains both the high and low-boiling fractions of the gasoline. Upon entering the crankcase the finely divided oil spray assists the condensation of the fuel, with the result that the lubricating oil in the crankcase absorbs and holds all of this condensed fuel that will not boil away or be redistributed at the engine temperature

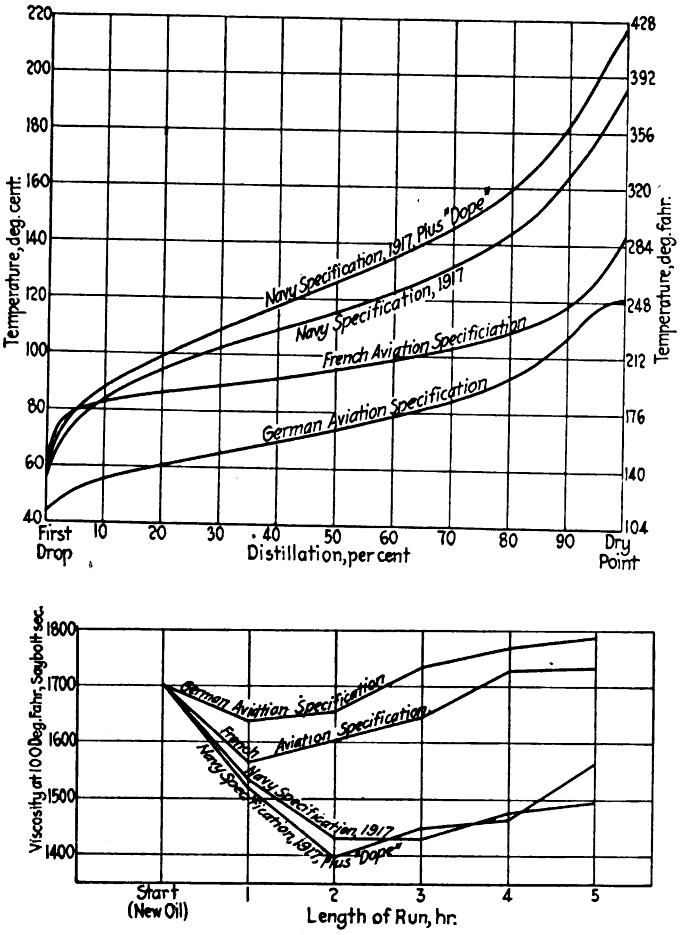

FIG. 2—CHART SHOWING HOW THE VISCOSITY OF A LUBRICATING OIL WAS AFFECTED BY DILUTION WITH FOUR GRADES OF GASOLINE, THE DISTILLATION CURVES OF WHICH ARE SHOWN IN THE UPPER PORTION OF THE ILLUSTRATION

and condition. As the engine and the oil become colder, the ability of the oil to hold the more volatile fractions increases. If the engine is operated at high temperatures, these volatile fractions will be liberated to a great extent. The fuel leakages are continuous irrespective of the amount of fuel held by the lubricating oil. When dilution is at a comparatively steady point, the total leakage per hour continues, but the engine disposes of the excess.

The amount of fuel escaping by the piston-rings during the compression stroke is naturally related to the fuel consumption per brake horsepower-hour. When fuel consumption is low, as a result of fine adjustment and careful operation, dilution of the lubricating oil in a given period of operation is less than when such care is not exercised and the fuel consumption per unit power-output is greater. Dilution is greatest when the operator is unskilful, the rings are worn and the compression and the power output are at a maximum, and intake-manifold vacuum is at a minimum.

TABLE 1—RELATION BETWEEN THE VISCOSITY OF THE LUBRICANT AND THE ENGINE EFFICIENCY

Viscosity at 100 Deg. Fahr., Saybolt sec.	Brake Horsepower	Consumption per Horsepower Hour, lb. Fuel	Oil
100	28.0	1.40	0.0430
125	32.6	1.05	0.0410
150	33.5	0.80	0.0380
175	33.7	0.70	0.0350
200	33.6	0.70	0.0320
400	33.4	0.70	0.0270
600	33.2	0.70	0.0254
800	32.8	0.71	0.0246
2,300	31.2	0.74	0.0225

The fuel leakage past the piston-rings is naturally dependent upon the condition of the pistons, cylinders and rings. Correctly designed and well-fitted rings, in true cylinders, resist these leakages until a point is reached where the dust and dirt that enters the engine with the carbureter air, in conjunction with the diluted and reduced lubricating oil film on the cylinder walls, brings about a cylinder wall and ring face wear, the worst effect of which is the gradual widening of the piston-ring groove caused by the up-and-down slapping of the ring. This creates a direct passage for the leaking fuel and the escaping or "pumping" oil.

The dilution of the lubricating oil in a given engine was shown in the fuel tests conducted at Washington[2] where four grades of gasoline of the various distillation ranges shown in the upper part of Fig. 2 were used. The lower portion illustrates the effect of the various fuels on the viscosity of the same lubricating oil.

VISCOSITY LIMITS OF OIL FOR LUBRICATION

The practice of lubrication engineering is based largely on experience. Certain definite rules have been laid down as the result of the experiences of many men who have worked on lubrication problems in all parts of the world. These rules establish limits in viscosity for various classes of work. For instance, the lightest and most rapidly running cotton spindle is lubricated with oils of between 54 and 70 viscosity Saybolt sec. at 100 deg. fahr. Oils lower than the minimum limit cause wear and the blackening of the oil in the spindle base, and that establishes the limit for thinness. Heavier oil than the maximum causes heating of the spindle and loss of power and speed; thus the limit of thickness is obtained. Between these two points the lubrication of the spindle is perfect,

TABLE 2—CHANGE IN VISCOSITY OF LUBRICATING OILS SOLD BY 10 COMPANIES FROM 1913 TO 1921

	Light	Medium	Heavy
Average 1913 Viscosity at 100 Deg. Fahr., Saybolt sec.	162.7	209.7	250.0
Minimum 1913 Viscosity at 100 Deg. Fahr., Saybolt sec.	132.0	156.0	210.0
Maximum 1913 Viscosity at 100 Deg. Fahr., Saybolt sec.	180.0	338.0	295.0
Average 1917 Viscosity at 100 Deg. Fahr., Saybolt sec.	167.5	259.4	392.0
Minimum 1917 Viscosity at 100 Deg. Fahr., Saybolt sec.	137.0	217.0	242.0
Maximum 1917 Viscosity at 100 Deg. Fahr., Saybolt sec.	205.0	330.0	868.0
Average 1921 Viscosity at 100 Deg. Fahr., Saybolt sec.	211.0	299.1	485.7
Minimum 1921 Viscosity at 100 Deg. Fahr., Saybolt sec.	176.0	196.0	226.0
Maximum 1921 Viscosity at 100 Deg. Fahr., Saybolt sec.	351.0	565.0	785.0

[2] See *Mechanical Engineering*, March, 1920, p. 164.

but if these limits are exceeded detrimental action results. It is to be noted in this connection that the spindle will continue to turn and put twist in the yarn even though it may be undergoing excessive wear or consuming power exorbitantly.

In exceeding the limits of good lubrication practice, the changes that generally take place are too slight and too gradual to be noted immediately. Accumulated, they become of the greatest concern. The experienced lubrication engineer can define limits for the proper lubrication of every classification of machinery. In the case of the internal-combustion engine, however, the lowering volatility of the motor fuels during the last 10 years has introduced several variable factors which make it virtually impossible to standardize the limits of proper lubrication.

When these engines were operated on fuels that caused but little dilution of the lubricants, the oils in common usage in this country were highly finished light-engine or dynamo oils. These oils had an average viscosity of about 190 sec. at 100 deg. fahr.

TABLE 3—RELATION BETWEEN THE LUBRICATION RECOMMENDATIONS OF THREE LEADING OIL COMPANIES AND THEIR ACTUAL SALES IN 1920

Company	Grade of Oil	Chart Recommendations, per cent	Actual Sales, per cent
A	Light	43	20
	Medium	45	60
	All Heavy Grades	12	20
B	Light	5	10
	Medium	85	57
	All Heavy Grades	10	33
C	Light	8	5
	Medium	86	49
	All Heavy Grades	6	46

Fig. 3, which is reproduced from a paper by C. W. Stratford entitled Automobile Lubrication,[8] shows graphically that an oil having a viscosity of 180 sec. at 100 deg. fahr. is the turning point for engine efficiency. Oils lighter than 180 sec. reduce the brake horsepower by increasing the frictional horsepower. The fuel and oil-consumption per brake horsepower-hour are also greatly increased. As the oils used are increased in viscosity, up to 2300 sec. at 100 deg. fahr. the brake horsepower decreases with a slight increase in the fuel-consumption and a reduction in the oil-consumption. The tests from which the results given in Table 1 were obtained were conducted in a testing laboratory owned by an oil company.

Influence of Dilution Upon the Oil Trade

An extremely interesting phase of the engine oil situation has been created by the dilution problem. Oil companies specializing for this trade have been forced by public demand to increase the body of the various grades of their oils gradually. The motoring public have apparently been the judges as to what their engines required, as evidence will be presented showing that quantities of the various grades of oils actually purchased have not balanced with the charts of recommendations that represent the seller's idea of what should be sold. The oil companies enjoying the bulk of the automotive oil business have seadily increased the viscosity of their brands. The data given in Table 2 bear this out.

[8] See TRANSACTIONS, vol. 10, part 2, p. 86.

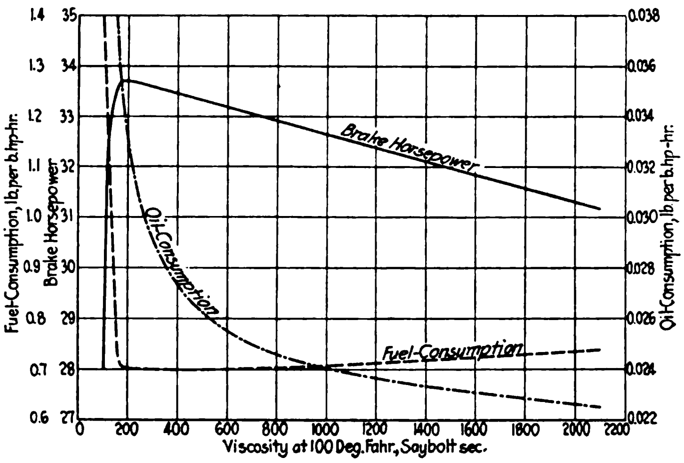

FIG. 3—EFFECT OF CHANGES IN VISCOSITY ON THE BRAKE HORSEPOWER AND FUEL AND OIL-CONSUMPTION OF AN INTERNAL-COMBUSTION ENGINE

The tendency is toward engine oils in general increasing in viscosity. While some of the lighter-bodied oils persist, it has been found that they, as a rule, are not being marketed by companies enjoying the widest trade. The greatest increase in viscosity for all three grades has been made in the best-known trademarked oils. An analysis of the recommendation charts published by three leading oil companies having national distribution shows the interesting tendency of the heavy-oils consumption. The total number of cars, trucks and tractors supposed to be in operation during 1920, as determined for all production models, were allowed a fair amount of oil per car for the season. It was assumed that all cars in operation during 1920 would use the grade of oil recommended by each respective oil company. The distribution of the three grades, in percentages of the total

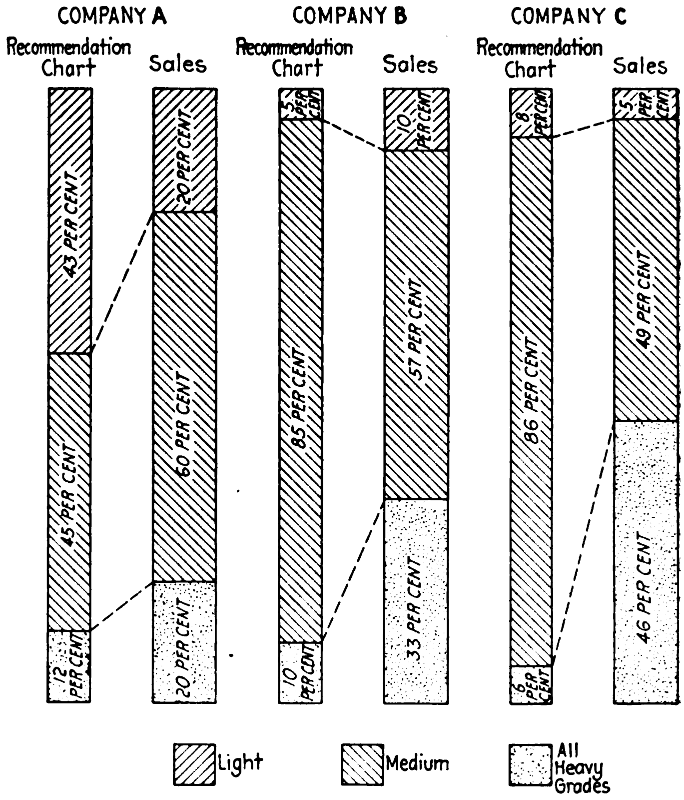

FIG. 4—RELATION BETWEEN THE RECOMMENDATIONS OF THREE OIL COMPANIES FOR AUTOMOTIVE LUBRICATION AND THE ACTUAL SALES OF THE VARIOUS GRADES

amount required, is given in Table 3 and shown in Fig. 4 with each company's actual sales for the year 1920 by grades expressed as percentages of the total. Thus it is possible to see what the experts of the various oil companies recommend for the lubrication of all automotive equipment in the country, and what the owners of the equipment actually purchase. The oils sold by Company *A* are refined from asphaltic and paraffin-base crudes; those marketed by Company *B* are obtained from asphaltic-base crudes only; while the paraffin-base crudes are the sole source of supply for the products of Company *C*.

These figures, dealing with many millions of gallons of oil, and with a considerable portion of the motor-car oil trade of the country, clearly indicate that the public in general demands heavier oils than the oil companies officially recommend in their published charts.

The oldest of the recommendation charts has not changed materially in respect to the grades recommended, since it was originally issued. The oils furnished under the original grade designations have, instead, been changed in viscosity. For instance, one company recommended the same brand and grade for Ford cars in 1913 as in 1921, but the viscosity of this oil has been changed from 170 Saybolt sec. at 100 deg. fahr. to 289 sec.; from 94 to 130 Saybolt sec. at 130 deg. fahr. and 43 to 52 Saybolt sec. at 210 deg. fahr. In accordance with popular grading, this oil was a light oil in 1913, but is now put out as a medium oil. The lubrication of the Ford car has not changed in principle. The engine and the lubricating system are substantially identical with those of the earlier model, and no condition has been introduced that would require a heavier oil, except the dilution, which now occurs but did not exist to a noticeable extent when the lighter oil was supplied.

The unprecedented and unlooked for demand on the part of the public for heavier engine oils greatly upset the oil market for the heavier stocks from which the heavy oils are made. Fig. 5 indicates the trend of automotive demand for lubricants for the years 1919, 1920 and 1921. The prices of the oils used as light oils and as compounding agents for the heavier oils are also plotted. The bright filtered stock, the most expensive of the stock oils, increased very rapidly in price as the extraordinary demands were made upon it. This shortage of the bright stock oil was reflected in the rapid increase in the sale and advance in price of the cheaper filtered cylinder stocks, and finally of the dark unfiltered stocks, which were substituted by a number of manufacturers for the brighter oils with the result that their finished engine oils were black. It is a matter of trade knowledge that every national marketer of automotive lubricants was unprepared for the high and overbalanced demands made by the public for heavy engine-oils during the 1920 season. Figs. 4 and 5 show the way in which the demand developed with the result on the heavy-stock oil market. The advertisements of at least one large oil company during 1921 gave many arguments for not using heavier oil, and can be taken as evidence that there was an attempt being made to counteract the tendency of the public to purchase heavier oils.

Carbon-Forming Properties of Oils

It is possible to make a laboratory predetermination of the carbon-forming nature of an oil by the Conradson Method, which has been officially adopted by the American Society for Testing Materials for this purpose. It

TABLE 4 — MAXIMUM PERMISSIBLE CARBON-CONTENT IN OILS SUPPLIED THE GOVERNMENT

Grade	Viscosity at 100 Deg. Fahr., Saybolt sec.	Conradson Carbon, per cent
Extra Light	140 to 160	0.1
Light	175 to 210	0.2
Medium	275 to 310	0.3
Heavy	370 to 410	0.4
Extra Heavy	470 to 520	0.6
Liberty Aero[5]	90 to 100	1.5
Liberty Aero[6]	125 to 135	2.0

[5] Viscosity values are at 212 deg. fahr.
[6] Summer grade. Viscosity values are at 212 deg. fahr.

has been determined that the Conradson carbon increases in all oils of the same base as they increase in viscosity. The report of Committee on Standardization of Petroleum Specifications for the United States Government[4] specifies the figures given in Table 4 as the maximum permissible carbon-content by the Conradson method.

Engine tests with various oils indicate that the amount of carbon found on the pistons and in the engine corresponds very closely with the relative amount of carbon of the same oils shown in the laboratory with the Conradson residue test. It is further a matter of common knowledge that the use of heavy oils leaves the engines in worse condition as to carbon. The engines now have to be cleaned of carbon deposits at least every 4000 to 6000 miles, when formerly it was unusual to have this done under 15,000 miles.

Other difficulties encountered while using the heavier grades of oil are lowered mechanical efficiency and difficult starting, especially in cold weather, with heavy loads on the starting-motor causing rapid discharge and short life to the storage batteries.

In general, the oil companies' resistance to the sale and use of heavy oils is based on sound engineering principles. To use one of the heaviest oils in a given engine would increase the frictional horsepower as much as 40 per cent, as compared to a light oil, and would also cause an increase of at least 50 per cent in the frictional temperatures of the bearings, considering the frictional temperature as the difference between bearing and atmospheric temperatures as is brought out in Fig. 6. C. F. Kettering in a paper entitled Fuel Research Developments says that the engine frictions are excessive, and that engine friction is one of the biggest problems confronting the automotive engineer.[7]

Lubrication engineers agree that it is desirable to use the lightest oil that will consistently keep the surfaces apart at working temperatures. The success of internal-combustion engine lubrication with lighter oils was established some years ago, the limits being determined by careful work at a time when there was practically no dilution. With the advent of the dilution problem it was no longer possible to work within such fine limits. In starting in with the correct oil, a high percentage of dilution is produced in a short time. Using a heavier oil merely delays arriving at the same point of thinness. There is an interesting, though complicated, situation in this part of the problem; the engine operates at maximum efficiency with a light oil; the oil industry desires to sell the lighter oil; but the public demands heavier and heavier oils.

The Dilution of Oil in the Present Automotive Engine

Dilution should be at a minimum when an engine is operating on the dynamometer stand under full load,

[4] See Bureau of Mines Bulletin No. 5; also THE JOURNAL, March, 1921, p. 220.
[7] See THE JOURNAL, November, 1921, p. 344.

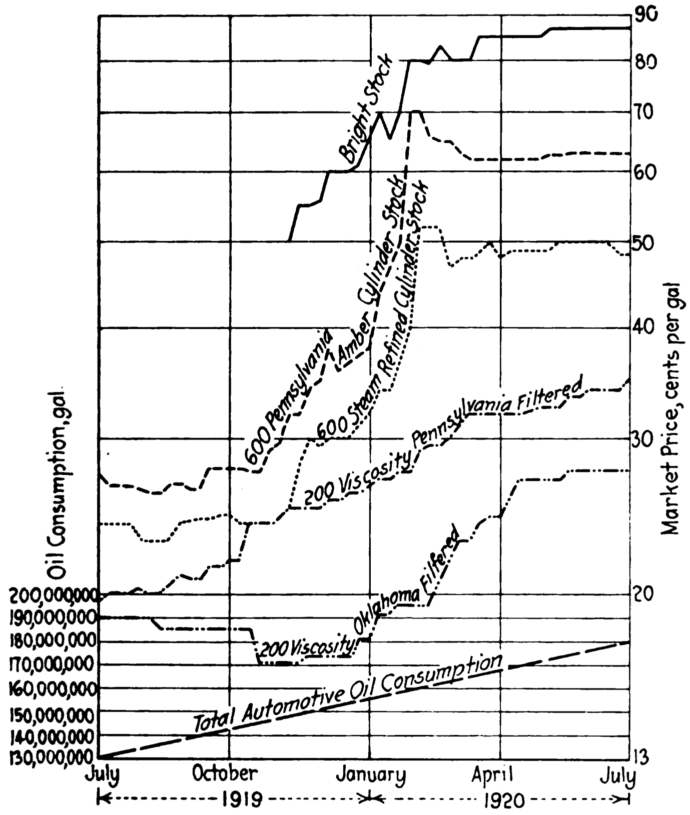

FIG. 5—CHART GIVING THE PRICES OF VARIOUS GRADES OF LUBRICATING OILS FOR THE YEAR ENDED JUNE 30, 1920

with the temperature and fuel controlled and without the effects of starting and stopping. The results of several block-tests of engines under full load conducted intermittently during the last 10 years, are listed in Table 5.

TABLE 5—RESULTS OF BLOCK-TESTS OF ENGINES

Year	Number of Cylinders	Number of Oils	Number of Refiners	Dilution, per cent
1911	4	7	6	2.0 to 9.0
1913	4	14	10	4.0 to 14.0
1919	6	7	5	5.5 to 9.0
1920	6	9	4	7.1 to 12.0

In all of the tests mentioned above, the engines were thoroughly cleaned out, flushed with the oil that they used in the test and run from 4 to 10 hr. on the block under prescribed control conditions. The oil was then removed and tested in every particular.

Tests of tractor engines on the block using gasoline of the grade sold in 1919 show dilution of from 8 to 15 per cent. Kerosene-burning tractor engines show dilution of from 30 to 50 per cent in 5 to 8 hr. using 1921 fuel. The question of dilution of the lubricating oil in tractors presents a far more serious problem than that of the passenger car or motor truck. This is because tractors are subjected to a more severe service and the burning of kerosene and the heavier distillates brings in additional complications.

However, from observations in various tests it may be considered that dilution of lubricating oil in tractor engines operating on gasoline in cold weather amounts to between 20 and 50 per cent in several days. In the hottest weather this dilution ranges between 10 and 30 per cent. With kerosene, the dilution runs between 30 and 70 per cent over a period of several days' operation, the per cent of dilution depending upon the original amount of oil in the crankcase, the highest percentages being obtained, of course, in the coldest weather.

SERVICE TESTS OF DILUTION

An examination of the dilution conditions in nine trucks and cars conducted in Chicago in the winter of 1920 revealed the fact that after runs of under 100 miles the fuel in the lubricating oil amounted to from 15 to 41 per cent and that much of this dilution, in some cases as high as 22 per cent, occurred the first day.

The following winter a special series of tests was made on the engines of cars and trucks equipped with either carbureter or manifold heating devices. One of these, a truck engine that was run the equivalent of 2 miles without going out of the garage, showed 1.5-per cent dilution. Later in service the distance was increased to 50 miles with a resulting dilution of 20.0 per cent and after a further 29.6 miles the dilution increased to 26.5 per cent. An air-cooled engine, run 2 weeks, showed 18-per cent dilution, while a roadster, run 524 miles in 1 week, showed 22-per cent dilution. A passenger car showed 20-per cent dilution after having run 576 miles in 3 weeks, while another of the same make and model showed a dilution of its lubricating oil of 31 per cent after having gone 668 miles in 3 weeks.

A medium-priced car, run 300 miles in 3 days, showed 14 per cent; another, after making 231 miles in 10 days showed 10-per cent dilution and a cheap car's lubricating oil was diluted 11 per cent after making 404 miles in 1 week. A sleeve-valve engine that operated 563 miles in 1 month, registered 47-per cent dilution.

In this last-named instance, the lubricating oil at the outset had a viscosity at 100 deg. fahr. of 360 Saybolt universal sec., while at the end of the test run the viscosity of the oil taken out of the engine was 46 sec.

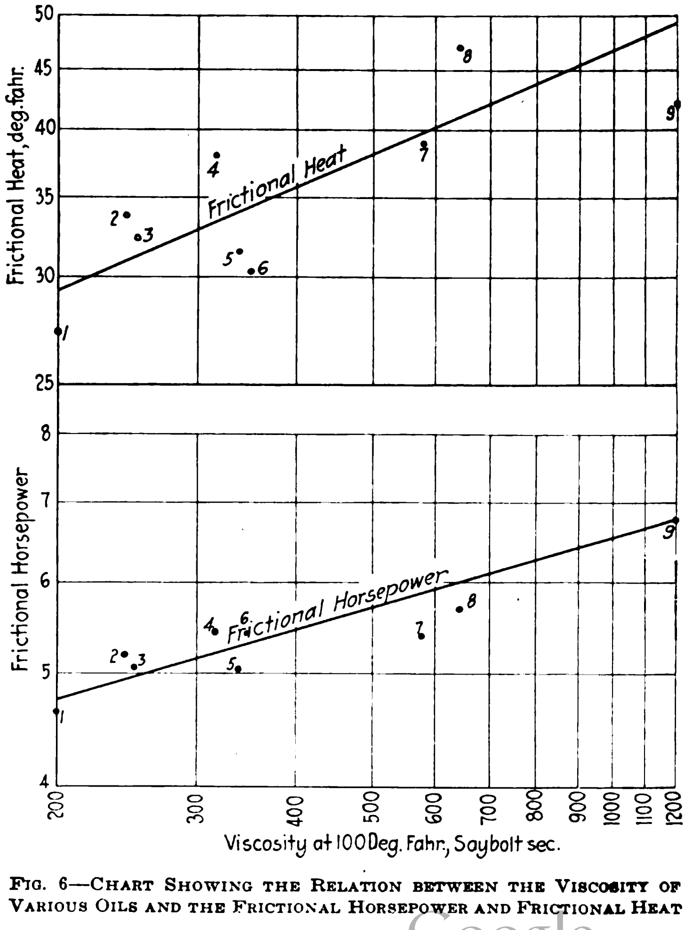

FIG. 6—CHART SHOWING THE RELATION BETWEEN THE VISCOSITY OF VARIOUS OILS AND THE FRICTIONAL HORSEPOWER AND FRICTIONAL HEAT

at 100 deg. fahr., demonstrating clearly the result of its mixture with the fuel. At a temperature of 212 deg. fahr. which would remove some of the diluent in the sample this mixture still had a viscosity of 36 sec. as compared with a viscosity of 54 sec. for the original oil at the same temperature.

After a run of 214 miles in five days a high-priced car showed a dilution of 22 per cent, while a low-priced car, after operating 380 miles in 4 days showed dilution amounting to 26 per cent. A medium-priced car that had run 445 miles in 1 month showed 27.5 per cent and a high-priced car that had covered 700 miles in 3 weeks revealed dilution of 9.5 per cent.

In a series of experiments with fuels having end-points averaging 436 deg. fahr. conducted between Nov. 12 and Dec. 20, 1920, the data on dilution presented in Table 6 were obtained. In this test, a truck engine showed dilution of its lubricating oil amounting to as high as 46 per cent. Other trucks belonging to the same fleet using the same fuel and the same oil over the same period of days, gave the following results:

TABLE 6—RESULTS OF DILUTION TESTS ON TRUCK ENGINES

Make	Tonnage	Dilution, per cent
G	5	17
N	2	13
G	3½	17½
I	2	25½
G	5	14
G	1½	12½
M	7½	11
N	2	31
G	3½	20
G	3½	11

Mass Figures on Dilution

In discussing a paper on Dilution of Crankcase Oil at a meeting of the American Society of Lubrication Engineers in October, 1921, an engineer who is engaged in the installation of plants for the reclamation of used engine oils, reported[8] that in 1921 from collections of 5000 gal. of oil per week in Kansas City he distilled 28.5 per cent of a fraction lighter than light engine-oil. The runs brought about 13.5 per cent of distillate from 55 deg. Baumé to 45 deg. and 15 per cent of distillate from 46 to 38 deg. Baumé.

Another reclamation project at Toledo reported that from more than 500,000 gal. of oil collected from garages and service stations in Chicago, New York City, Detroit and Toledo during the summer of 1921, the company distilled 30 per cent of fractions lighter than light engine-oils. The runs averaged 10 per cent of naphtha of about 52 deg. Baumé gravity, and 20 per cent of distillate of from 45 to 46 deg. Baumé.

The Viscosity of Diluted Oils

The effect on the viscosity of various oils by the mixture of fuel is shown clearly in Fig. 7. The diluent removed from the lubricating oil after it has run in the engine resembles kerosene more than it does the original gasoline. This is illustrated in Fig. 8. The curves also show that the diluent has a range of boiling points from 100 to 150 deg. fahr. higher than the same fractions of the gasoline that was used for fuel in the engine.

Previous investigations have determined that as a rule the oil-consumption in an engine is related to the viscosity or body of the oil itself.[9] A Curtiss 100-hp. aeronautic engine, for example, has a consumption rate of 0.880 gal. per hr. of lubricating oil with a viscosity of 54 sec. at 212 deg. fahr. and 0.194 gal. per hr. for oil with a viscosity of 135 sec. at 212 deg. fahr. The lighter oil is consumed 4.5 times faster than the heavier oil. Dilution in this case ran 2.0 per cent with both oils. The fuel used had an end-point of 350 deg. fahr.

When dilution thins down the lubricating oil, the consumption naturally increases. This feature must be given due consideration whenever an attempt is made to offset dilution by the use of heavier oils. The heavier oils produce the more carbon and the dilution of these oils with fuel increases the consumption of the oil and yet does not reduce the carbon-forming nature of the oil in the mixture. The formation of carbon is a continuous process and has a definite relation to the amount of oil that creeps up to and over the piston-heads where it is decomposed. In recording engine test data, the amount of dilution in the oil should be deducted from the gross quantity of oil remaining in the crankcase, as shown by C. M. Larson in his paper entitled Determination of the Percentage of Dilution.[10] The remainder will be the actual amount of oil consumed. In the years when dilution was not recognized, it was customary to make no correction in the consumption of lubricating oil. The records of the extent of the dilution in the oil are incomplete and inaccurate. For instance, if at the expiration of a 10-hr. run, the dilution amounts to 20 per cent and the same amount of "oil" is removed as was originally put in the engine, it has been customary to report "no consumption." As a matter of fact, one-fifth of the original oil has been consumed and its place taken up by condensate from the leaking fuel.

Effect of Diluted Oil on Engine Friction

Approximately one-quarter of the engine friction is due to the bearings and the auxiliaries, the other three-quarters being caused by the piston-rings rubbing on the cylinders. Reducing the viscosity of a heavy oil by mixing kerosene with it will reduce the power required to overcome the friction in the main bearings, until the reduction in the viscosity reaches a point where the surfaces strike. Then there is a rapid increase in frictional power as the striking area becomes larger. Reducing the viscosity of a heavy oil with kerosene for the lubrication of the cylinders will not bring about a reduction of the frictional horsepower to the same degree as with the lubrication of the bearings because the heat of the cylinder-walls will throw off some of the diluent at that point; consequently the diluted oil on the cylinder will be thicker and produce more resistance than the same oil in the crankcase.

My records show that used oil taken from automotive engines with a viscosity of 200 sec. at 100 deg. fahr. scarcely ever appears. Most of the oils taken from the engines are under 150 sec., many are under 100 sec. and a large number are in the 40 and 50-sec. viscosity class, all readings being in Saybolt seconds, at 100 deg. fahr. With any comprehensive standard of proper lubrication for the automotive engine in mind, the automotive or lubrication engineer will readily appreciate that with these viscosities the low limit for safety has been passed and that wear is taking place to such an extent that it will rapidly become detrimental to the engine.

Oil, primarily, is for the purpose of keeping surfaces apart so that they will not strike. If the oil becomes so thin that the film interposed between the two surfaces

[8] See *Scientific Lubrication and Liquid Fuel,* November, 1921, p. 11.

[9] See *Journal of the American Society of Naval Engineers,* vol. 32, p. 45.

[10] See *Scientific Lubrication and Liquid Fuel,* June, 1921, p. 16.

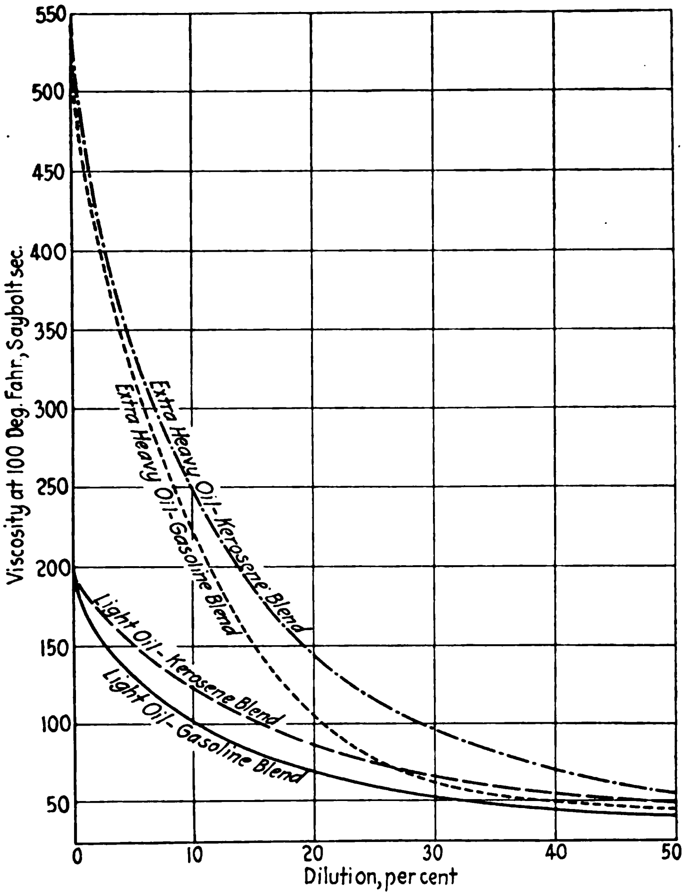

Fig. 7—Effect on the Viscosity of Various Oils of Adding Fuel to Them

is insufficient to prevent contact of those surfaces, local heating is immediately established where the striking takes place and the oil at that spot becomes suddenly thinner at the very moment when thicker oil is needed. As the oil becomes thinner through heat or dilution the locality where the striking occurs becomes more extensive in area and the wear is increased. In the automobile engine this wear may continue for some time as in the case of the wear of the cotton spindle, but eventually it causes a marked deterioration in the physique of the engine. Many of the ills of the engine can be traced directly to this wear. Among the common automobile engine ailments caused by this condition are oval cylinders caused by wear, worn piston-rings, pumping of oil into the explosion chambers, loss of compression, the noisy operation of the engine and burning out of the bearings. In the last analysis engines show such great depreciation largely because of the lack of lubricating properties in the mixtures they are forced to use as lubricants.

The Development of the Lubrication of Prime-Movers

It is of extreme importance to know the problems of lubrication that have influenced the development of all prime-movers. In general problems of lubrication are the determining factors as to whether a type of machine shall exist or not. A machine that cannot be lubricated will not survive. The history of this development shows a long lane, strewn with the skeletons of discarded engines, once popular, but now obsolete, largely due to lubrication troubles. Types of machines that it has been possible to lubricate have survived and have been of the greatest use to our civilization. Types have improved to stages where they give no trouble and develop into prime necessities.

The Westinghouse crankcase steam engine, popular some 30 years ago, was lubricated by a mixture of water and oil placed in the crankcase. Due to leakages of condensation into the crankcase lubricant and also to leakages or pumping of oil by the piston-rings, the original mixtures were never constant; consequently much difficulty was experienced through the lubricating mixture "livering" up, with a consequent failure of the lubrication, burning out of bearings and repairs due to these causes. Eventually the type was discontinued.

The small high-speed engine of many makes, used extensively before the general introduction of the turbine, was lubricated with oil put into the crankcase. Water leakages caused serious emulsion troubles and many improvements were made in the type before these difficulties were overcome. The engines were never free from trouble from a lubrication standpoint until the oil was run out of the crankcase and through filters and separators. The Willans and similarly lubricated English types all experienced troubles due to the dilution of the lubricant by water and the salts carried by the various waters. The types that have prevailed are those in which provision was made to drain the oil from the crankcase into an outside system of filters, settling tanks, heaters and coolers as conditions required.

All modern steam powerplants, all hydroelectric plants and all of the large gas-engine powerplants throughout the world, have adopted as standard, the principle of treating the oil outside of the engine. This treatment includes removing the dirt and the water, cooling the oil

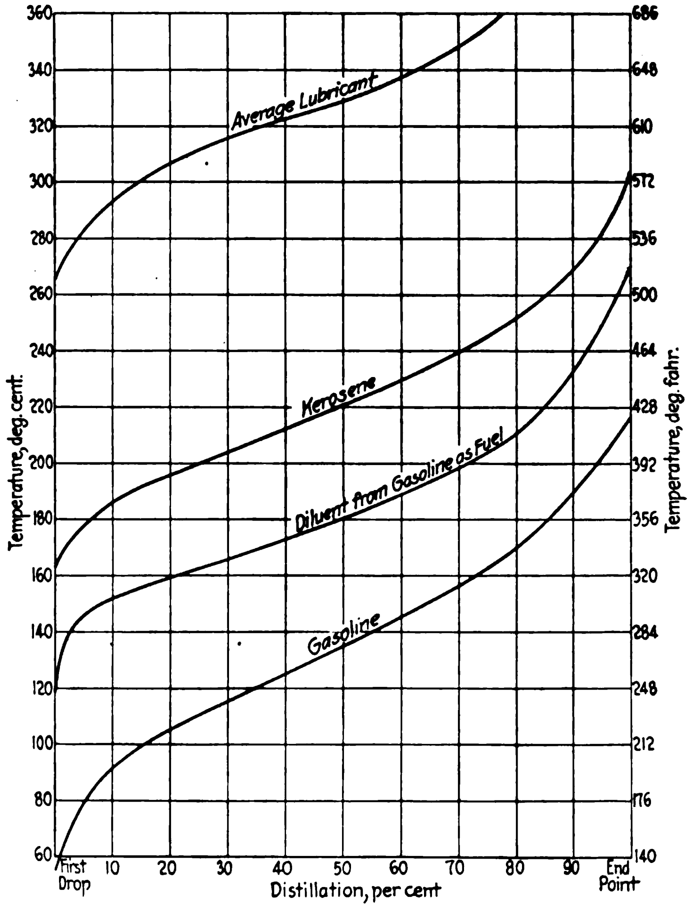

Fig. 8—Distillation Curves of the Average Lubricating Oil, Kerosene, the Diluent Removed from the Lubricating Oil and Gasoline

by water or air or large-capacity storage tanks, and thus securing lubrication with clean, cool oil from which all contamination has been removed.

The development of the steam turbine was marked with about the same cycle of events, but with more disastrous effect, than has transpired in the automobile engine so far as lubrication and dilution are concerned. The first turbines had small reservoirs in the base from which the lubricating oil was circulated fiercely in small quantities. This oil was subjected to changes in temperature and the diluent was water. The general effect was to thicken the oil. The pipes stopped up, the bearings burned out, the blades engaged and the entire machine, in many instances, was wrecked. At the present time, however, as shown by the installation drawings of a modern steam turbine, especially of the marine type, the turbine itself occupies an inconspicuous part, while the drawing is devoted mainly to the details of the lubricating system with its coolers, filters, settling tanks, strainers, pumps, traps and other refinements. Oils have not changed radically but turbine lubrication methods have and now oil is given a better chance to perform the services for which it was intended. As a result, the turbine today is one of the most efficient of power generators and does its work with a minimum of upkeep and depreciation.

The development of lubrication of the Diesel engine was along almost similar lines. In America the first Diesel engines were lubricated by the splash crankcase system. Water and oil were employed. The heat of the engine evaporated the water and the mixture then became too rich in oil with bad "livery" emulsions which, of course, failed to lubricate the bearings. The original American type of Diesel engine is now extinct.

Today, in the installation plan of a modern Diesel engine, the coolers, heaters, filters, separation tanks and pumps make an installation by themselves. The lubrication of the original Diesel engine presented a serious problem. The modern Diesel operates most effectively on any number of good oils and the engine itself is one of the most efficient of all of the prime-movers. The cost of its lubrication is small and the upkeep of parts which have to be lubricated is extremely little.

In considering the lubrication of the automotive engine it is interesting to note that we have developed from the 81 per cent splash systems and 19 per cent pressure systems in the cars of 1911 to the better proportion, in the 1920 cars, of 43.52 per cent splash-pressure, 33.33 per cent pressure and 23.15 per cent splash systems.[11] However, fundamentally we are still lubricating the automobile engine, as in the original designs, with oil placed in the crankcase. In these systems, the oil circulates within the engine and carbon from the burned fuel and oil with dust and dirt from the road are retained by the oil. Particularly is this true of tractors. There are many instances in which tractor piston-rings and engine bearings have been destroyed in a few days due to dust acquired by the lubricant. In addition to this dust and dirt there are also the iron and metal particles, arising from the wear that has taken place. All of the foreign substances help to break-down the oil and wear out the bearing surfaces. The mechanical development of the lubrication of the automotive type of internal-combustion engine has not gone far beyond the original stages. In the meantime the fuel has changed radically, and the fact has been established that the fuel finds its way into the crankcase and mixes with the oil, thus preventing lubrication.

Prof. C. E. Lucke in a paper entitled Rising Importance of the Oil-Injection Type of Internal-Combustion Engine[12] makes the statement that the present engine is no longer commercial unless the leakages of fuel are stopped and the engine is properly lubricated. With the present engines and fuel there is a direct loss of lubricant due to the more frequent drainages of the thinned crankcase lubricant. These drainings are at least twice as often as they were 5 years ago and this represents millions of gallons of oil per season. There is a secondary loss due to the oil being thinner and working up by the rings at a more rapid rate. The use of the heavier oils in an attempt to reduce these losses has placed an extra first cost on the consumer as these oils are more expensive per gallon to make and to buy. The dilution condition has easily doubled the internal-combustion engine lubricating bill of the Nation without giving an adequate return in better lubricated engines. The use of the heavy oils also increases the gasoline consumption, due to the higher engine friction, and the engines have to be cleaned from the carbon from these heavier oils at more frequent intervals than in the past, two items of increased cost of considerable magnitude.

These losses are of small value as compared to the losses involved in upkeep repairs and adjustments necessitated by the lack of lubrication. Add to the above items the depreciation of equipment that, with the hardest service engines, offers almost the greatest sales resistance, and there are sufficient grounds from the technical, the economical, the operating and the commercial standpoints to demand a considerable change in the one feature necessary to put the engine back on its original plane of excellence as far as lubrication is concerned.

Solving the Lubrication Problem

The problem of dilution is caused through the mixing together of the heavy hydrocarbons of the lubricant with the higher boiling-point hydrocarbons from the fuel. These hydrocarbons can be separated by heat. In order that the separation can be carried on at temperatures considerably below the ordinary boiling-points of the lighter fractions, vacuum distillation must be used. It has been found that even the petroleum oils constituting the lubricants will distill without breaking down or having decomposition take place.

Another way of reducing the temperature necessary to distill the lighter portions from the heavier is by agitation while boiling, a familiar example being the determination of the flash-point of fuel oil in the Pensky-Martin closed tester, where the oil must be stirred when taking the flash to liberate the most volatile fractions at low temperatures. J. E. O'Neill in a paper entitled Fractional Distillation of Lubricating Oils,[13] states that his experiments have shown that by violently agitating the flask it was found that the light oil distilled off at a much lower temperature than it would have if not agitated. He also points out in the same paper that the introduction of steam or carbon dioxide accomplishes the same result. As a laboratory proposition the removal from the crankcase lubricating oil of all the diluent caused by the use of gasoline as fuel can be accomplished without difficulty. In the case of dilution where kerosene is used for fuel there is an overlapping of the end-point of the diluent and the initial points of the lubricant, and possibly but 90 per cent of the diluent can be removed continuously, which would require all kerosene-

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[11] See *Automotive Industries*, Feb. 17, 1921, p. 311.

[12] See *Mechanical Engineering*, October, 1921, p. 653.

[13] See *Journal of the American Society of Naval Engineers*, vol. 28, p. 465.

burning engines to operate with a lubricant containing 10 per cent dilution of a very heavy end of the fuel.

It has been established that used oil, with the diluent removed and the dirt and carbon taken out, is as good a lubricant for the internal-combustion engine as new oil. Evidence has been presented to prove that oil is much better after it has once been used in an internal combustion engine, then cleaned of foreign matter and the diluent removed, than when new.[14] It it known that the use of oil that has passed through the engine, and been cleaned of diluent and dirt, will produce less carbon with each successive recovery. This is an indication that the carbon-forming part of the oil is gradually burned out by the engine, and that the most carbon in the engine is produced by the newest oil.

Temperature of Oil in the Crankcase

The viscosity of all oils is reduced as the temperature increases. At a temperature of 150 deg. fahr. in the crankcase, the light, medium and heavy oils will all be of different viscosities. If the light oil can then be made to operate at a temperature of 130 deg. fahr., it will have slightly more viscosity than the medium oil at 150 deg. If the temperature can be lowered to 120 deg. the viscosity of the light oil will be equal to the viscosity of the heavy oil at 150 deg. fahr. It is possible, therefore, to have the equivalent of a heavy oil in the engine by using a light oil and controlling the temperature.

The viscosity of a diluted light-oil may be the same as the viscosity of the undiluted light-oil but heated to a high temperature. It is, therefore, necessary, after heating the oil and fuel mixture to remove the diluent, again to reduce the temperature of the clean oil to a point at least the same as the temperature at which it is taken out of the crankcase.

The Crankcase Oil Vac-Refiner

This system for automatically removing the fuel and water dilution from crankcase oil and filtering out the sediment, composed of carbon, sand, metallic particles and the like has conclusively demonstrated both in the laboratory and when attached to the engines of trucks, tractors and passenger cars, as illustrated in Fig. 9, that it is possible to correct all of the present difficulties in the lubrication of internal-combustion engines. The soundness of its basic principles and method of operation has been confirmed by the work of five different groups of engineers, working on the same problem without each other's knowledge or interchange of ideas, who now have combined their respective data and other information into one organization.

This new system of crankcase oil regeneration consists of four main units, (*a*) the heating element, (*b*) the filter, (*c*) the refiner proper, and (*d*) the cooler. The entire system is extremely simple, light in weight, and occupies about the same space as a vacuum fuel system. It does not interfere with the present lubricating system of the engine. It functions equally well with the splash and forced-feed systems. It can be readily installed on most every type of car, truck, or tractor.

Fig. 10 gives a plan of this system, showing the flow of the diluted oil from the crankcase to the heating element and on to the filter and refiner, from where the oil is discharged to the cooler and back to the crankcase. The force causing the circulation of oil through the system is obtained from the vacuum present in the intake-manifold, which ranges from 2 to 25 in. of mercury, according to the type and condition of engine, revolutions per minute, manifold design and other conditions. A tube extends from the top of the refiner to the intake-manifold through which the vacuum, or suction, is transferred to the system, and through which the vaporized diluent is drawn off and burnt in the cylinders as power.

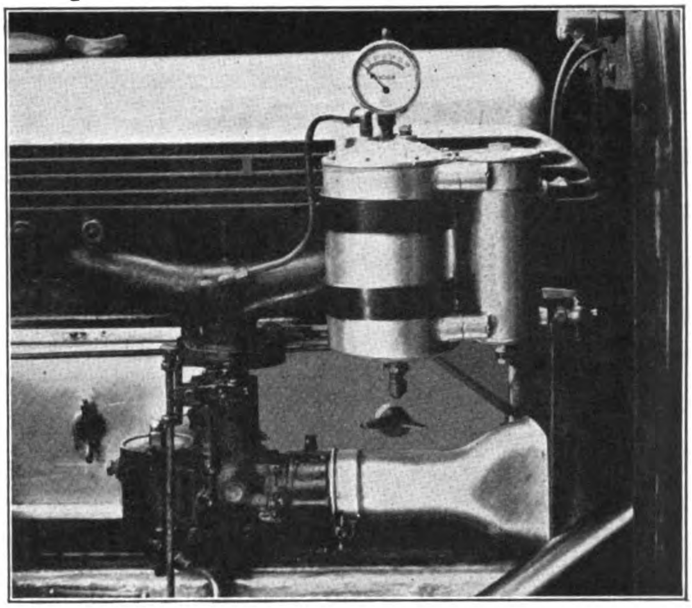

Fig. 9—Application to a Passenger-Car Engine of a System That Has Been Developed for Automatically Refining Crankcase Oil

The temperatures shown in the sketch are given as approximately 130 deg. fahr., for the oil from the crankcase to the heater; 400 deg. fahr. from the heater to the refiner and 350 deg. fahr. in the refiner, which temperature plus the vacuum and agitating effect, quickly removes the diluent. From the refiner at 250 deg. fahr. the oil passes to the cooler, with a further falling in temperature in the cooler until the oil will go to the crankcase at about 125 deg. fahr. The diluent, in the form of a fog, or gas, goes into the intake-manifold at tem-

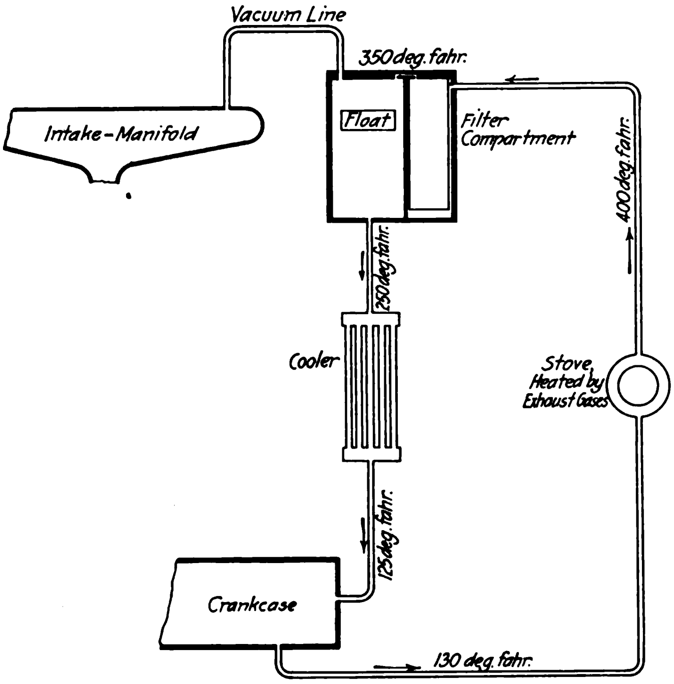

Fig. 10—Diagram Showing the Operation of the System

[14] See *National Petroleum News*, May 21, 1919, p. 20.

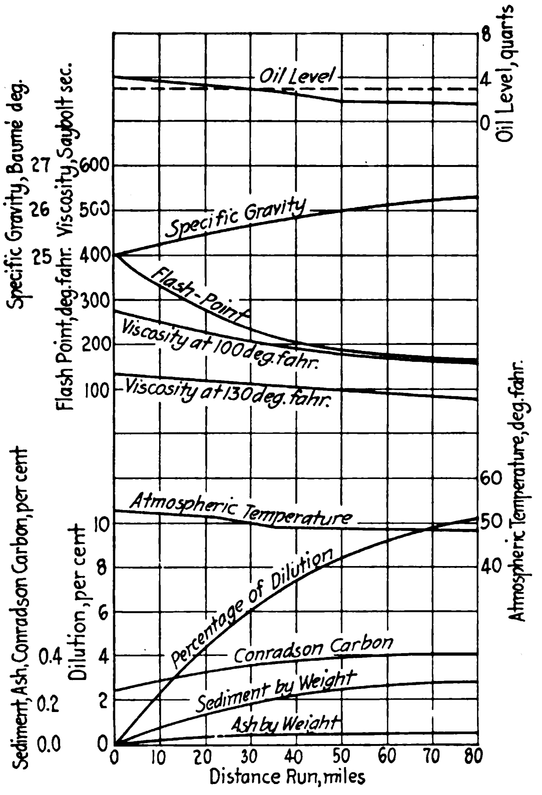

FIG. 11—CHART SHOWING THE CONDITION OF THE OIL IN A PASSENGER-CAR ENGINE THAT WAS NOT EQUIPPED WITH THE AUTOMATIC OIL REFINER

peratures up to 200 deg. fahr., according to the distance of the refiner from the intake-manifold.

Heat is taken from the exhaust by any one of several very efficient ways. There is sufficient heat in the exhaust under practically every condition of operation to allow the removal of most of the diluent from the oil. The heater, while in operation, is either filled with oil to the exclusion of all air, or is working with oil passing through the heater under the force of a vacuum. In this way charring of the oil is prevented. Heaters are cylindrical shells slipped over the exhaust-pipes, or coils of various designs, either machined or made from tubes or piping, inserted in the exhaust pipe or manifold. Heaters of several kinds have been cut open after thousands of miles of operation and have been found free from carbon deposits.

The refiner, which acts as a distillation flask in the removal of the fuel-content from the lubricating oil, is integral with the filter and settling system. The oil comes from the heater to the first filter and settling chamber where it passes through the screen or filter from where the cleaned oil is drawn to the still proper and deflected by a baffle to a thin film of heated oil. The combination of the heat and the great reduction in boiling-point produced by the vacuum, plus the agitation from the moving vehicle, causes a very rapid vaporization of the diluent. The diluent, in the form of a heated fog, then passes along the vacuum line into the intake-manifold and then to the cylinders where it is consumed as fuel. The bottom of the still is arranged as a second settling chamber for the collection of such sludge and dirt as passes the first settling chamber and the filter. Means are provided to clean out the accumulated dirt when necessary, quickly and easily. The still contains the float mechanism that actuates the air and vacuum valves. This part of the system is similar to that employed for the vacuum fuel-tanks, which is an item of value in considering service. There is the one feature of interest as influencing the wear of the only moving parts in the system. With the fuel vacuum-tanks the operating mechanism is mostly dry and occasionally covered with a sulphur powder from the gasoline fumes, while similar parts in this oil refining system are continually covered with oil. The perfectly lubricated parts should, therefore, outlast the engine to which the system is attached.

One of the most important elements in the system is the cooler. The cooler is placed where the air from the fan will dissipate the heat being thrown off from the oil. On engines where there is no fan for cooling, the cooler is placed near the flywheel.

COMPARATIVE RUNS WITH AND WITHOUT THE REFINER

Fig. 11, which shows the characteristics of many comparative tests, indicates the condition of the oil at various periods in the engine of a car operated in city traffic. The oil has become diluted to an extent of 10.2 per cent in 89 miles. The oil was drained and the engine filled with new oil of the same make and grade, some of the former 10.2 per cent diluted oil remaining in the engine and diluting the new oil about 3 per cent as shown in Fig. 12. After 36 miles the dilution was 1.5 per cent and at 74 miles the dilution was under 1 per cent, where it remained. The viscosity of the oil at the end of the 98-mile run was within 5 sec. viscosity at 100 deg. fahr., of the new oil. The viscosity remained the same in this engine during subsequent runs to a total of 2180 miles. Occasionally during these runs, which were in city traffic, 1 pint of raw gasoline would be put into the crankcase. This would be taken out of the oil inside of 30 miles. One pint of raw kerosene would be taken out of the oil in about 60 miles. The sludge removed from the bottom of the reclaimer tank generally contains a characteristic sediment of 75 per cent of oil, 12 per cent of carbon and 13 per cent of an ash composed of silica and metal. The ash and the carbon will vary with the nature of the roads and the character of the service.

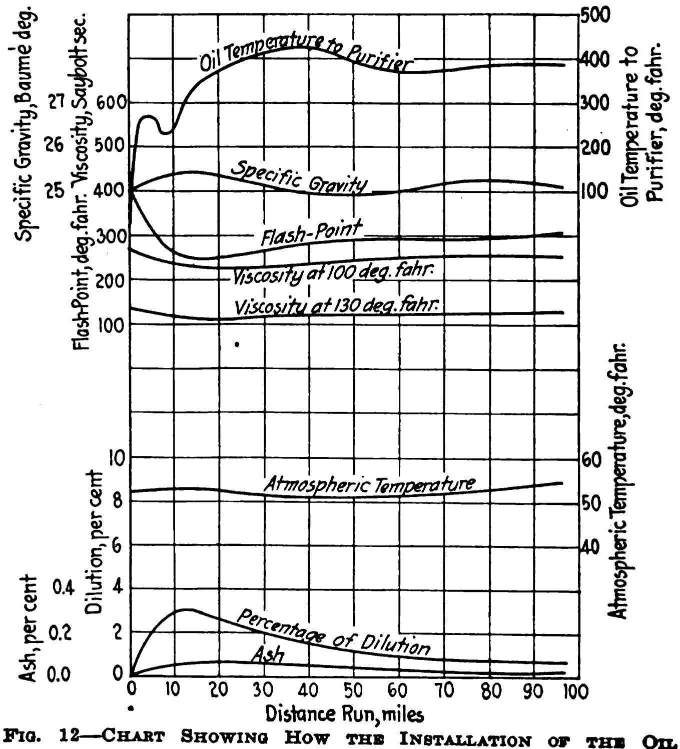

FIG. 12—CHART SHOWING HOW THE INSTALLATION OF THE OIL REFINER MATERIALLY REDUCED THE PERCENTAGE OF CRANKCASE DILUTION

While there will probably be minor difficulties to be overcome in the matter of installation and workout, this system solves the worst angle of the heavy fuel problem, namely, crankcase dilution and its effect upon the lubrication of the engine, and leaves the problem of the proper carburetion or the prevention of flooding of the manifold and cylinders by raw gasoline to be solved at a later date, either through changes in the fuels supplied for use in the future in automotive engines or by the perfection of special mechanical devices.

THE DISCUSSION

GEORGE L. MCCAIN:—How would you explain the effect of fuel made after the German and French Navy specifications, if after a certain time the viscosity of the oil in the engine were above that at the time the oil was put into the engine?

WILLIAM F. PARISH:—If an engine is run for some time on artificial gas, the viscosity of the engine oil always increases. This is due to the distilling off of the lighter fractions of the oil. In former days when we had a light fuel similar to those made under the above-mentioned specifications, we used light lubricants; the tendency was for the oil to increase slightly in viscosity during use. Where there is no dilution, there is generally an increase in the body of the oil.

GEORGE MEREDITH: — In E. W. Roberts' book[15] it is stated that the claim of dilution due to the incorporating of the lubricating oil with gasoline in the two-cylinder two-cycle engines, is erroneous. What condition would we find?

MR. PARISH:—I do not know where he got his data. He may have found the effect stated in some experiments.

GEORGE A. BREEZE:—What is the temperature loss between the heater and the refiner?

MR. PARISH: — We have had thermometers located right at the heater and at the refiner. There is possibly a loss of 50 or 75 deg. fahr. in the line between those two points.

MR. BREEZE:—Have you made check readings on temperature in the exhaust pipe?

MR. PARISH:—That was covered by O. C. Berry's paper on Manifold Vaporization and Exhaust-Gas Temperatures,[16] in which he shows the range of maximum temperatures in exhaust manifolds. Temperatures are as low as 450 and as high as 1400 deg. fahr.

MR. BREEZE:—What temperature do you find it necessary to maintain to get distillation?

MR. PARISH:—We should have a temperature at the refiner of 350 deg. fahr. when gasoline is used as fuel. To that would be added the effect of the variables, vacuum and agitation, which would be equal to 100 deg. or 150 deg. fahr. There are too many inconsistencies in the practical workout to obtain 100-per cent results. The temperatures and vacuum change as the speeds and loads increase or decrease. Refining the diluted lubricating oil in apparatus on an engine in service is a flexible sort of process that does not work constantly like the speedometers. Of the many variables there is a basic variation or difference in the viscosity of the oil due to the heat in the crankcase. The speed of oil through the system is governed largely by the temperature of oil in the crankcase and this speed causes changes in the oil temperature.

We build up the heat in the oil to 400 or 500 deg. fahr. for kerosene-burning engines. This heat must be liberated in some way before the oil gets back to the engine, and that is done while holding the oil in the cooler during the time the instrument is taking in the fresh charge of diluted oil.

MR. BREEZE:—Do you find that you get your distillation at temperatures less than that of the end-point?

MR. PARISH:—Yes. It frequently is supposed that one must have a temperature at least as high as that of the end-point, but we find that it is not necessary to have so much heat, the vacuum and constant agitation making considerable difference in the problem.

A MEMBER:—Does the refining system have a tendency to increase the viscosity of the oil?

MR. PARISH:—We have not been able to eliminate 100 per cent of the dilution in any of the practical work we have done so far. We think that when we do, the oil will increase in viscosity through the working of the system.

MR. MCCAIN:—If a certain amount of dirt is drawn through the system, does that separate out in the filter and can it be drawn away from there?

MR. PARISH:—Dirt is taken out of the oil in the main filter, which is easily cleaned, and there are two drain-cocks for sludge that passes the filter and settles in the spaces provided; the dirt will be road dust and carbon. If you lubricate properly there should not be much metal in the oil.

L. A. CHAMINADE:—Some of the oils we get at present, such as those made from California crudes and other crudes that have not been properly refined, are emulsifying oils. Due to the products of combustion that pass the pistons and rings, particularly in winter, such an oil soon becomes an emulson. How does an oil of this type affect this system? How often should the system be cleaned out?

MR. PARISH: — Water in the oil aids the distillation process very materially. The steam bubbling through the oil carries off the light stuff, all going into the engine for power.

MR. MCCAIN: — How often would it be necessary to clean the filter?

MR. PARISH:—Generally, in city driving, one will need to clean the filter every 1000 miles, which can be done very quickly. For tractors in dusty soil, cleaning may have to be done every time oil is put in the engine or water in the air-cleaner. This system holds about 100 cc. of oil (6.1 cu. in.) as a settling space for dirt. Using oil for the collection of dirt, so it can be drained from the two drain-cocks, is a very nice point, because there is always oil containing some dirt that can be taken away from the engine. When there is water in the oil, this aids the distillation and the diluent will distil off about 100 deg. fahr. earlier. In coming off, the water will precede the diluent to the cylinders, being more rapid in travel. With oil containing a compounding material, which emulsifies upon working with water, this refining system would clean the oil and allow the water to go back into the engine cylinders.

MR. BREEZE:—With your system it seems the fuel has slightly different characteristics than one would expect from it. That may be due to the cracking of the fuel in the heater.

MR. PARISH:—In every paper I have read on the subject of dilution it is stated that the dilution comes from three causes: (*a*) the loss of fuel during the compression cycle, (*b*) draining of fuel that has accumulated in the intake manifold and (*c*) some cracking or decomposition of the lubricating oil. In assembling the demonstration apparatus I tested some oil made here in Detroit of 230 Saybolt-sec. viscosity at 100 deg. fahr., running it through the refiner for 30 min. at from 500 to 600 deg. fahr. or

[15] See The Gas Engine Handbook, by E. W. Roberts, p. 236.
[16] See THE JOURNAL, March, 1922, p. 171.

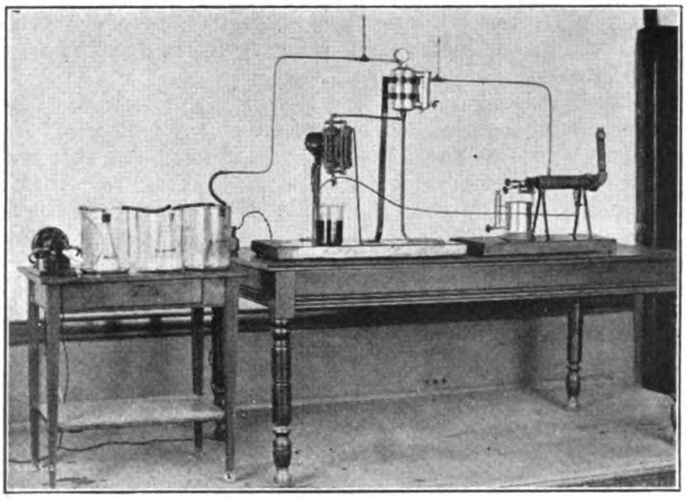

FIG. 13—SET-UP OF APPARATUS TO DEMONSTRATE THE PRINCIPLE ON WHICH THE CRANKCASE OIL VAC-REFINER OPERATES

1000 cc. (61 cu. in.) of new oil with no diluent added, the gravity of the new oil being 23 deg. Baumé. There was a recovery of 1 per cent of clean bright light oil that had a specific gravity of 43 deg. Baumé.

J. R. WADIA:—If under certain conditions in this Country the dilution of oil be somewhere about 10 per cent, will it be more or less in the same engine in India?

MR. PARISH:—It will be much less. The Rangoon and Burma fuels used in India are much more volatile fuels than those we have in this Country. Also, heavier lubricating oil is used on account of the trade and climate. The "light" oil in India and Europe is always as heavy as the "heavy oil" in this Country. It would be an advantage to use a lighter-bodied oil with the more volatile fuels. The conditions are overbalanced in India, the same as they are in England and France. Where there are light fuels and heavy lubricants, the heavy oils last longer, which is the main excuse for their existence. With volatile fuels, the lighter bodied oils should be used for the efficient action of the engine.

MR. WADIA:—Are the oils sold in India under the same brands as the oils sold in this Country the same in quality?

MR. PARISH:—The only way to tell is to analyze the oils. No one on the outside would know without analyzing the oils. As a matter of fact, if a large company were marketing lubricating oil and the brands were well established, there would be no reason against maintaining the brand and meeting the demand for the body by varying it.

G. J. LUX:—How would this refining system handle a compounded oil?

MR. PARISH:—Such an oil might stick up inside the pipe a little, and perhaps the inside of the refiner would look about as bad as the inside of the engine crankcase when using the same compounded oil.

TABLE 7—DISTILLATION TEST OF A GASOLINE-OIL MIXTURE

Time	Oil Circulation Through Refiner					Temperature, Deg. Fahr.				
	Intake-Manifold Vacuum,	Time to Fill		Time to Drain		Heat of		Drain from Cooler	Oil in "Crankcase" Beaker, cc.	Total Oil Shown in "Crankcase" Beaker, cc.
	In.	Min.	Sec.	Min.	Sec.	Refiner	Vapor			
9:55	11.00	..	42	......	9	260	100	110a	74	1,200a
10:00	10.50a	..	45	......	10	370	190	120	78	1,100a
10:05	10.25	1	35	......	13	400	240	140	84	1,000
10:10	9.00	1	40	......	15	490	220	150	90	925
10:15	8.00	1a	45a	......	14a	470	208	145a	94a	900a
10:20	12.00	1	55	......	$11\frac{3}{5}$	560	210	142	98	875
10:25	9.00	1	37	......	$16\frac{1}{5}$	470	265	163	99a	750
10:30	7.00	2	12	......	$12\frac{2}{5}$	490	250	160	100	750

a—Taken from other tests.

K. K. HOAGG:—From a practical standpoint, how often would one need to change the oil in the crankcase, with this refining system in constant operation?

MR. PARISH:—One could run on the same oil all of one season. Why drain the oils when they retain their body and are clean?

L. C. FISK:—Have you any practical figures on the better fuel economy? You are reusing much of the fuel that would otherwise be thrown away. Do you know the percentage of economy you obtain?

MR. PARISH:—The figures I have are not accurate enough to make a statement as to fuel savings. There are many variables and it is not wise at this time to give figures. When carbureters are set for lean mixtures without the refiner, the addition of the refiner makes the mixture rich.

In regard to the demonstration of the Gross crankcase-oil refiner, the apparatus is shown in Fig. 13. The manifold vacuum was represented by using a motor-driven vacuum-pump. The diluent from the mixed oil and fuel was drawn from the refiner through three condensing bottles that were packed in ice. The heated oil was obtained from a type of heating element developed for the use of this refiner system, for the Fordson tractor on which the apparatus is installed as shown in Fig. 14. Heat was supplied from a gasoline blow-torch. A thermometer attached to the refiner showed the temperature of the fuel and oil mixture at that point. Another thermometer was placed in the vapor line to show the tem-

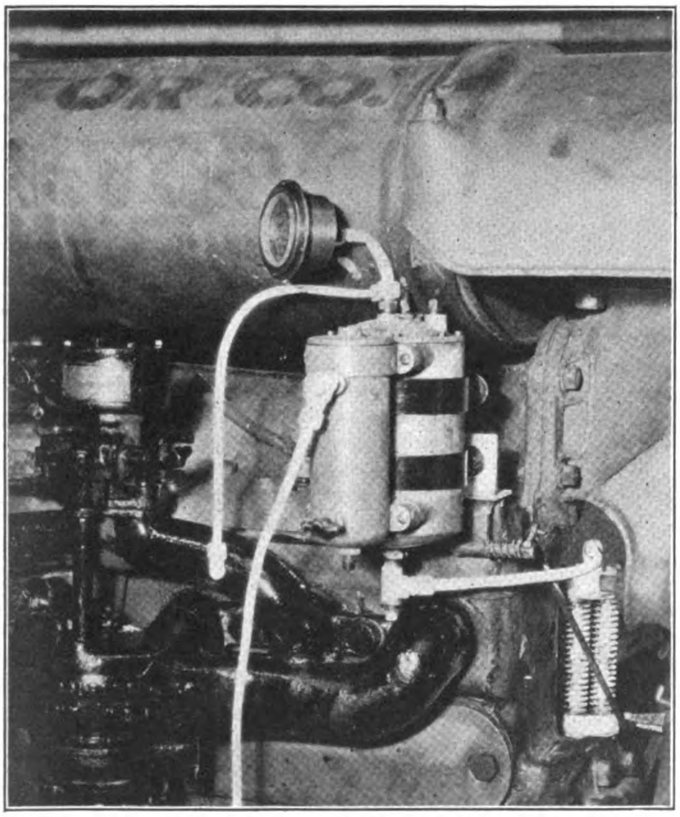

FIG. 14—THE CRANKCASE OIL VAC-REFINER APPLIED TO A TRACTOR ENGINE

perature of the diluent, or distilled fuel, as it left the refiner and, under engine conditions, would have gone into the engine for fuel. A thermometer was placed also in the graduated beaker, which held the mixed oil and fuel. This beaker represented the crankcase. When the refined oil left the refiner it went through a cooler, back of which was a small electric-motor fan for cooling.

A mixture was made of 750 cc. of Texaco Motor Oil Light and 750 cc. of Red Crown gasoline. This was put into the beaker representing the crankcase. The refiner system required about 300 cc. of the oil to fill the heater, filter, refiner and pipes. After the system drained through the cooler, the number of cubic centimeters in the beaker was measured, the receding amount being caused by the fuel distilling off and being caught in the bottles. Attention was called to the exhaust from the vacuum-pump, which was made up of gas that would not condense. In former experiments this consisted of about 10 per cent of the amount of diluent that had been put into the oil.

The results of the demonstration made at this meeting on a mixture of 50 per cent Texaco motor oil and 50 per cent Standard Oil Red Crown gasoline are shown in Table 7. Upon completion of the demonstration, the diluent in flask No. 1 was 260.0 cc.; that in flask No. 2, 28.5 cc.; and that in flask No. 3, 6.0 cc. The total, 294.5 cc., represents 40 per cent of the original diluent off and condensed in 30 min.

WHAT GERMANY CAN PAY

WHILE elaborate and apparently convincing arguments can be built upon the present economic facts to prove the contention that Germany will not be able eventually to meet the schedule of payments under the London settlement, an examination of the data upon which all such arguments have been based raises the questions whether on the one hand Germany's permanent losses have not been considerably overestimated and whether on the other hand sufficient consideration has been given to certain intangible, but none the less important, factors in Germany's future productive capacity.

The loss of political control over rich natural resources, upon which other important industries depend, does not necessarily mean the complete loss of economic control. It is true that a change in political control may in part divert the flow of natural resources from their normal channels, but there is much recent evidence that economic forces are overcoming political barriers and it now appears likely that a larger part of the prewar resources of Germany will find their way back to German industries than was at first supposed.

There is also evidence that the economic power of substitution is already working in Germany to supply, from within her boundaries or from neutral countries, materials to take the place of those lost under the treaty of peace. It further appears that insufficient importance has been given to the great possibilities of the development of neighboring territories by Germany and of her ability to establish an even larger German economic unit than existed before the war.

Perhaps most important of all, any estimate of Germany's economic position that considers her saving capacity permanently reduced proportionately to the temporary losses of her natural resources overlooks the fact that one of the most important elements in Germany's productive capacity could not have been permanently reduced by the war, namely, her genius for applying science to business and industry. Indeed, there appears evidence that this genius is not only left intact, but has been increased greatly by the war. Moreover, her extraordinary capacity for organization, which before the war was partially diverted to military operations, can now be directed exclusively to the organization of industry and commerce, not only to the commerce of Germany but also in part to that of the surrounding countries, including Russia.

In reviewing the principal items that condition her future capacity it is possible to believe that Germany will eventually regain an important part of her prewar saving capacity. In any case we are justified in recognizing that there has not yet been produced convincing evidence that Germany will not eventually develop a very great saving capacity.

Such capacity, however, must rest upon the condition that what is left of the German economic unit can become readjusted and operated with an approach to the degree of efficiency that prevailed before the war. There is much to indicate that substantial readjustment is possible provided crushing public financial burdens do not continue to dislocate industry and commerce.

The outstanding aspects of the German problem can be summarized as follows:

(1) The present German problem undoubtedly constitutes one of the largest obstacles to the recovery of trade throughout the world. Indeed, wherever one turns in the maze of European economic and financial problems one is squarely confronted with the German problem

(2) A very large part of the present German problem, so far as it relates to general business conditions and uncertainties in foreign trade, arises directly or indirectly from the deficit financing occasioned by the unbalanced budget

(3) After allowing for all seemingly practicable retrenchment in expenditure and for practical increases in revenue, it appears clear that the balancing of the budget has been impossible in the face of the total of currently accruing charges in the very nature of things, therefore, it is to the interest of the world at large and specifically to the interests of the Allies that there be made whatever temporary arrangements are necessary to enable the budget to be balanced

Once the budget is balanced Germany will be in a position to accomplish the readjustment of her production machine necessary to increase her capacity to make the fullest possible reparations payment.

In the recent tentative readjustment of indemnity payments advancement has been made in the recognition of the real problem. Whether the full temporary relief necessary to permit Germany to recover her maximum paying capacity is to come in the form of a combination of internal and external loans, as a partial moratorium, or through a general readjustment of international obligations is a question upon which much progress is yet to be made. Any acceptable plan must, of course, comprehend the clearly recognizable needs and rights of the Allied Governments, some of which also face tremendous fiscal problems.—*Commerce Monthly.*

HOT-SPOT METHOD OF HEAVY-FUEL PREPARATION

(*Concluded from page* 32)

carbureter opening and additional contact with the exhaust-manifold a little farther along, usually at a point of division of the fore-and aft reaches. Apparently the second "spot" catches some of the particles that elude the first one and gives a more complete and steady evaporation.

One important requirement of a successful fuel-charge heater is that it should warm-up and get underway quickly. The walls should be thin, and, if cast, should be lightly ribbed on the exhaust side. Aluminum combines low specific heat and rapid conduction and is a very suitable material for a cast hot-spot heating surface, if there are no shielded parts that become heated to such a degree as to melt.

Preventing Liquid Fuel Reaching the Cylinder

It is recognized generally that it is desirable to prevent liquid fuel from reaching the cylinder and it has been claimed for many designs that they have this action. We have tried models of a number of them and have found that few impede the travel of liquid fuel to the cylinders in even a slight degree. With transversely ribbed elbows, for instance, the fuel drops are caught off the tips of the ribs by the air eddies and snatched through the elbow as if no ribs were present. This, of course, is with air velocities above 70 ft. per sec., and part of the lively

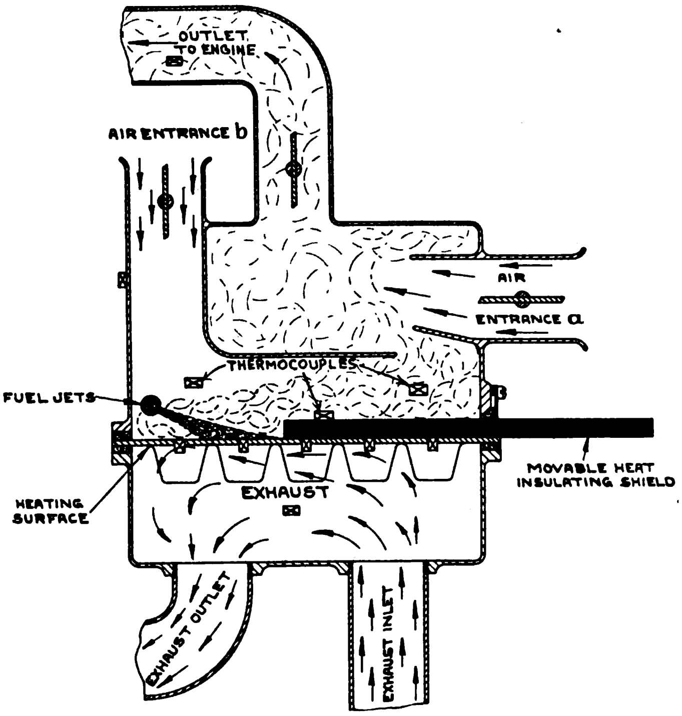

Fig. 8—Diagram Showing the Construction of the Experimental Fuel Heater Used

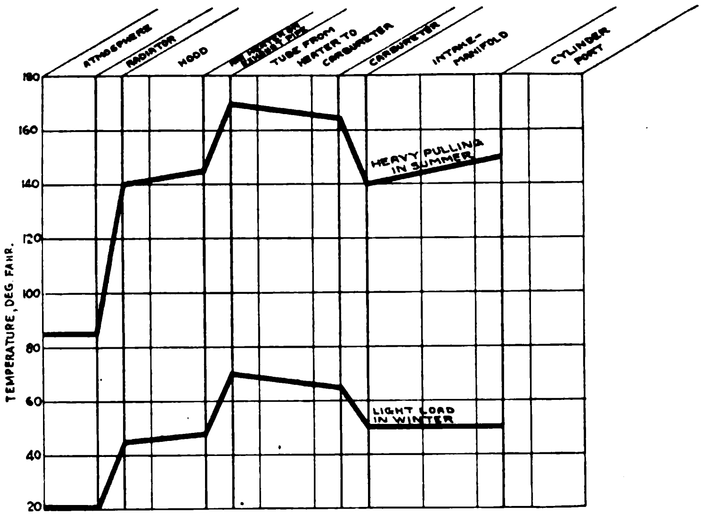

Fig. 9—Curves Showing the Range of the Natural Temperature Variation of the Air Charge

action naturally is due to the spheroidal condition already described.

We have used centrifugal force, surface tension and the force of gravity to separate the unvaporized drops. Careful combination of all seems to be required to achieve complete separation. A partial separation, which should be very effective at low engine speeds, can be obtained by abruptly increasing the manifold area above and beyond the hot-spot. This would allow the heavy drops to settle down and again be hurled against the heating surface. The separation and recirculation would obviously be beneficial to the action of either Figs. 4 or 5, but *the heat supply must be adequate or the fuel will not reach the engine*, with an actually functioning liquid-fuel separating device.

In our work with various types of fuel heater, we have experienced a slight but important change in viewpoint, perhaps a keener realization of the truth, in the problem of supplying fuel to internal-combustion engines. This I would like to communicate to the Society. After watching the fuel, in an accumulation equal to many cylinder charges, bubbling, splashing, sometimes lying quiescent on the heating surface of glass-walled hot-spots, and sometimes swept through in a high-velocity spray, one fact stands out: the *rate of fuel-feed from the manifold to the cylinder* primarily governs the conditions of combustion, and the rate of fuel-feed to the manifold is an indirect rather than a direct controlling factor, as regards the mixture proportion of the charge actually used by the engine.

This point of view, we believe, is the proper one from which to consider the problem of the efficient use of fuel in our engines of today.

The Mechanism of Lubrication[1]

By Robert E. Wilson[2] and Daniel P. Barnard 4th[3]

Annual Meeting Paper *Illustrated with* Charts and Diagrams

THE authors state that the coefficient of friction between two rubbing surfaces is influenced by a very large number of variables, the most important being, in the case of an oiled journal, the nature and the shape of the surfaces, their smoothness, the clearance between the journal and the bearing, the viscosity of the oil, the "film-forming" tendency or "oiliness" of the oil, the speed of rubbing, the pressure on the bearing, the method of supplying the lubricant and the temperature. The primary object of the paper is to present the best available data regarding the fundamental mechanism of lubrication so as to afford a basis for predicting the precise effect of these different variables under any specified conditions.

Definitions of the terms used are given and the laws of fluid-film lubrication are discussed, theoretical curves for "ideal" bearings being treated at length. The application of the recommended method of plotting to data in the literature of the subject is described, the thought then including consecutively oiling method and oiliness-factor effects; the effect of variable clearance; bearing metal and clearance effects on the critical point of film rupture; oiliness and miscellaneous effects. A summary is given of the essential conclusions and a description of the method of their application to practical problems.

THERE is no need to discuss or stress the importance of lubrication and a more thorough-going fundamental knowledge of its mechanism before an audience of engineers. The wide discrepancy between the indicated horsepower of the engine and the power delivered to the rear wheels of an automobile calls attention more forcefully to the lack of proper lubrication and a low coefficient of friction than could an entire paper devoted to this phase of the subject.

The mechanism of lubrication admittedly is a very complicated subject. The coefficient of friction between two rubbing surfaces is influenced by a large number of variables whose effects are separable only with difficulty. The most important variables in the case of an oiled journal are the nature and shape of the surfaces, their smoothness, the clearance between the journal and the bearing, the viscosity of the oil, the "film-forming" tendency or "oiliness" of the oil, the speed of rubbing, the pressure on the bearing, the method of supplying the lubricant and the temperature.

It is difficult to separate and determine quantitatively the effect of any one of these variables. For example, in comparing two surfaces it is never possible to be sure how much of the difference is due to the kind of metal used and how much to the degree of smoothness attained in their preparation. In comparing two lubricants, the absence of a definite measure for the film-forming tendency of an oil makes it difficult to know how much of the difference in its behavior is due to variations in viscosity and how much to this rather indefinite oiliness factor. In comparing bearings the variations can be attributed to differences of clearance and of methods of feeding the oil, or to the condition of, and pressure on the surfaces. Similarly, it is very difficult to come to definite conclusions as to the effect of even such simply and readily measurable variables as load and speed. For example, some results seem to show that increasing the load or the speed tends to increase the friction coefficient, while other experiments indicate the opposite. Such discrepancies generally are due primarily to failure to distinguish between the regions of partial lubrication and complete fluid-film or perfect lubrication, the fundamental laws of which are different. If we are to make progress in designing bearings and improving lubricating oils, it is essential to understand the fundamental mechanism of lubrication and to be able to predict the precise effect of the different variables under specified conditions.

Definitions

Speed is the relative motion of the bearing surfaces in feet per minute. This is obviously equal to $\pi N D/12$, where N is the number of revolutions per minute and D the diameter of the shaft in inches.

The bearing pressure, p, is defined as the total load on the bearing (figured vectorially if more than one load is acting in different directions) divided by the projected bearing-area in square inches. In other words, for a cylindrical bearing $p = l/L\ D$, where l equals the total load and L equals the length of the bearing. This is obviously a fictitious value, since the pressure-distribution curve varies considerably in different portions of the bearing, but it serves as a convenient basis for comparison.

The *clearance, c*, as used throughout this paper, is the difference between the diameters of the journal and the bearing. It is therefore twice the radial clearance used by some writers, frequently without clear distinction.

The *frictional resistance, F*, is measured by the amount of force that must be applied in a direction tangent to one of the bearing surfaces to cause it to move past the other surface at the desired speed.

The *coefficient of friction, f*, is equal to this frictional resistance divided by the load; it is that portion of the force pressing the surfaces together that is required to move the surfaces relative to one another. For flat surfaces it can be visualized most clearly as the tangent of the angle at which one loaded surface would just slide down the other at the desired speed under the action of gravity alone. This figure is the best single measure of the efficiency of a given bearing. It does not, however, represent in any sense the ratio between the power transmitted by a journal and that which is dissipated in the bearing.

The *lost power* is the product of the frictional force times the speed. The term "lost work" is sometimes used

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[1] This represents in amplified and slightly modified form the first part of the paper presented under this title at the Annual Meeting of the Society, in January 1922. The second part will appear in the August issue of The Journal under the title The Measurement of the Property of Oiliness.

[2] Director of the research laboratory of applied chemistry, Massachusetts Institute of Technology, Cambridge, Mass.

[3] Research associate, Massachusetts Institute of Technology, Cambridge, Mass.

loosely in the same connection, but it is generally the lost work per unit of time or the lost power that is really of interest.

The *carrying power* of the film will be discussed more in detail later. For the present it merely will be pointed out that a rotating bearing with an adequate supply of oil tends to drag a fluid film of lubricant between the bearing surfaces. The thickness of this film increases with the speed of rotation and the viscosity of the lubricant but decreases with the load. The carrying power of a bearing is the amount of load that can be carried by a given bearing operating under specified conditions without decreasing the thickness of the fluid film below some definite limit where practical experience shows that there is danger of having actual metal-to-metal contact and abrasion. There is considerable disagreement as to the magnitude of this limiting thickness, which undoubtedly varies with the smoothness of the surfaces.

The *"oiliness"* of a lubricant, for the purpose of this article, is defined as that property by virtue of which one fluid gives lower friction coefficients at low speeds or high loads than another fluid of the same viscosity. It is possessed in varying degrees by different lubricants, ordinary mineral oils generally being somewhat deficient in this respect as compared with most animal and vegetable oils. It is tacitly assumed throughout this paper that the oiliness of a lubricant is connected in some way with the adsorption of some constituent in the oil by the metal surface and this assumption is believed to be justified fully by data to be presented in the subsequent paper on The Measurement of the Property of Oiliness.

Fundamental Laws of Fluid-Film Lubrication

Since nearly all well designed journal bearings operate under conditions of perfect fluid-film lubrication during the major portion of their operating lives, no consideration of the mechanism of lubrication would be complete without a detailed discussion of the fundamental laws of fluid-film lubrication and their application to practical problems in the design and operation of bearings. As already indicated, the coefficient of friction is a function of a large number of variables and the comparison of results obtained by different experimenters under varying conditions therefore becomes extremely difficult; in fact, it is so difficult that little has been attempted along these lines. The potential value of such a correlation, however, is great enough to justify the attempt.

The best methods of comparing such results will be graphical in nature, rather than involving the application of a complicated equation with the necessarily large number of variables. Certain graphical methods have been employed in foreign publications, particularly those by Lasche.[4] His results are presented in the form of a series of three-dimensional diagrams in which the coefficient of friction is used as the ordinate in all the diagrams and different pairs of the four principal variables, viscosity, speed, load and clearance, are used for the other two coordinates in a given diagram. This requires six three-dimensional diagrams to cover the subject fully, and they are very difficult to construct and use. It seems desirable, therefore, to make use of another and much more convenient expedient.

A recent publication by Wilson, McAdams and Seltzer[4] has shown how the very complicated case of the variations in the friction coefficient f in Fanning's formula with velocity, viscosity, density, pipe diameter and roughness, can be simplified greatly by taking advantage of the fact that f is a factor "without dimensions" and therefore cannot be an *independent* function of the four variables, v, z, s and D, but only of their *combination* in the form $D\ v\ s/z$. By plotting observed values of f against varying values of this ratio, it is possible to bring out very clearly the nature of this functional relationship and the effect of the fifth variable, the degree of roughness, on the values of f. Similarly, for the case of lubrication, Hersey[4] has shown by dimensional reasoning that, while f, the coefficient of friction, tends to increase approximately in proportion to the speed and the viscosity of the lubricant and in inverse proportion to the load or pressure on the bearing, it is also a factor without dimensions, and hence it cannot be an independent function of these variables, but of a combination of them in the form $z\ N\ D\ L/l$, or its equivalent $z\ N/p$ where

z = Viscosity of the lubricant at the operating temperature
L = Length of the bearing
D = Diameter of the journal
l = Load on the bearing
N = Revolutions per minute
p = Pressure on the bearing

It is, of course, also a function of other factors without dimensions, such as the ratio of the clearance to the diameter, the ratio of the length to the diameter, the smoothness of the bearing, the method of oiling and the like, and possibly of some "oiliness" factor possessed by the lubricant. By plotting the values of f, obtained by various investigators under different conditions against $z\ N/p$, it should be possible to tell just how important these other variables are and whether or not they can be ignored within the limits of good bearing design.

In spite of the great potential value of this method for correlating the experimental results of different investigators, it apparently has never been made use of for this purpose. While the results of the dimensional reasoning can be accepted without question, it must be kept in mind that dimensional reasoning specifies only the existence and not the nature of the functional relationship between f and $z\ N/p$.

For most actual bearings with their many disturbing factors of variable clearance, surface roughness, end-leakage and the like, the calculation of the precise functional relationship is difficult if not impossible, and plotting experimental values of f against $z\ N/p$ is the only way to establish the characteristic curve for a given type of bearing. Once this curve is established, however, for a given bearing, dimensional reasoning *does* demand that the same curve represent all other bearings that are geometrically similar in their ratios of clearance and length to diameter, the nature of the surface and the like. It may aid in the interpretation of such curves to first consider quantitatively the behavior of an "ideal" bearing, and qualitatively the probable effect of certain disturbing factors in which all actual bearings differ from the ideal.

Theoretical Curves for Ideal Bearings

If an ideal bearing be defined as one in which the journal is perfectly centered in the bearing, with absolutely smooth bearing surfaces, no oil-grooves, no end-leakage and the clearance space filled with fluid, a little consideration indicates that, since the frictional resistance is entirely in the fluid film, it is determined solely by the force required to move the molecules of the fluid past one another. In accordance with Poseiulles' law for viscous flow, this force must be directly proportional to the

[4] See bibliography at end of paper.

speed of motion and to the viscosity of the lubricant and independent of the pressure.[5]

The coefficient of friction would therefore vary inversely as the pressure, and if the values of f were plotted against $z\ N/p$, they should all lie on a straight line passing through the origin as indicated by the full lines in Fig. 1. It also is obvious that the friction will be greater the smaller the clearance is. Indeed, it is readily possible to calculate the equation for the ideal line, using merely the laws of fluid friction, such calculation giving $f = 48 \times 10^{-9} \cdot D/c \cdot z\ N/p$, where c is the diametrical clearance. For a typical clearance ratio of 1/1000, $f = 0.000048\ z\ N/p$. The lines drawn in Fig. 1 were calculated for clearance ratios of 1/1000 and 1/250 respectively.

Actual and Ideal Bearing Differences

In regard to the probable effects of different factors the most important deviation in actual practice from the ideal conditions are that

(1) The journal is never perfectly centered in the bearing
(2) End-leakage is usually an important factor rather than a negligible one
(3) The bearing surfaces are never smooth
(4) Many bearing surfaces are cut up with oil-grooves, although it is now recognized generally that the presence of such grooves on the high-pressure side of the bearing is very undesirable

Sommerfeld[6] has taken the first step in modifying the above ideal equation and has calculated the corrections arising from the first of the above-mentioned non-ideal conditions, the fact that a loaded journal never operates in a perfectly central position. Such a position is, of course, approached fairly closely at high speeds or viscosities and low loads; or in other words at high values of $z\ N/p$. If the journal does not deviate more than 20 per cent from the central position, the increased rate of shear on the high-pressure side is almost counterbalanced by the decreased rate on the other, and the value of f is therefore affected but little. As $z\ N/p$ decreases still further and the eccentricity increases correspondingly, the value of f deviates more and more from that calculated for the central position. The results of Sommerfeld's rather complicated derivations may still, however, be plotted against $z\ N/p$ and the lines thus calculated for values of c/D of 1/1000 and 1/250 are shown dotted in Fig. 1. It will be noted that these lines do not continue indefinitely to approach the origin, but pass

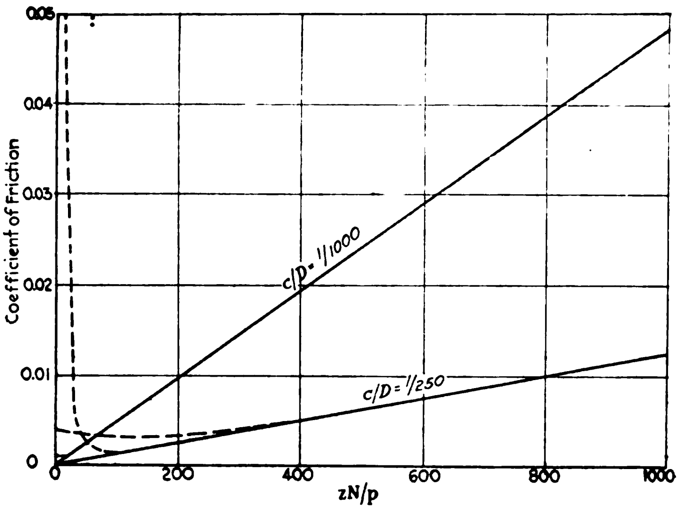

Fig. 1—Predicted Curve for the Ideal Bearing

through a minimum and then rise slightly for further decreases in $z\ N/p$. The magnitude of the deviation and the value of $z\ N/p$ at which it becomes appreciable are both much greater at the higher clearances, as is indicated by the two dotted curves in Fig. 1 that actually cross at low values of $z\ N/p$. It must be emphasized, however, that Sommerfeld's derivations make correction for only one of the four important disturbing influences present in most actual bearings and do not take into account the effects of end-leakage, surface roughness and oil-grooves which may well be of equal or greater importance.

The probable qualitative effects of some of these other disturbing factors can, however, be anticipated. Thus end-leakage undoubtedly will tend to keep the supporting film thinner than would be the case otherwise, especially if the clearance is large or the length of the bearing small compared to its diameter. As a result, bearings with large clearances should behave more like those with smaller clearances and give higher coefficients than those predicted by Sommerfeld. The magnitude of these effects can be estimated only by plotting actual results as is done in the next section of this paper.

The impossibility of getting perfectly smooth bearing surfaces also has a very important effect on the problem. Sommerfeld's calculations indicate a practically constant value of f for low values of $z\ N/p$ clear down to zero; whereas it is common knowledge that, if $z\ N/p$ is decreased too far, by making the appropriate change in any of the three factors, the fluid film is ruptured and the value of f increases enormously. This discrepancy between prediction and observation is recognized by Sommerfeld as arising from his assumption that *any* film, however thin, would keep perfectly smooth metal surfaces apart; while actually, when the thickness decreases to a certain point comparable with the roughness of the surface, metal-to-metal contact and abrasion take place with a sharp rise in the coefficient. Contrary to the implication of several writers, this critical point does *not* bear any real relation to the "point of minimum friction" calculated by Sommerfeld. In practice, the rise in the coefficient is very sharp. In general it is similar to the dashed-line curve drawn in Fig. 1.

The effect of oil-grooves on the high-pressure side of a bearing also would be to thin out the pressure film because of the tendency of oil to flow through the grooves from the high to the low-pressure portion. This effect probably would be less pronounced at high values of

[5] It must not be overlooked that both the pressure and the speed of motion may have an additional indirect effect on the friction by changing the viscosity of the lubricant, and it is essential throughout this discussion to consider the viscosity as that of the lubricating film at the actual pressure and temperature prevailing therein.

As to the effect of pressure, recent published work by Hersey, supplemented by a private communication, and in England, has indicated that very high pressures markedly increase the viscosity of oils, especially mineral oils as contrasted with animal and vegetable oils. These increases are, however, only a matter of 5 to 15 per cent at 40 deg. cent. (104 deg. fahr.) for the maximum nominal pressures of 1500 lb. per sq. in. for which journal bearings are generally designed. For pressures tenfold as great as this, which may be approached in portions of a poorly aligned bearing or in reciprocating parts, the increase in viscosity is about threefold for animal and vegetable oils and sixfold for most mineral oils. Recent work done here by Professor Hersey indicates that these effects are much less marked at higher temperatures.

The indirect effect of speed on the viscosity is due, of course, to the change in the temperature of the oil-film in the bearing and is not so readily calculable; but by installing a properly insulated thermocouple in the bearing, with the junction in direct contact with the oil-film and polished off to conform with the bearing surface, it is possible to determine the working temperature of the film with very satisfactory accuracy, except possibly at very high speeds. The viscosity of the lubricant at any temperature can be determined readily by measuring it at two or three temperatures and plotting the results according to the methods suggested by Dean and Lane, by W. H. Herschel and by Wilson, McAdams and Seltzer.

[6] See bibliography at end of paper.

$z\ N/p$, where the oil could not escape nearly as fast as it was being brought in, and also at very low values of $z\ N/p$, where the pressure film is extremely thin and the oil would find difficulty in escaping rapidly from the film *to* the oil-grooves, unless they were very close together.

In spite of the fundamental importance of Sommerfeld's work and the probability that factors which he neglected might be of equal or greater importance in actual practice, no one appears to have made a serious attempt to check his predicted curves with the rather extensive data available in the literature as to the behavior of actual bearings. Accordingly, we have undertaken to do this in the next section, which treats of the application of the recommended method of plotting to data in the literature.

Application of the Recommended Plotting Method

It has been shown in the previous section that, on the basis of dimensional reasoning, the coefficient of friction of any given bearing must be *some* function of the modulus $z\ N/p$, rather than an independent function of the separate variables as is generally assumed. In other words, if a series of observed values of f on a given bearing are plotted against $z\ N/p$, they will all approximate a smooth curve, regardless of what combinations of z, N and p determine the value of the modulus. The position of the curve thus obtained for a given bearing should indicate the effects of various factors in bearing design, such as clearance, surface roughness, end-leakage and the like. It should afford a logical method of comparing the efficiencies of different bearings even though they may have been tested under widely varying conditions.

A cursory study of the literature appears to indicate that a great wealth of data is available for drawing such plots; but in many cases closer inspection indicates the absence of some one or two essential items of data, generally the temperature-viscosity curve of the lubricant used, or the operating temperature of the bearing. These can and have been approximated from data in the literature in the cases of some well known animal or vegetable oils, but for mineral oils this method fails.

Oiling Method and Oiliness-Factor Effects

Investigating the effects of the method of oiling and the oiliness factor in the fluid-film region, the most extensive series of tests on any single bearing on which full data are available, where the effects of variable clearance, length and the like do not enter, is in an unpublished thesis at the Massachusetts Institute of Technology by Professor Hersey. The tests were made on a normal full 1 x 3-in. bronze bearing with a clearance ratio of 1/250. Using a variety of speeds and loads, Professor Hersey tested four oils that might be expected to vary widely in oiliness and did vary greatly in viscosity. These were sperm, lard, machine and engine oil. He also used four different rates of oiling: 2 drops per min., 10 drops per min., 30 to 60 drops per min. and bath lubrication. By plotting 400 observed values of f against $z\ N/p$ on a single chart,[7] about 30 x 120 in. in size, and using different colors for the several methods of oiling and a different kind of point for each oil, it was possible to develop many features not capable of clear demonstration within the limits of a printed page. For example, an inspection of the plot indicated very clearly that 90 per cent of the points lay fairly close to a single straight line and that the remaining 10 per cent, which were scattered and higher, were all of a single color and corresponded to the 2-drop-per-min. rate of oil-feed. The different kinds of oil showed no consistent deviations either above or below the line that represented all the points.

Fig. 3 shows *all* Hersey's points for values of $z\ N/p$ below 1100, except the points mentioned above which were obtained when the rate of oil feed was only 2 drops per min. The results with the different oils are differentiated by the type of point used. The remarkable simplification and coordination produced by this method of plotting the data obtained under such a wide variety of conditions are brought out clearly by the fact that a single straight line represents practically every point within the limits of error. These are certain to be fairly large in any measurements of friction. The equation of this line is $f = 0.002 + 0.000018\ z\ N/p$. The slope is higher than that specified by Professor Hersey[8] which, in these units, would be 0.0000164; but it fits the points better, especially those of the engine oil at very high values of $z\ N/p$. Professor Hersey's article contains no plot for comparison.

This degree of agreement would indicate the validity of the following important, though tentative conclusions:

(1) Confirming the conclusions of dimensional reasoning, the method of plotting f versus $z\ N/p$ for a given bearing, method of oiling and the like, gives a consistent series of points approximating a definite line, regardless of what combination of the three variables determined the value of $z\ N/p$. In other words, it does not matter whether the coefficient of friction is measured at a high load and a low speed, or vice versa, provided the resultant value of $z\ N/p$ is the same.

(2) For a given value of $z\ N/p$, the method of oiling, not including the effect of oil-grooves, does not affect the values of $z\ N/p$, providing *enough* oil is supplied to keep the bearing *full*. Lesser amounts give high and scattered results, but anything more, however supplied, has no effect on the fundamental relationship between f and $z\ N/p$.

(3) Within the range studied by Professor Hersey, the oiliness factor of different lubricants had no effect upon the results, the viscosity being the only significant variable

The importance of the last tentative conclusion is so great that the results of a single series of experiments, however carefully carried out, must not be considered as adequate proof. But the conclusion is in entire agreement with the results of Herschel.[8] Holde,[8] and others, according to which even non-lubricants such as glycerine or sucrose solutions gave the same results as the best lubricants in the region of fluid-film lubrication. If oiliness is due to the adsorption of a very thin film of something on the metal surface, this film would scarcely be expected to affect the results appreciably when the surfaces were separated by a relatively thick fluid film in the range covered by Professor Hersey's experiments, however important it might become in the region of partial lubrication where the fluid film has been ruptured.

It should be said, however, that a few isolated experiments, which have been reported, appear to contradict this conclusion, but all these results were obtained on abnormal types of bearings or parts of bearings, where

[7] Convenient engineering units rather than absolute ones were chosen for all the calculations. Thus N is in revolutions per minute, p is in pounds per square inch of the projected area, and z is in centipoises relative to water at 68 deg. fahr., where it has a viscosity of 1 centipoise = 0.01 poise. In Hersey's experiments, z varied from 38 to 440; N, from 280 to 1200; and p, from 40 to 255. Fig. 2 gives a conversion chart, modified from Herschel's data by a method first used by MacCoull, for obtaining the viscosity in centipoises from the density and time of efflux of any fluid in any of the standard viscosimeters.

[8] See bibliography at end of paper.

there is always a likelihood that some portion of the bearing, especially the "on" edge, may have been operating under conditions of partial lubrication. Again, as we *approach* the point of film rupture at very low values of $z\ N/p$, some results which will be discussed later indicate that the property of oiliness may begin to manifest itself. There appears to be little doubt, however, as to the lack of significance of the oiliness factor in the major portion of the region of fluid-film lubrication.

In addition to the results shown in Fig. 3, Professor Hersey made a few observations at values of $z\ N/p$ between 1100 and 8000. Most of the high values were obtained by using a very viscous engine-oil. The results have been plotted on a large scale and are rather scattered, some approximating the line fairly closely but others giving considerably higher coefficients, the average deviation being about 20 per cent. This can well be explained on the basis of the observation by Professor Hersey that it was exceedingly difficult to get the very viscous oil to feed-in properly through the small oil-holes. The abnormally high values of f probably are due to the bearing not being entirely filled with oil and are similar to those obtained with the less viscous oils when an insufficient quantity of lubricant was supplied. On the whole, the results with the engine oil confirm, rather than invalidate, the extrapolation of the previously obtained straight line to these very much higher values of $z\ N/p$.

As already indicated, not only does the agreement of all the observed points with a single line permit us to draw important conclusions, but the position and shape of this line is a measure of the efficiency of the bearing, and also affords a method of determining the actual effect of the non-ideal conditions neglected by Sommerfeld in his calculations regarding the behavior of theoretical bearings. Before proceeding to this phase of the discussion, however, it seems desirable to present other data on other bearings for comparison with the results of Hersey and the predictions of Sommerfeld.

Effect of Variable Clearance

Probably the most important single factor in the design of bearings is the clearance. Practically the only

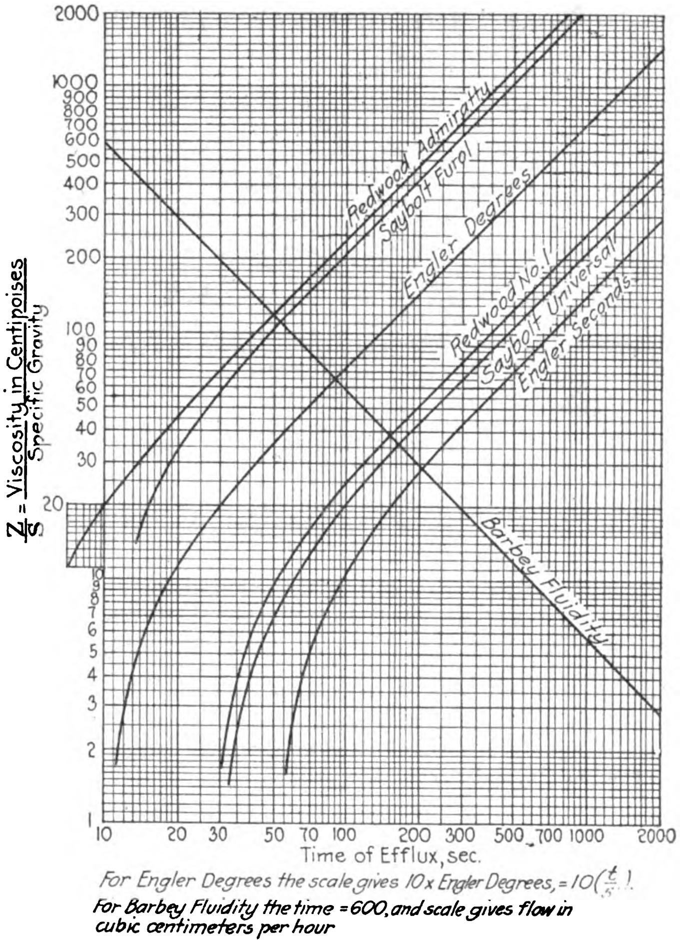

Fig. 2—Conversion Diagram for Viscosimeters

extensive published results in which this is the only variable appear to be those obtained by Lasche on a gun-metal bearing 260 x 110 min. (10.20 x 4.33 in.) against a steel shaft. Lasche's article[9] does not give his original experimental points but expresses his results in the form of a series of surfaces on three-dimensional diagrams in which two factors are kept constant. Using the rec-

[9] See bibliography at end of paper.

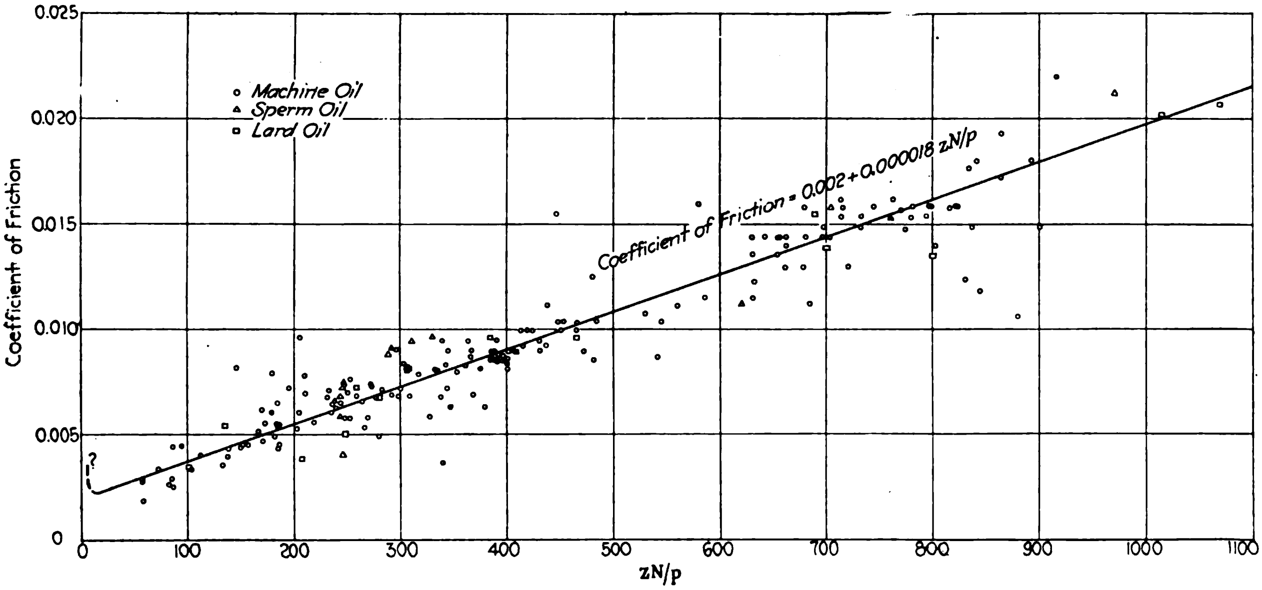

Fig. 3—Results Obtained by Hersey in Lubricating a Steel Journal 1 In. in Diameter and 3 In. Long and Having a Bronze Bearing with a Clearance-Diameter Ratio of 1 to 250 with Lard, Machine and Sperm Oils

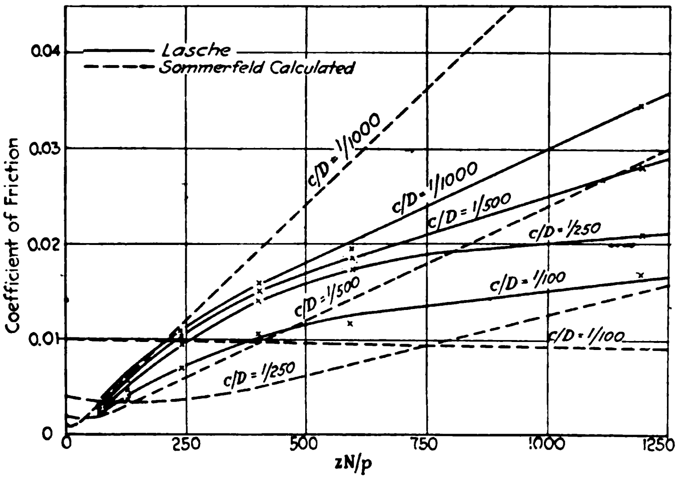

FIG. 4—COMPARISON OF LASCHE'S DATA FOR VARIOUS CLEARANCE-DIAMETER RATIOS OBTAINED AS THE RESULT OF EXPERIMENTS WITH THE CALCULATED RESULTS OF SOMMERFELD

ommended methods of plotting, his results therefore take the form of a smooth curve rather than show the agreement of experimental points. These curves are shown in Fig. 4 for four clearance-ratios, together with the corresponding dotted curves that represent the theoretical calculations of Sommerfeld for the same clearances. The outstanding differences between Lasche's observations and the theoretical curves are that

(1) The effect of clearance is in the expected direction, but is much less pronounced than that calculated
(2) The observed curves are convex upward rather than concave
(3) The relative position of the lines for the different clearances remains unchanged down to the lowest observed values of $z\ N/p$, instead of crossing as Sommerfeld's calculations would indicate

These discrepancies are so striking that Lasche's results could be accepted only with reservation, especially since his surfaces were badly cut-up with oil-grooves, were it not for the fact that the three differences are all confirmed by the rather extensive results of Heimann.[10] Unfortunately, Heimann did not record the viscosities of his lubricants and hence his results cannot be plotted against $z\ N/p$ to determine their positions, but he presents several curves for two different clearances in which the oil and the load were identical and only the speed was varied. The shape and relative positions of the curves f versus N for different clearances were almost identical with those of Lasche in the foregoing three respects, as is illustrated by two typical curves in Fig. 5. Heimann's bearings were ring-oiled and had no oil-grooves on the pressure side. The interesting portion of Heimann's curves in the critical region are discussed in the following section of this paper.

[10] See bibliography at end of paper.

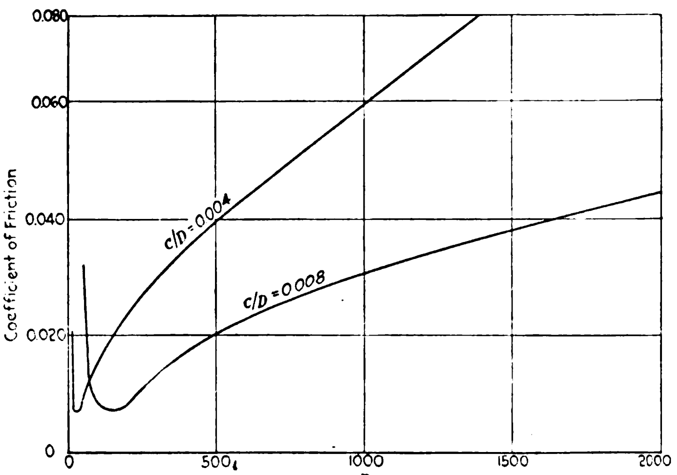

FIG. 5—RESULTS OBTAINED BY HEIMANN FOR DIFFERENT RATIOS OF THE CLEARANCE IN A BEARING TO ITS DIAMETER

The comparison of Hersey's results with those of Lasche for the same clearance indicates a fairly good agreement at high and low values of $z\ N/p$, but considerably higher results in the intermediate region. As indicated in the previous theoretical discussion, this is what might be expected due to the effect of the oil-grooves present in Lasche's bearing.

FACTORS AFFECTING THE CRITICAL POINT

The results obtained by Stribeck and Heimann are considered in discussing the effect of bearing metal and clearance on the critical point of film rupture. Unfortunately, neither Hersey nor Lasche extended the measurements to values of $z\ N/p$ low enough to reach the point of film rupture and enter the interesting region of partial lubrication where the effects of the oiliness factor and of the nature of the bearing metal should begin to manifest themselves. By far the most extensive results in this region for which all the necessary data are given are those of Stribeck.[10] He worked with two bearings, both of the same diameter, 70 mm. (2.76 in.), but of different lengths, 137 and 230 mm. (5.4 and 9.06 in.), and made of different metals. Data are not given as to the clearance, both having been scraped-in. From the appearance of the curves it would seem that the bronze bearing had the larger clearance and hence gave somewhat lower coefficients over most of the range covered. Stribeck's original data are presented in a long series of families of curves but, again, plotting against $z\ N/p$ all his observed values of f (except a very few on the first run on white metal before the bearing was properly "run-in") produces a remarkable simplification of results, the points for each bearing approximating a smooth curve, as is shown clearly in Figs. 6 and 7, the latter expanding the lower part of the scale.

Fig. 7 is the more interesting in that it shows very clearly the behavior of these two bearings in the critical region and brings out the sharp rise in the coefficient as the value of $z\ N/p$ is lowered below the critical point. In this region it is apparent that a change in the viscosity, the speed or the load has exactly the *opposite* effect on f that the same change had in the region of fluid-film lubrication. This accounts for the apparently contradictory results of a number of investigators who did not distinguish clearly between the two fields. The most important point brought out by Fig. 7 is that for some reason the two bearings differed enormously in their ability to maintain a continuous fluid-film and keep within the realm of perfect lubrication. The curve for the bronze bearing breaks away when $z\ N/p = 13$, while the white-metal bearing kept its fluid film until $z\ N/p = 1$, which is as low as any observations of which we are aware.

The importance of these differences is even greater than might appear from casual inspection. If we take a factor of safety of 5 in calculating the proper value of $z\ N/p$ for given operating conditions, this would permit operation of the white-metal bearing at a $z\ N/p$ value of 5 and a coefficient of friction of 0.0033; whereas, with the bronze bearing, operation with an equal factor of

safety would have to be carried on at a $z\ N/p$ value of 65 and a coefficient of 0.0065.

It becomes extremely important, therefore, to determine the fundamental cause for this marked difference in the behavior of the two bearings. A similar advantage of white metal over bronze was reported by Herschel.[11] It might be due to any one of the following causes, or to a combination of them: (*a*) a smaller clearance in the white-metal bearing that might help to maintain the fluid film longer; (b) a smoother initial surface on the white-metal bearing that permitted the film to thin out farther before rupture; (*c*) the softer character of the white metal that makes it possible to bed-down surface irregularities and prevent highly localized pressures and a consequent rupture of the film; (*d*) a specific adsorptive property of the white metal that might tend to hold a very thin semi-solid film of lubricant and prevent metal-to-metal contact even after the true fluid-film had become extremely thin.

There are two other conceivable causes of the differ-

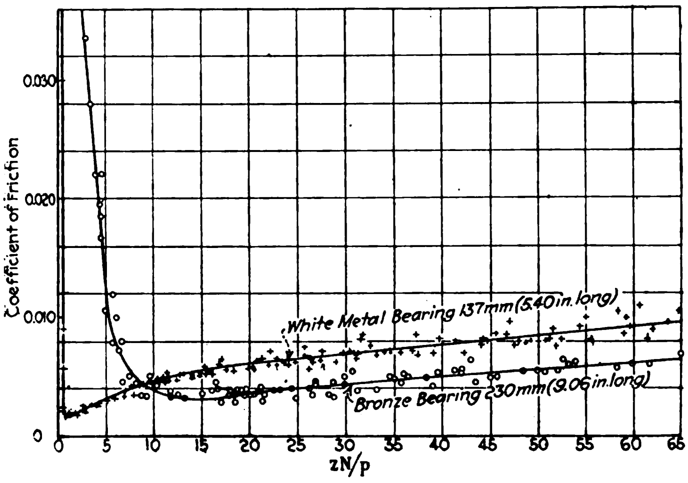

FIG. 7—DATA OBTAINED BY STREIBECK ON BRONZE AND WHITE METAL BEARINGS WITH A STEEL SHAFT 70 MM. (2.76 IN.) IN DIAMETER

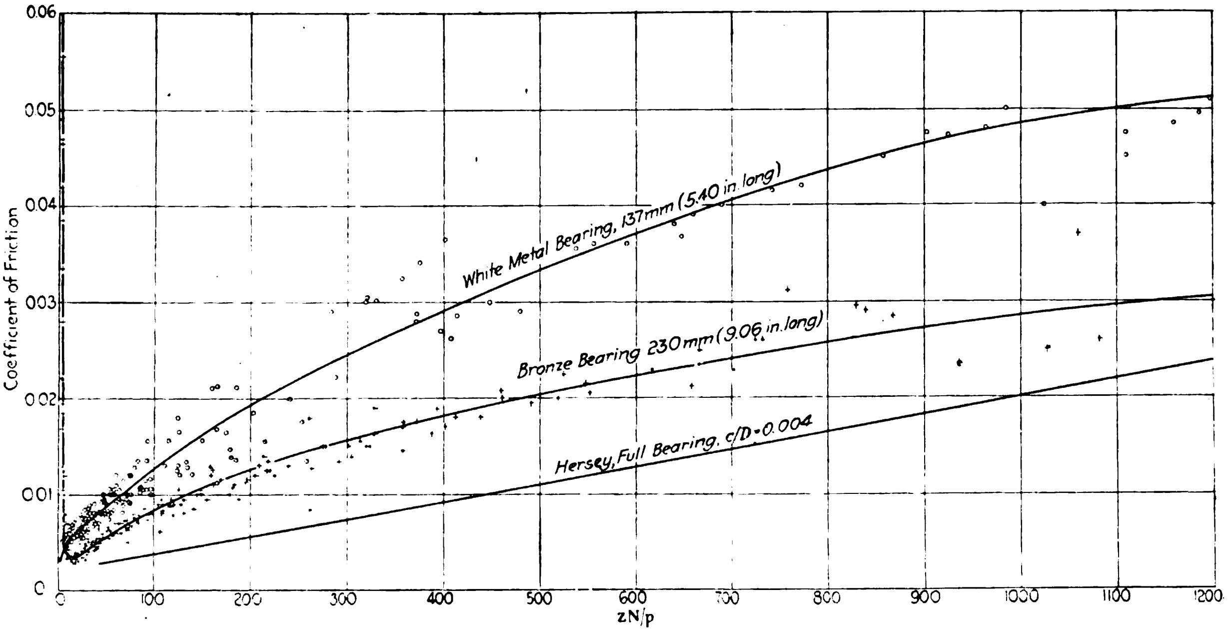

FIG. 6—COMPARISON OF STRIBECK'S RESULTS WITH THOSE OBTAINED BY HERSEY

tioned in the above list: (*a*) length, because the longer bearing would be expected to hold the film better, although actually it held it more poorly; (*b*) the possibility of improper alignment of the bronze bearing, or a similar disturbing factor. This possibility seems to be
ence in behavior of the two bearings that are not men-
ruled out because the bronze bearing gives lower coefficients than the white metal for the higher values of $z\ N/p$.

There is evidence in favor of each of the four possibilities above mentioned. The results of Heimann given in Fig. 5 show similar results where the clearance was apparently the only variable; but in Stribeck's case the difference in the clearances does not appear to have been great enough to account for the great lowering of the critical point. Other evidence appears to support some of the other possibilities. To make a definite choice from these four possibilities until more evidence is available would be to speculate unduly. The data presented bring out the importance of comprehensive experimentation with different types of bearing in this critical region of lubri-

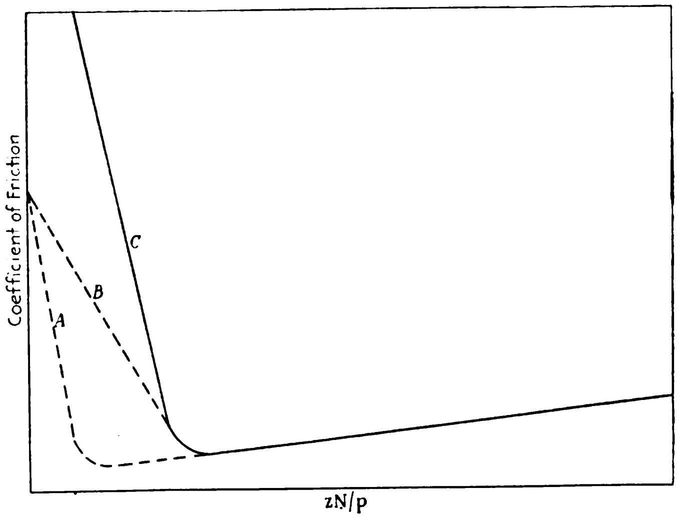

FIG. 8—CURVES SHOWING THE EFFECT OF THE OILINESS OF THE LUBRICANT AT THE CRITICAL POINT

[11] See bibliography at end of paper.

cation and indicate that great improvements can be expected in bearing design when it is known which of these factors are the most important.

OILINESS EFFECTS

One of the most important questions regarding the location of the critical point is whether it can be changed by varying the oiliness of the lubricant. This factor does not appear to affect the results in the major portion of the fluid-film range; but, on the other hand, it does have an effect in the region of partial lubrication. The question naturally arises as to whether a lubricant of high oiliness will give a curve such as A or one such as B, in Fig. 8, compared with Curve C for a less oily lubricant. This point is of great practical importance, because a lowering of the critical point as in Curve A would make it possible to *operate* with a given factor of safety at a much lower coefficient of friction, while a curve such as B could only affect the results under abnormal conditions of very low $z\ N/p$. In spite of its importance no data appear to have been published on the subject based on experiments with a normal full bearing. The results of Hersey, Lasche and most other observers, never reached the critical point. Stribeck obtained many valuable data in this region but used only a single mineral oil. The only data that appear to bear at all on the subject are those of Archbutt and Deeley.[12] Unfortunately, they used a Thurston type of machine where the method of building up the fluid film is abnormal, which detracts from the weight to be placed upon the results. Furthermore, a few other experiments made at very low speeds in the same region did not give results concordant with those plotted. It is also rather surprising to note that the critical point seems to vary in almost exact inverse ratio with the viscosity of the lubricant.

Recognizing then the doubtful validity of these results, it nevertheless seems worth while to present the curves in Fig. 9 to indicate what *may* be found as to the behavior of different oils in the same bearing. If results of this type can be duplicated on normal bearings, it will prove conclusively the great value of lubricants with a high degree of oiliness, in permitting bearings to be operated at much lower values of $z\ N/p$ without danger of seizing or abrasion. Some of Heimann's experiments appear to confirm these conclusions in that the critical value of $z\ N/p$ for some glycerine solutions, such as are shown in Fig. 10 and on which he gives enough data to permit the approximation of $z\ N/p$, is apparently well over 100, which is very much higher than any other value that has come to our attention.

We cannot emphasize too strongly the desirability of carrying the results down through the break point, when comparing oils on oil-testing machines, rather than devoting attention to experiments in the fluid-film region where the viscosity of the lubricant is the only factor of importance. Such experiments should be carried out also on normal rather than on Thurston or other abnormal types of bearing. The loads used need not be excessive, as the critical point can be reached more readily by reducing the speed than by increasing the load. If the present volume of experimental work on lubrication could be concentrated on this point instead of being scattered over the whole field where the results have comparatively little significance, we might confidently expect to add greatly to our knowledge of lubrication within a comparatively short time. It is possible that there are many unpublished data that, if put into the recommended form, would settle some of the very points about which we are now most uncertain.

It must, of course, be recognized that working in this range of partial lubrication will result in abrasion of the surfaces and require their frequent renewal if check results are to be secured. The data obtained by Stribeck indicate, however, that this factor may not be so serious as anticipated if one is careful not to go too far into the region of partial lubrication. In a subsequent article we shall discuss the quantitative *measurement* of this property of oiliness and the methods by which one can investigate the region of partial lubrication much farther than is possible or desirable by using a conventional bearing.

MISCELLANEOUS EFFECTS

There are a number of other isolated experiments covering more limited ranges that seem worth presenting. The best of these are the results cited by Kingsbury[12] on a light, high-speed bearing with a very small clearance, 1/3750, in which air alone was the lubricant. These results are shown in Fig. 10 in comparison with the calculated results of Sommerfeld for the same clearance. It will be noted that in this case the correspondence is very good and probably is due to the remarkable smoothness of the bearing surfaces and to the minimization of end-leakage by the small clearance and the fact that one end was closed. The normal operating conditions of the Sperry gyrostabilizer are indicated also by a point on the same figure.

A considerable amount of data is available on more or less "imitation" bearings, where the normal conditions of pressure distribution and constant total clearance do not hold and which are therefore of much less value from a practical standpoint. Among these are the Tower type of machine with a one-half bearing or less and the Thurston type where the load is applied by pressing two one-quarter bearings together. In neither of these cases does a film build up in the normal way. There is also a possibility of one part or edge of the bearing section operating under conditions of partial lubrication while the remainder is in the fluid-film region and the resulting curves are meaningless. Again, it has been found experimentally that the coefficient of friction obtained in some of these machines depends to a considerable extent on the precise method of rounding the bearing at the edge where the oil enters. If this is kept sharp the coefficients generally are higher than those obtained under similar conditions for full bearings, while if the edge is ground off in a proper curve to wedge in a thick film of oil it will sometimes give *lower* coefficients than normal full bearings of average clearance.

As a result of all these sources of trouble in these special machines, it is not surprising that many of the data are discordant and frequently do not give a good line when plotted against $z\ N/p$. Some of the best data, however, give fairly definite lines and to make comparisons possible, some results recently obtained by Prof. H. W. Hayward, of the Massachusetts Institute of Technology, on a Thurston-type Riehlé-machine with well rounded entrance edges, are plotted in the customary manner and are shown in Fig. 11. The agreement is obviously very good, in spite of the fact that N varied from 158 to 389, z from 55 to 105, and p from 20 to 500. Furthermore, the whole curve is remarkably low, indicating that the wedging effect of the rounded entrance edges produced a large effective clearance. For comparative purposes, lines representing Hersey's results and the friction experiments by B. Tower[12] are shown in the same illustration.

[12] See bibliography at end of paper.

As noted previously, there are many additional data in the literature that appear to afford further evidence on some of the points as yet unsettled; but an inspection has always indicated that some of the essential data were missing. These are generally either the temperature-viscosity curve for the lubricant or the operating temperature of the bearing. The failure to secure and record such data in any modern investigation is virtually unpardonable.

Summary of Essential Conclusions

In view of the necessary length of the foregoing detailed discussion, it seems desirable to state briefly the essential conclusions that appear to be justified by the data now available. We appreciate fully the fact that in some cases the data are not really adequate to substantiate these tentative conclusions, but nevertheless it seems desirable to state them in definite terms so that confirmation or contradiction can be offered more readily by other investigators, possibly from data already obtained but not published. Certainly such a procedure is preferable to the perpetuation of the present hazy uncertainty and empiricism that pervade most of the literature on the subject. With this object in view, therefore, the following tentative conclusions are submitted.

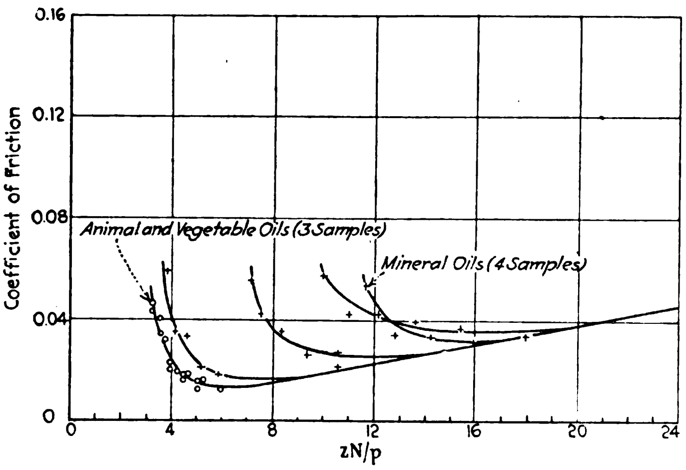

Fig. 9—Results of Lubricating the Same Bearing with Different Oils

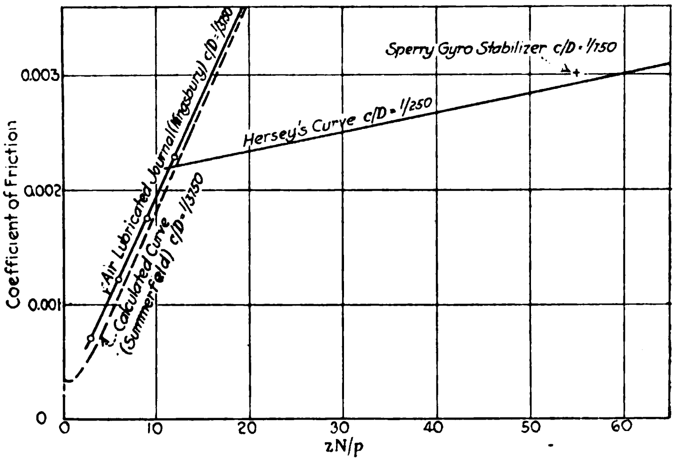

Fig. 10—Comparison of the Results Obtained by Kingsbury with an Air Lubricated Journal with the Predicted Curve for the Ideal Bearing of Sommerfeld and the Results Obtained by Hersey

I—For any normal journal bearing operating in the region of fluid-film lubrication, the coefficient of friction is not an independent function of the speed, the load and the operating viscosity of the fluid at the operating temperature, but is rather a function of their *combination* in the form $z\ N/p$.

II—If the bearing and lubricant are both kept constant, all the observed values of f approximate a smooth curve when plotted against $z\ N/p$. Starting at very high values of $z\ N/p$ and decreasing them, all the curves gradually approach a small value of f, generally in the neighborhood of 0.002. Before the ordinate axis is reached, however, the fluid-film is ruptured, the coefficients rise very sharply with a further decrease in the $z\ N/p$ value and metal-to-metal contact and abrasion take place. The first region is termed the region of fluid-film or perfect lubrication; the second, the region of partial lubrication. The intermediate point at which the coefficient is a minimum is the critical point of fluid-film rupture.

III—In the region of fluid-film lubrication, the effect of various secondary factors, other than speed, load and viscosity as already discussed, on the position of the characteristic curve of f versus $z\ N/p$ is as follows:

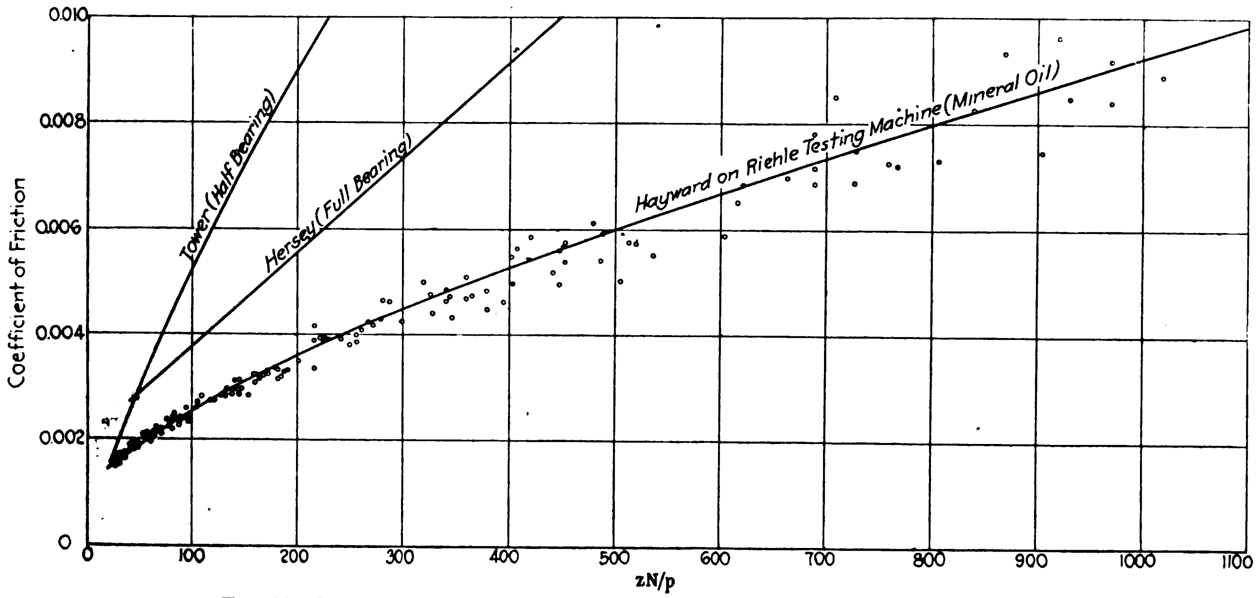

Fig. 11—Comparison of the Results Obtained by Tower, Hersey and Hayward

(1) It is unaffected by the method of oiling, provided enough oil is supplied to keep the bearing full

(2) It is unaffected by any variation in the "oiliness" of the lubricant, except possibly when the point of film rupture is approached very closely

(3) The curve is higher at small clearance-ratios than at large ones, but the differences are by no means so great as predicted by the general theory of lubrication for *ideal* bearings

(4) End-leakage tends to give abnormally high friction for bearings that are very short or have very large clearance

(5) The presence of oil-grooves in the pressure side of the bearing tends to give curves that are markedly convex upward, with abnormally high values in the intermediate range of zN/p. This is due unquestionably to interference with the normal building up of the fluid film in this range. In the absence of such grooves the lines are generally nearly straight, although they do not pass through the origin

(6) Within the limits of good practice, the character of the bearing metal does not affect the location of the characteristic curve

IV—The position of the critical point is very important, since this largely determines the safe operating value of zN/p, and hence the working value of f. Apparently, it is affected as follows:

(1) Within the limits of good practice, the use of smaller clearances tends to lower the critical value more than enough to counterbalance the increase in the height of the curve in the fluid-film region

(2) The use of lubricants of high oiliness tends to lower the critical point, probably because the presence of an adsorbed semi-solid film on the metal surfaces helps to prevent rupture of the fluid lubricating film, even after it has become extremely thin

(3) Various bearing metals differ appreciably in their ability to maintain the fluid film at extremely low values of zN/p

The greatest chance for improvement in the design and lubrication of bearings appears to lie in a thorough study of the effect of these variables on the critical point of film rupture.

V—In the region of partial lubrication,

(1) Viscosity is by no means the most important property of the lubricant

(2) The oiliness factor comes very largely into play

(3) The character of the bearing metals has an important effect on the friction

(4) The coefficient of friction is no longer *necessarily* a function of zN/p, even if the bearing and the lubricant are both kept constant, because a delicate semi-solid adsorbed film has now become a fundamental factor in the behavior of the bearing, and this may be affected *differently* by corresponding inverse changes in the load and the speed. The case is exactly analogous to conditions that might prevail in the region of fluid-film lubrication if the load were high enough to deform the bearing metal, which can happen without rupturing the fluid film if the speed is high enough, or if the speed were so high that the evolution of heat fused its surface. Either of these conditions would change the essential character of the bearings; hence, results for the same value of zN/p might give different values of f. Such conditions are so far outside ordinary operating conditions in fluid-film lubrication that they can be neglected in a practical treatment of the subject, but in the region of *partial* lubrication we are dealing with a much more delicate bearing surface in the form of this adsorbed semi-solid film, which is much more susceptible to high pressures and temperatures than is any bearing metal

By the same reasoning, the critical value of zN/p may be somewhat different for different combinations of the load and the speed and may change with the temperature. There is undoubtedly a fairly wide region of load and temperature values within which the critical point and f in the region of partial lubrication will be a definite function of zN/p for a given bearing and lubricant, and experimental work should be undertaken to define these limits

VI—The simple equations for ideal bearings approximate more closely to the observed curves for actual bearings, especially at the larger clearances, than do the much more complicated equations derived by Sommerfeld on the basis of correcting for the eccentricity of the journal, but neglecting the important effects of end-leakage and surface roughness, which are calculable only with difficulty. Furthermore, the points of minimum friction calculated by Sommerfeld, on the basis of hydrodynamics only, bear no relation to the actual critical points of film rupture.

VII—The true carrying power of a bearing can be calculated readily, once the critical point is known. For example, if the critical point is found to come at $zN/p = 10$, the maximum pressure that the bearing will handle without rupture of the film is $zN/10$. In other words it is directly proportional to the speed and to the working viscosity of the lubricant, divided by the critical value of zN/p. The only *absolute* limit to the carrying power of a given bearing is, therefore, the pressure that will deform the bearing or seriously deflect the shaft.

VIII—The efficiency of a bearing should not, however, be judged on the basis of its carrying power nor by the minimum coefficient of friction at the critical point, but rather by the coefficient of friction obtained when it is operating with a reasonable factor of safety. The proper factor of safety will vary with conditions but as a general proposition *the efficiency of a bearing can be measured in terms of the value of f obtained when operating at the value of zN/p which is five times that at the critical point.*

Application of the Conclusions

The foregoing conclusions bring out very clearly the futility, if not the absurdity, of many of the common methods of testing oils and diagnosing the causes of lubrication difficulties. The application of these conclusions and methods of treatment to practical lubrication problems is therefore worthy of further brief consideration.

Since practically all ordinary "oil-testing" machines operate in the region of fluid-film lubrication, they are in reality nothing more nor less than a very poor type of viscosimeter and can give no information as to the behavior of an oil that is not given much more quickly and accurately by the intelligent use of a Saybolt viscosimeter or similar instrument. This naturally raises the question as to why such machines are not customarily designed to operate in the region of partial lubrication, where they *could* measure the effect of the oiliness factor. The reason is that under conditions of partial lu-

brication there is inevitably abrasion of the bearing surfaces; this abrasion changes both the clearance and the smoothness of the surfaces and makes it almost impossible to get concordant results. The majority of bearing machines are therefore designed for and operated under conditions of perfect fluid-film lubrication to get reasonably reproducible results, apparently regardless of their lack of real significance.

Several investigators have realized the need for reaching or approaching this point of rupture, even at the expense of having to change or renew the bearings frequently. But almost invariably this point of rupture of the film has been reached by greatly increasing the loads. This necessitates heavy cumbersome apparatus, tends to deflect the journals or cause the bearing metals to flow and makes accurate measurements almost impossible. It would be much better to keep the load at a value approximating service conditions and to *reach the critical value of $z\ N/p$ by lowering the speed instead of increasing the pressure.* The film rupture can then be measured on lighter apparatus, with greater accuracy and with much less abrasion of the surfaces per unit of time.

Similarly, in determining the shape of the characteristic curve at very high values of $z\ N/p$, it is desirable in general to obtain these high values by lowering the load rather than by increasing the speed, since working at high speeds with the consequent heating effects[13] makes it difficult to measure accurately the working temperature and the viscosity of the lubricating film.

The present methods of diagnosing and attempting to cure lubrication troubles are extremely haphazard. The customary rule of using the oil of the lowest viscosity that will operate satisfactorily has, of course, a fundamentally sound basis, but there is no definite criterion for choosing oil except by the method of trial and error. The fact is often overlooked that *every* method of lowering the coefficient of friction, either by reducing the speed, the viscosity of the lubricant or the area of the bearing, accomplishes its effect by bringing the bearing closer to conditions where $z\ N/p$ has a very low value and where the film is likely to be ruptured and the bearing injured. In other words, as is frequently the case in engineering work, these savings are accomplished by the simple expedient of reducing the factor of safety. If the factor of safety is unnecessarily high, this is desirable, but the essential point to determine is *what factor of safety is necessary* for given conditions of operation. It is then a comparatively simple matter to choose the lubricant to give neither more nor less than this proper factor of safety. For a given bearing, working under given conditions, there is a *safe* value of $z\ N/p$ that will give a reasonably low coefficient of friction and the viscosity of the lubricant should be selected to give this proper value.[14]

Where the bearings are those of line-shafts, countershafts and machine tools that are started and stopped rather frequently, subjected to peak loads, not given constant attention to prevent the temporary failure of the

[13] It should be noted that equal values of f corresponding to equal values of $z\ N/p$ do not necessarily mean equal amounts of heat evolved. The power lost in a given bearing equals the product of the factors k, f, p and N. If we assume as an approximation that $f = K_1\ (z\ N/p)$, the lost power will equal $k_2\ (z\ N^2)$, which is far from being a function of $z\ N/p$. This does not in any way invalidate the general proposition that the operating value of f is the best single measure of the efficiency of a given bearing for a given purpose, because the speed of the journal in revolutions per minute is almost invariably fixed by other considerations.

[14] This calculation is, of course, complicated by the necessity of knowing the approximate operating temperature of the bearing for the conditions in question. This generally can be estimated with sufficient accuracy except at high speeds. Even then, if a part of the energy now devoted to determining the high-speed coefficients of friction for a given lubricant in a given bearing, the data for which can apply only to the precise combination on which it was determined. were directed toward the much simpler problem of measuring and deriving formulas for the *rate of dissipation of heat* as a function of the temperature of the lubricating film, the calculation of the equilibrium operating temperature would be merely a matter of finding at what temperature the rate of heat generation in British thermal units per hour, which equals $0.202\ f\ p\ N\ D^2\ L$ (where f is calculated from the corresponding value of $z\ N/p$, and D and L are given in inches) just balances the rate of heat dissipation. The rate of heat dissipation equals $k\ (T_b\text{-}T_r)$, where $T_b\text{-}T_r$ is the difference between the temperature of the bearing and that of the room and k is the coefficient of heat transfer for the bearing in question. Stodola has determined such data for steam-turbine bearings.

TABLE 1—VALUES OF $z\ N/p$ FOR VARIOUS RECOMMENDED CONDITIONS OF OPERATION

Location of Bearing	Lubricant	p	N	z	$z\ N/p$	c/D
Automobile Crankshaft	Medium Machine Oil	300 to 700	900 to 1,400	7 to 8	15 to 25	<0.0010
Aeronautic Engine Crankshaft	Heavy Engine Oil	300 to 1,800	1,800 to 2,000	7 to 8	15 to 25	<0.0010
Stationary Gas Engine Main	Medium Machine Oil	500 to 700	250 to 800	30	25	0.0010
Stationary Gas-Engine Crankpin	Medium Machine Oil	1,600 to 1,800	250 to 800	50	15	<0.0010
Stationary Gas-Engine Crosshead	Medium Machine Oil	1,500 to 2,000	250 to 800	40	10	<0.0010
Diesel Engine Main	Heavy Engine Oil	250 to 600	60 to 160	30	15	0.0010
Diesel Engine Crankpins	Heavy Engine Oil	1,500 to 4,000	60 to 160	40	2 to 5	0.0010
Marine Steam Engine Main	Machine Oil	275 to 500	180	30 to 40	20 to 30	<0.0010
Marine Main Crankpin	Machine Oil	400 to 500	180	30 to 40	20	0.0010
Stationary Slow-Speed Main	Heavy Machine Oil	80 to 400	40 to 80	70	20	<0.0010
Stationary Slow-Speed Crankpin	Heavy Machine Oil	800 to 1,300	40 to 80	80	6 to 8	<0.0010
Stationary Slow-Speed Crosshead	Heavy Machine Oil	1,000 to 1,500	40 to 80	70	5	<0.0010
Stationary High-Speed Main	Engine Oil	60 to 250	360	15	25	<0.0010
Stationary High-Speed Crankpin	Machine Oil	400 to 1,500	360	30	6 to 15	<0.0010
Stationary High-Speed Crosshead	Machine Oil	1,500 to 1,800	360	25	5	<0.0010
Locomotive Drive-Wheel	Heavy Machine Oil	550	250	100	30 to 50	<0.0010
Locomotive Crankpin	Heavy Machine Oil	1,500 to 2,000	250	100	5 to 8	<0.0010
Locomotive Cross-Head	Heavy Machine Oil	3,000 to 4,000	250	130	6 to 8	0.0010
Marine Steam Turbine	Light Machine Oil	85	2,000	10	250	0.0010
Stationary Steam Turbine	Machine Oil	400 to 950	2,000	20	100 to 200	0.0010
De Laval 7-Hp. Steam Turbine	Light Machine Oil	7 to 15	30,000	1	1,500 to 3,000	0.0020
De Laval 300-Hp. Steam Turbine	Light Machine Oil	20 to 25	10,500	2	1,000	0.0020
Railway Car Axle	Heavy Machine Oil	300 to 450	300	100	50 to 100	
Generator and Motor	Engine Oil	30 to 80	150 to 500	25	200	0.0010
Rolling Mill Main	Hot Neck Grease	1,800 to 2,500	60	—	<1	—
Cotton Mill Spindle	Spindle Oil	1	8,000 to 12,000	2	10,000	0.0050
Gyroscope		750 to 850	800 to 1,500	60 to 30	55	0.0013

oil supply, and the like, a fairly high factor of safety, probably around 15, is required. For continuously loaded bearings that are better cared for, such as generator and turbine bearings, a factor of safety of 5 should be adequate.

Where the direction of the application of the load is changed during every revolution, it is much easier to prevent the rupture of the lubricating film as there is insufficient time for the oil to be squeezed out from the region of high pressure. Small clearances are of great assistance in this respect and, under these special conditions, well designed bearings may operate satisfactorily and give very low coefficients of friction by working at values of $z\,N/p$ that are very close to the critical point.

The use of these factors of safety obviously requires more definite data as to the location of the critical point for different types of bearing. Some indirect evidence on this point can be obtained by calculating the operating values of $z\,N/p$ corresponding to the best practice. The values in Table 1 are calculated primarily from the recommendations in Alford's book on Bearings and Their Lubrication.[15]

Another important practical application of the $z\,N/p$ concept is in connection with diagnosing the fundamental cause of lubrication troubles. The method is especially helpful in studying the behavior of the special types of bearing frequently but incorrectly termed "oilless," such as genelite, and the like.

Suppose a particular bearing with a certain oil is giving trouble, presumably due to an abnormally high coefficient of friction. As has been indicated, such high coefficients may be caused by a variety of conditions, the cure for each of which is different. If, however, $z\,N/p$ is calculated for the condition in question, it will be found that the case falls within one of the three classes that follow:

(1) In the case where $z\,N/p$ is very low, say below 10, the difficulty almost certainly is due to operating under conditions of partial lubrication, and the simplest cure is to increase the viscosity of the lubricant to give a higher value of $z\,N/p$. Some improvement can be obtained also by using a lubricant having more oiliness, or possibly by changing the metal in the bearing to make the point of rupture somewhat lower. In general it is unsafe to operate with such low values of $z\,N/p$

(2) In the case where $z\,N/p$ may be very high, say above 600, the difficulty can be overcome most readily by decreasing the viscosity of the lubricant, or by decreasing the area of the bearing and hence increasing the pressure thereon

(3) In the case where $z\,N/p$ is found to lie in a proper intermediate range, say between 25 and 200, the abnormally high coefficient of friction can be due only to highly abnormal conditions in the bearing, such as a poorly aligned shaft, too tight bearings, or improper methods of feeding the oil that do not keep the bearing full. These possibilities must be investigated and this generally can be done readily without complicating the case by trying out other lubricants of different viscosities that could only aggravate the situation

The value of ball and roller types of bearing lies, not in their giving coefficients as low as those obtainable near the critical points for good journal bearings but in the fact that they give a very flat curve of moderate height, with no critical point and sharp rise in the coefficient at low values of $z\,N/p$.

[15] See bibliography at end of paper.

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Progress Made in Garage Equipment

By H. C. Buffington[1]

Chicago Service Meeting Paper *Illustrated with* Photographs

THE paper relates specifically to the type of garage equipment that is used to handle the motor vehicle in preparation for its repair. The devices illustrated and described are those designed to bring in disabled cars, and include wrecking cranes and supplementary axle trucks; portable cranes and jacks on casters for handling cars in a garage; presses, tire-changing equipment and wheel alignment devices; engine and axle stands; and miscellaneous minor apparatus.

The different factors mentioned emphasize the great need of standardization. The thought is not to do away with a car's individuality, but to construct all parts so that cars may have efficient service to the highest degree through the agency of every serviceman.

IT is interesting to note that during 1921 alone over 10,000,000 cars and trucks were registered in the United States. It is safe to say that during 1922 another 1,000,000 cars will be added. Does it seem any wonder then that the service requirements throughout the entire automotive field are demanding such careful attention? Actual practice, resulting from the scientific study of the automotive service problem has at last reached a point where it is possible to handle automobiles in a satisfactory manner. This fact is the result of the gradual development of a general standard design. However, there is yet much standardizing to be done in details that will help to simplify the service work. Therefore, it is from the manufacturing standpoint of service equipment that I shall make the following suggestions, briefly touching upon the progress already made in garage equipment. There are, speaking approximately, two kinds of garage equipment; (*a*) that which actually makes and repairs the necessary parts and (*b*) that which handles the automobile in preparation for repair. It is this latter class that I wish to discuss.

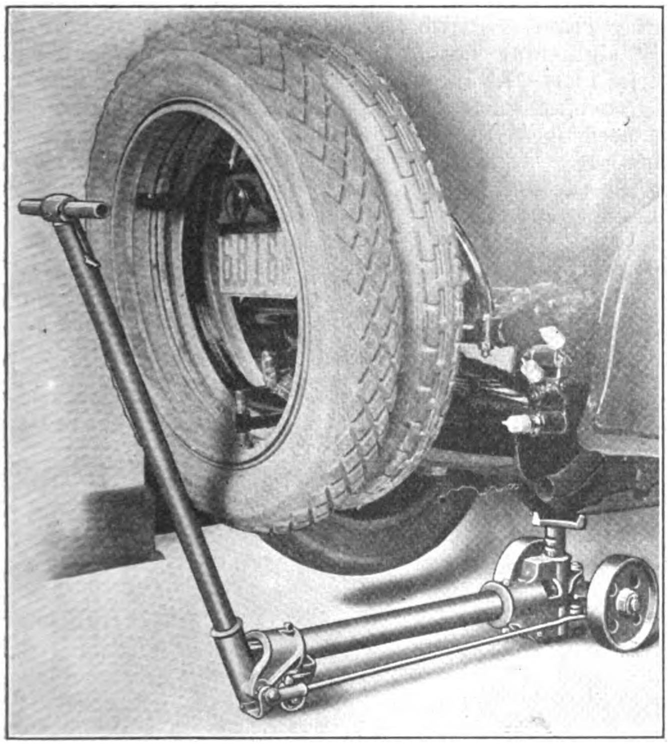

Fig. 1—Jack for Lifting One End of a Car Clear of the Floor and Then Moving It to Enable the Mechanic To Get at Any Part Needing Attention

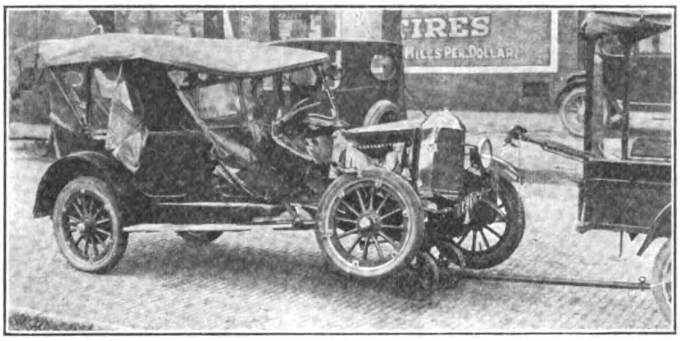

Fig. 2—Device for Bringing In Disabled Cars

One of the first problems that confronts the repairman is lack of floor space. When we consider that a car covers approximately 90 sq. ft. of the floor, we realize the necessity of contriving to overcome this. It was with this in mind that the jack shown in Fig. 1 was developed. This can be placed under the front and rear ends and, by caster wheels, the car can be pushed in any direction, thus enabling the mechanic to get at any part of it; he also can transport it to any part of the garage or the yard when desired. The next problem that arose was

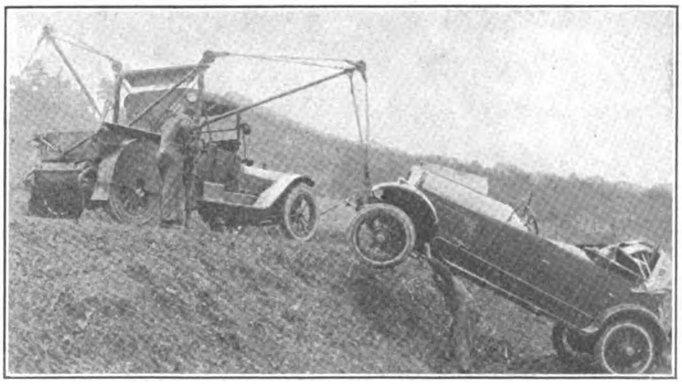

Fig. 3—Wrecking Crane Lifting a Car Out of the Ditch

[1] M.S.A.E.—Chief engineer, Weaver Mfg. Co., Springfield, Ill.

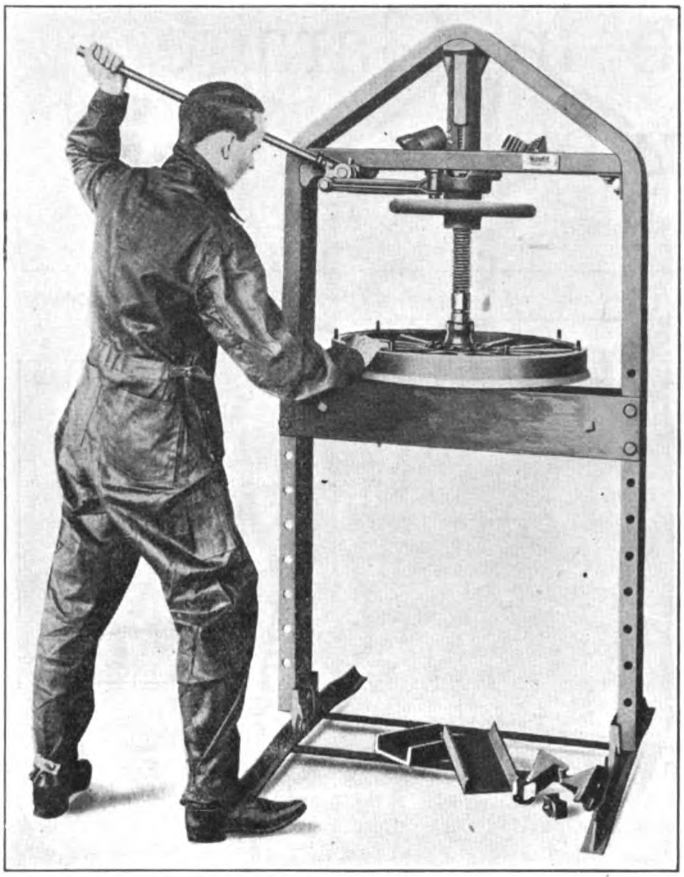

FIG. 4—PORTABLE PRESS OF THE SCREW TYPE FOR STRAIGHTENING BENT PARTS

FIG. 5—TIRE CHANGER THAT IS EQUIPPED WITH ATTACHMENTS FOR HANDLING PRACTICALLY ALL SIZES AND TYPES OF TIRE

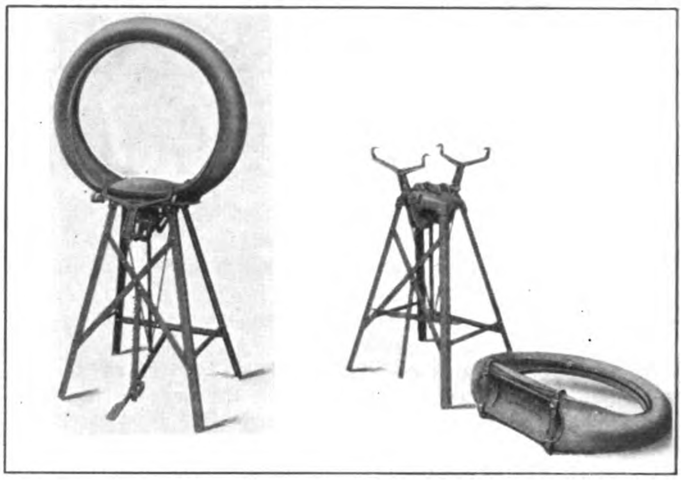

FIG. 6—TIRE SPREADER THAT NOT ONLY SPREADS THE CASING TO ENABLE REPAIRS TO BE MADE, BUT ALSO TURNS IT INSIDE OUT AT ANY DESIRED POINT FOR INSPECTION

the lack of a contrivance to bring in a disabled car, especially if a wheel or an axle were broken. After considerable experimenting, a two-wheeled device such as is shown in Fig. 2 was developed, having a long telescoping pole. It proved successful and thousands of wrecked cars have been brought into the garages by this means. Much development work has been carried on with the wrecking crane, and both very simple and very elaborate types have been brought out. This resulted from the difficulty experienced in rescuing ditched or overturned cars. Many garages are making a special feature of operating wrecking cranes to increase their repair business. Such a crane is shown in Fig. 3.

Straightening bent parts is another task requiring special tools. As the work of putting wrecked cars back into running condition increased, the garage owner felt the need of having the work done in his garage. The arbor press was not powerful enough and a press having the necessary power was too heavy and occupied too much floor space. Therefore, a portable press having the necessary power was designed, as illustrated in Fig. 4. This press is of the screw type and is adjustable, making it possible to handle either large or small work. It can be used in either a vertical or a horizontal position.

TIRES AND WHEELS

Few garagemen pay enough attention to the loss of time in changing tires. It is a common sight to see a

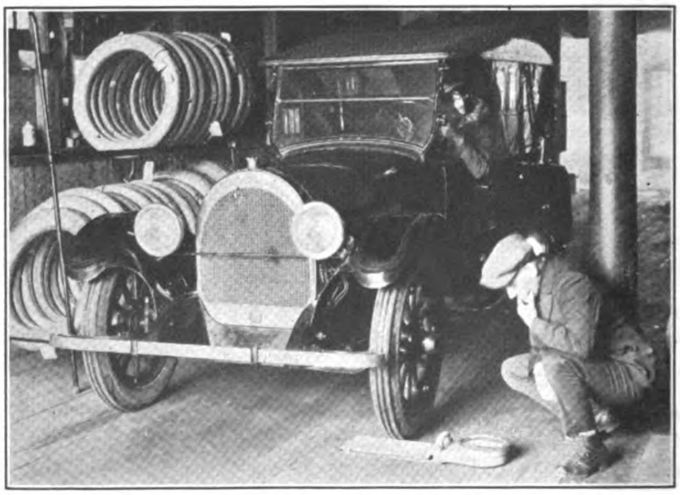

FIG. 7—WHEEL ALIGNMENT INDICATOR

rim battered and bent out of shape. A successful tire-changer, as shown in Fig. 5, has now been put on the market. It is supported on a stand of suitable height, and is provided with different attachments for putting on or removing the different types and sizes of tire. In touching upon the subject of tires, the development of tools and fixtures for the handling of tires brings home very forcefully the desirability of paying closer attention to the adoption of standards, especially where it is evident that replacements are necessary. The automotive engineer should have in mind two things if he expects his automobile to be popular; (*a*) he must adopt existing standards and adhere to prevailing designs

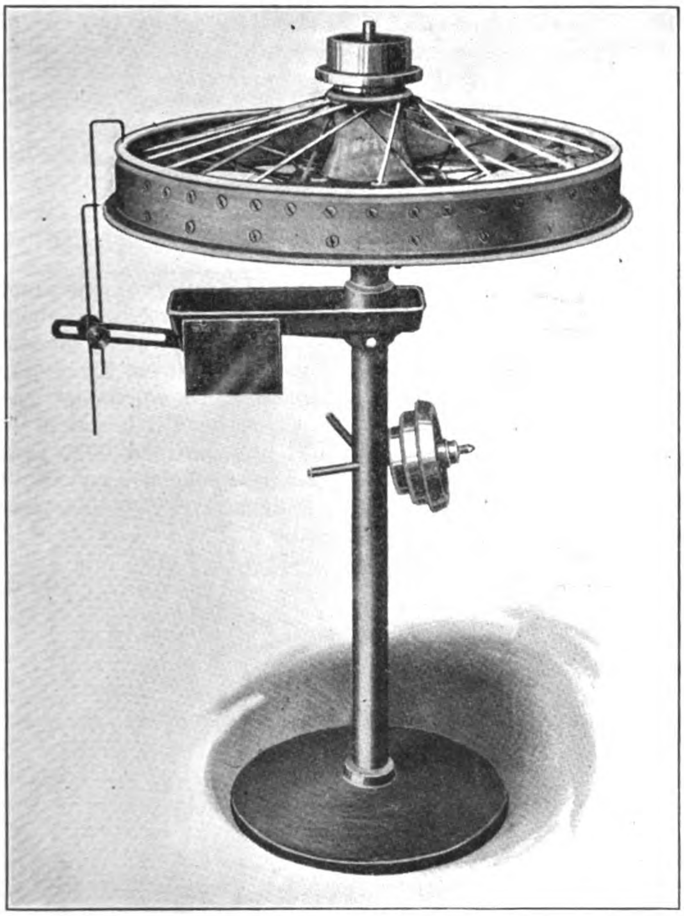

Fig. 8—Wire-Wheel Stand

wherever there is wear and (*b*) he must build his individuality around the features that, for ordinary use, need no replacement. For example, we have the clincher, the quick-detachable and the split-rim types of tire attachment. All of these are effective in holding the tire in place. Individually, any one of them would do the work, even if the others were not in existence. The garage-man is familiar enough with any one of the types, but his difficulty lies in his need of an inexpensive tool that will remove and replace the tire without requiring the adjustment of attachments to meet the needs of the different types of tire. Standardization of tire types will be a progressive step in making for better service.

After the tire-removing machine was brought out, it was seen that it often was advisable to inspect the inside of the casing when a tire had been removed. However, the casing is very stiff and it requires some effort to examine it properly. Fig. 6 shows a tire spreader that, in addition to spreading the casing for repairing the

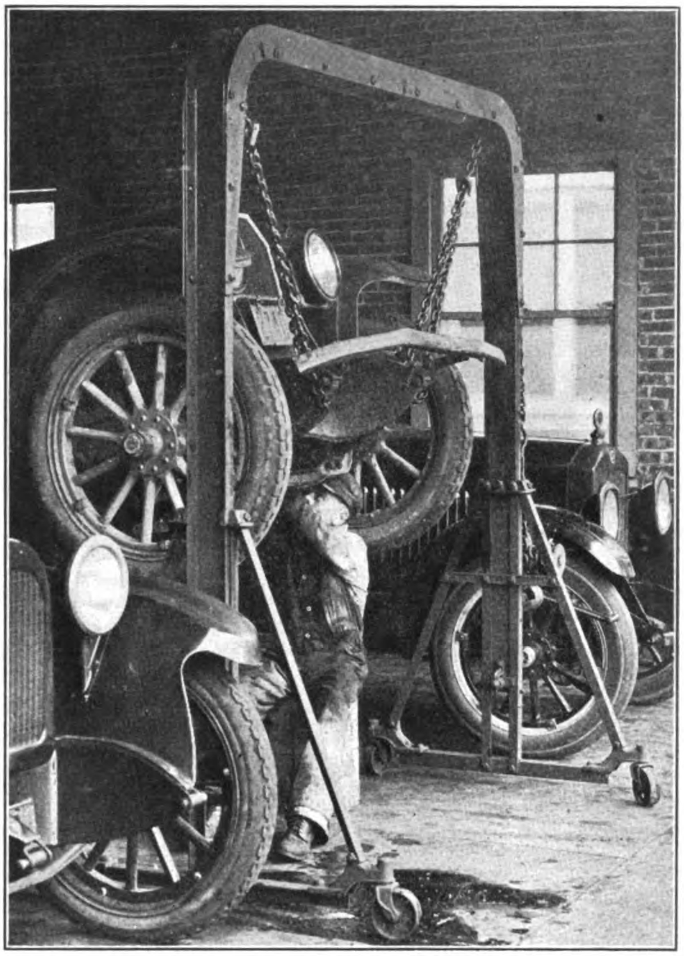

Fig. 9—Portable Hoist That Will Do Practically Everything That a Traveling Crane Will and in Addition Can Be Pushed Aside with the Car Suspended in the Air if Necessary

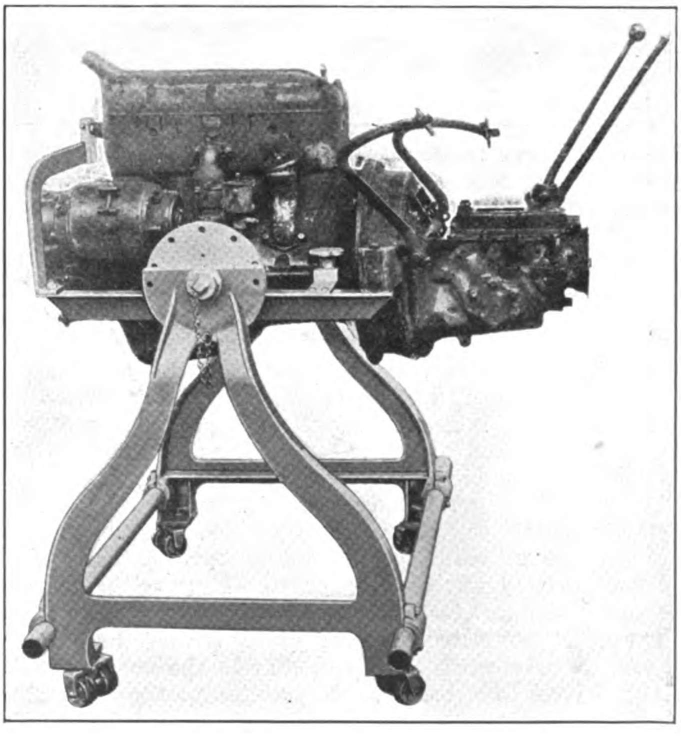

Fig. 10—Stand That Facilitates Engine Repairing

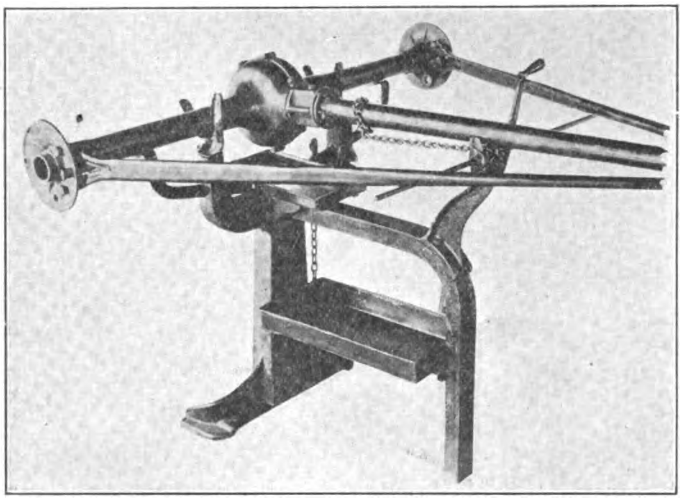

Fig. 11—Stand for Holding an Axle While Repairs Are Being Made

tire, has the advantage of permitting one to see the condition of the casing. Besides spreading the casing, the machine serves to turn it inside out at any desired point. Hundreds of rim removers and tire spreaders have been put on the market, and this points out the need of this equipment to the repair-man.

How many car-owners, or salesmen even, know why one tire will often wear out much more quickly than the

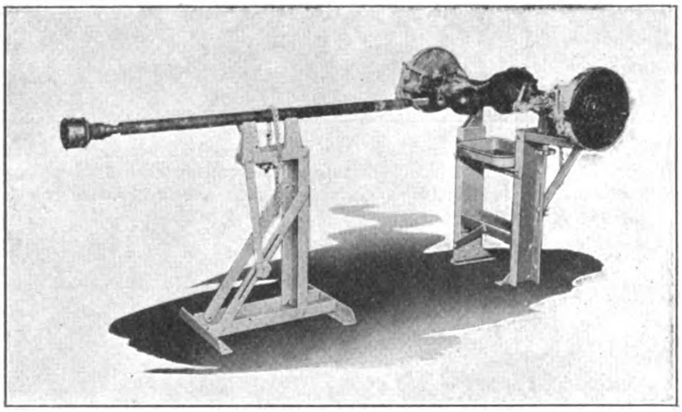

Fig. 12—Another Form of Axle Stand

other three? How many know how to correct this condition? The first time one will doubtless think the tire is defective. How much better it would have been if, by some simple method, one could have discovered the trouble and remedied it. An indicator to show the misalignment of wheels has been developed, which is shown in Fig. 7. By running the car over a plate that is mounted on rollers and connected to an indicator, an incorrect setting is gaged. By straightening the front and rear wheels, and sighting across from the rear to the front, the trouble sometimes can be seen very easily. The trouble often is due to the unusual wear in the steering-knuckles and connecting links. In the process of developing the indicator, it was discovered that a well-known factory was sending out machines with the rear axle out of line, causing unusual wear on one of the tires. The dealer corrected the fault and informed the builder.

A few wire-wheel stands similar to the one shown in Fig. 8 have been brought out because of the increasing number of wire wheels on the market; the service importance of such a stand has not been appreciated fully. The wire wheel is another example of extravagant waste of service energy and, because of lack of standardization, it is suffering unpopularity, not alone from the owner's standpoint but from that of the service-man as well. This is due to the spoke, a simple little wire upset on one end and having a thread on the other. Before the wire wheel will enjoy the popularity it deserves, every wheel maker must adopt a standard spoke, with the same "crooks" and the same lengths for different sizes of wheel. A good

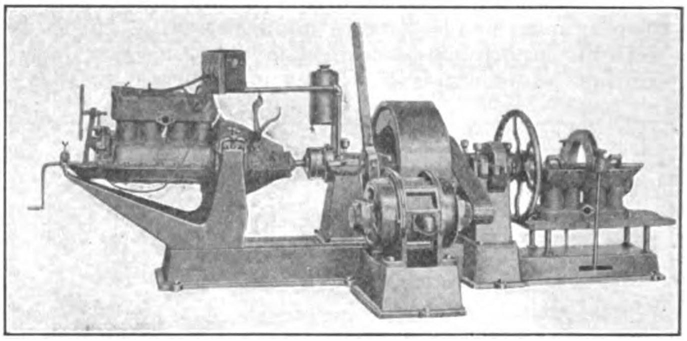

Fig. 13—Stand for Running-In New Engine Bearings

revenue can be obtained from replacing spokes and straightening wheels, but repair-men will not touch them because of the confusion of sizes and types of spoke and because of his difficulty in purchasing them for stock. The foregoing points are mentioned to emphasize the great need of standardization. The thought is not to do away with a car's individuality, but the idea is to construct all parts so that cars can have efficient service to the highest degree, through the agency of every service-man.

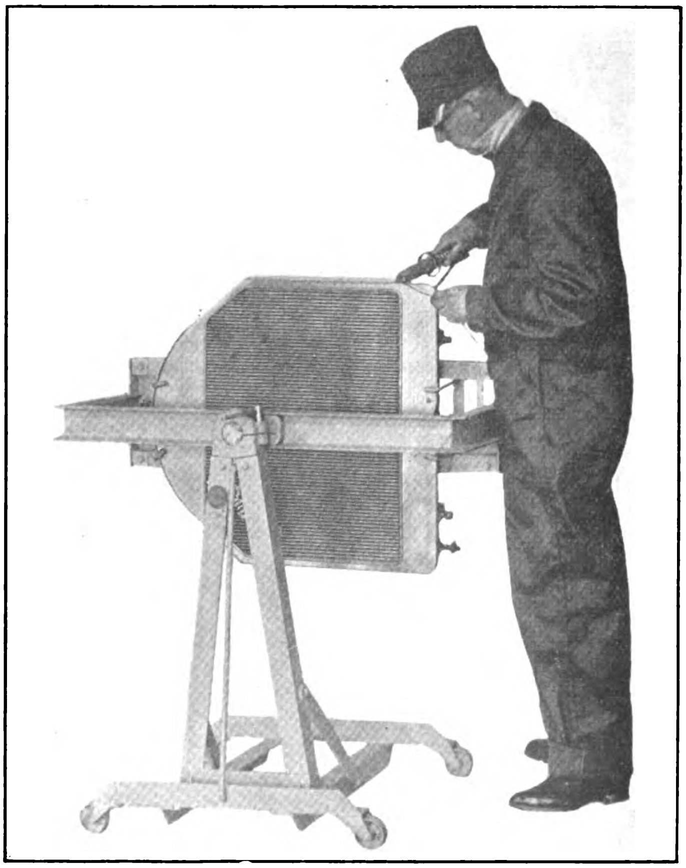

Fig. 14—Making Repairs to a Radiator Is Simplified by the Use of a Special Stand

Perhaps one of the most useful pieces of garage equipment is the portable hoist shown in Fig. 9. It performs every operation that a traveling crane can perform except the lifting of the whole car. It can be pushed aside with the car partly suspended in the air if the repair work is delayed.

Hoists and Jacks

In mentioning the portable jack, our attention has been called to the fact that a car should not be jacked up under the differential housing, because some axles are built of pressed steel or aluminum. Engineers who think that a car is not jacked up at this point will do well to realize the desirability of placing a special flat pad there; it is from this place that a car is always lifted. Another point is that of ground clearance of the front and rear axles. It seems that a standard minimum height should be established that will permit the design of a jack having a reasonable amount of lift. Our attention was called recently to a taxicab that had only a 7½-in. clearance from the ground and called for a special jack. The portable jack is used with hardly an exception in every garage. It is necessary for quick service. When there is

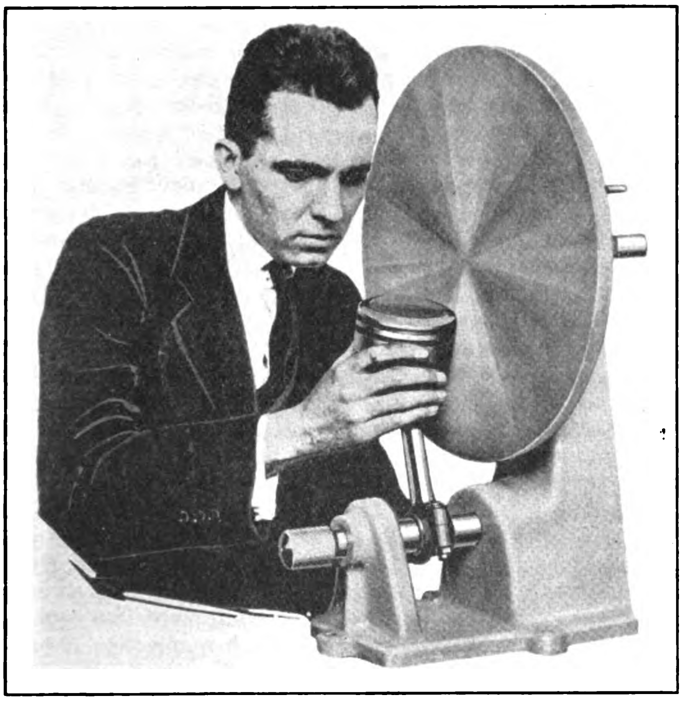

Fig. 15—Aligning a Piston and Piston-Rod

practically no room underneath for the jack, the delay in lifting the car is apparent. A jack built for the height previously mentioned, would not be practical for general use.

Engine and Axle Stands

The automobile engine has remained a mystery to the average man for so long a time that the designer still thinks he is privileged to keep away from conventional design. The engine designer does not appreciate the convenience of handling an engine in an engine stand. There are different numbers and sizes of cylinders, but it is important that a standard fastening be adopted. It would be well if an extra pad or arm could be installed at some convenient place on the crankcase. It would not cost any more, as it is done in the manufacture of engines to accommodate the tools and fixtures. There are a number of cleverly constructed engine-stands on the mar-

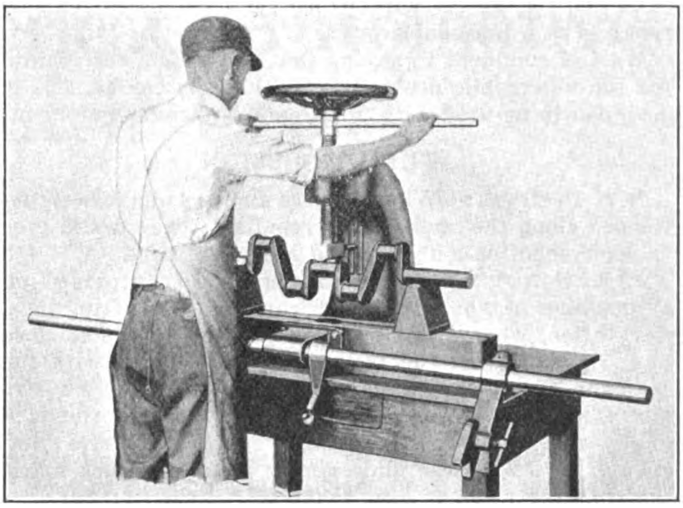

Fig. 16—Bench Press

ket, an example being shown in Fig. 10, which have increased the efficiency of overhauling and repairing engines wonderfully.

Another useful device that has come into general use is the axle stand, shown in Figs. 11 and 12. It holds the axles rigid while the work is being done on them. Garage equipment for many other purposes has been brought out, such as the running-in stand for running-in new engine bearings, shown in Fig. 13; the radiator stand, Fig. 14; the truck-wheel dolly for moving truck wheels; and various small tools and devices, some of which are shown in Figs. 15 and 16.

The application of oils and greases has received much attention from the service-men. Aside from the various makes of grease guns and machines, one of the most familiar pieces of equipment about the garage is the oil-bucket pump shown in Fig. 17. This pump is arranged so that it will draw out the old oil as well as supply new lubricant and, at the same time, measure the amount put in. As a rule, engineers have always realized the importance of accessibility in oiling a car, and it can be seen that there is an improvement in the oiling system each year.

Although I have only touched upon the progress made in garage equipment, I believe this to be an entirely new angle in approaching the subject. I wish to emphasize the necessity for standardization and that we wish to make it possible for the repair-man to obtain standard

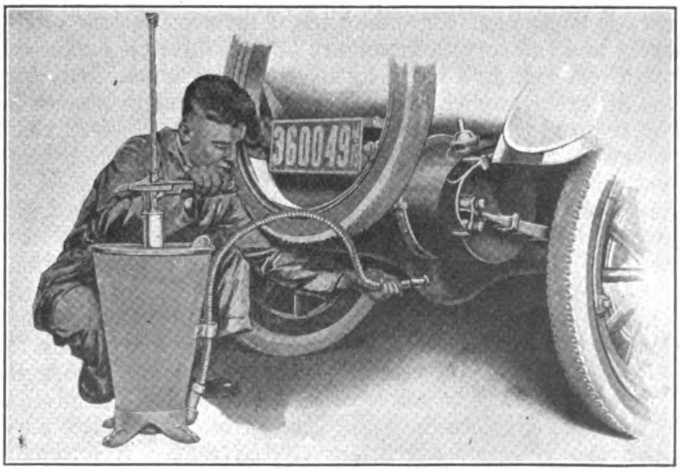

Fig. 17—Pump That Not Only Drains Used Oil from the Crank-case but Supplies and Measures the New Lubricant

tools and equipment with which all types of car can be repaired at a reasonable price.

We feel confident in saying that one of the best points for the automobile dealer will be that the car he sells is designed to be used with all standard garage equipment.

THE DISCUSSION

B. S. PFEIFFER:—What fixtures and tools have been developed along the line of Ford repairs? They would give us some good ideas about more general repairs.

J. E. SCHIPPER:—Some place should be provided on automobiles at which the owner can apply the lifting jack. Very often, when a front tire is flat, the axle is so close to the ground that the jack which is provided with the car cannot be placed under the axle. When a rear tire blows out, the average person will try to put the jack under the spring; the spring generally deflects just enough to let the car slide gently off of the jack about the time the rim has been taken off. Provision for lifting-jack application is one feature that has been overlooked. There ought to be some kind of a projection, so that a jack can be placed under it and the car raised high enough to facilitate changing the tire.

A. H. PACKER:—The question of maintaining the car or truck in operative condition depends upon the electrical as well as the mechanical features because, according to the records of the Chicago Motor Club, a greater percentage of the troubles which tie up a car are of an electrical nature. There is a need for good, low-priced electrical-testing equipment, and a need for education. Regardless of how the men can get the education, service-stations will have a difficult time in handling the service work with men who do not understand their work, particularly in regard to the electrical equipment. There are several electrical schools that are making it possible for electrical work to be performed intelligently. Therefore, the equipment lines should be extended to include magnetizers, testing equipment, test benches and armature testers. A number of high-grade manufacturing companies are putting out that type of equipment. Another type of equipment makes a spectacular test that means little, if anything. A man who ties up money in equipment of that kind will find it comparatively useless and, eventually, this will reflect against the makers of legitimate apparatus and increase the resistance that their salesmen meet. In developing or exploiting any device or apparatus for electrical or mechanical testing, I think it should be carefully analyzed to make sure that it is legitimate and intended to facilitate service.

W. C. ALLEN:—Having been engaged in the sale of garage equipment for some time, I believe that the engineers and the service managers of the different makes of motor car can gain much by selecting particular tools for the different applications that are made and recommending them to their dealers. The present figures show that somewhere near $68,000,000 worth of garage equipment was sold during 1921. I believe that 40 per cent of it has been discarded since. The car dealer or garage owner, in trying to create more efficient methods has bought different sorts of tools and, after finding that certain tools did not do the work, he discarded them. The previous remarks concerning good electrical-testing devices that are made to use and others that are made to sell apply also to the garage-equipment field. There is not enough cooperation between the car dealer and the builder in regard to the right kind of tools needed to perform the different operations in the repair of automobiles. If 40 per cent of this $68,000,000 worth of garage equipment that was bought last year has been discarded, somebody must pay for it, and this usually reverts to the consumer. There is much room for improvement in the selection of proper tools to give the customer the most economical service.

MR. PFEIFFER:—It is the lack of standardization between other makes of car that gives the Ford service-station the advantage over the ordinary garage-man. When a garage does repair work on 100 different makes of car, no two of which are alike in even major dimensions, it cannot install special tools and fixtures for each one. The Ford service-station can and does this very thing and, after a number of years, the company has developed a system of standardized flat prices for a job. The total investment in this specialized equipment is not large, and the company accomplishes wonders with its help. Its service organization is trained as it must well be. I recall a Ford dealer who used one man for 8 hr. to overhaul an engine. This consisted of taking the engine out of the car, tearing it down, reboring the cylinders, fitting new pistons and rings, grinding the valves and putting in new main and connecting-rod bearings; in other words, it was a complete job. How long would it take an ordinary garage to do the same job, even on a Ford car? Most of this equipment has been designed and developed by the Ford engineers. It seems that other companies should give this important matter their attention. Anything that enables the dealer to give better, quicker and less costly service will help him and, in turn, the manufacturer.

G. A. TOAZ:—Lifting jacks are selected usually by the engineers, and I think the responsibility should rest there. I know of one very recent instance in which a car stripped a tire. The driver could not replace the tire himself because the cogs on his lifting jack would not hang together. The interest the engineer has in the service end is not active enough. We must remember that we are not only selling cars but selling service. If we do not make proper provision for that service in the tools actually sent with the car, we are not doing what we should do. In selecting the tools to be supplied with the car, we should provide usable tools or omit them from the list entirely.

H. C. BUFFINGTON:—The company I represent builds this equipment because there is an actual demand for it. It has worked on many devices but it does not put them on the market until there is a demand for them. It was said previously that much of such equipment is not used, but there are many tools that go with a car that are not used. The closer we can approximate a general standard car, the easier it will be for every garage-man to handle the car. There will then be no need of a special service-station for every special car. The special service-station does not exist in small towns. In a small town, if a tool can be purchased that will do general work on all cars, it will sell.

B. M. IKERT:—Concerning the small-town garage, in a certain Western town of about 100,000 population, one service-station handles a very high-class car and did not get service orders on more than 20 per cent of the cars known to be in its territory. I could see the reason after taking one look at the shop and its service methods. It had very little equipment and no system whatsoever. Consequently, the owners were not taking their cars there. In another town of 2800 people, a man has a shop and the agency for one popular make of car. He can do everything in that shop but cut gears, as he has grinding, drilling and shaping machines. There are many tractors

(*Concluded on page* 80)

European and American Automotive Brake and Clutch Practice

By H. G. FARWELL[1]

METROPOLITAN SECTION PAPER *Illustrated with* PHOTOGRAPHS AND DIAGRAMS

THE author describes the major features of brake and clutch practice that he observed in 1920 while traveling in England, Belgium, Italy and France, comparing them briefly with American practice of the same period. He analyzes the types of brake and clutch used on 165 cars exhibited at the London automobile show of that year, giving the percentage of the different types in evidence.

Numerous illustrations that are described and commented upon in greater or less detail appear in the paper and in the discussion which followed it, these being inclusive of most of the best-known types of brake and clutch in use in the United States and in Europe.

BEFORE recounting the major features of European brake and clutch practice that I observed during several months of travel in England, Belgium, Italy and France in 1920, I will outline the general practice in the United States as it existed at that time. The ordinary type of external brake was in general use for service work; in Europe it is known as the foot-brake. What we call the emergency brake is known as the hand-brake in Europe because it is operated by the hand-lever; it is nearly always cam-operated in this Country. There were also some cases in which one brake, sometimes the service and sometimes the hand-brake, was used on the transmission; cases in which both sets of brakes were internal, on the rear wheel-drums; cases where these were in concentric drums; and, in some instances, the two brakes were placed side-by-side. Regarding clutch practice in the United States, we had seen a gradual reduction in the number of cone clutches used, until comparatively few cars are so equipped in proportion to the number of cars built. We have seen a gradual increase in the use of the disc clutch; the single-driven-plate and multiple-disc types being used in high-grade cars. One striking feature is the size of brake-bands. In this Country the diameter of the rear-wheel brakes usually is from 35 to 45 per cent of that of the rear wheels. It is not unusual to find 12, 14 and 18-in. drums on cars weighing from 2100 to 5500 lb.

ENGLAND

In England the use of the internal brake was fairly general on all classes of cars and, with comparatively few exceptions, the use of the transmission brake was rather more marked there than in this Country. In general, very little criticism was found on the operation of these brakes; but there is one exception in the case of the Daimler car which, until recently, has always used an external service-brake on the rear-wheel drums. I understand that the company is planning now to change that to the conventional internal expanding type. On several of the post-war cars weighing 1600 lb. and even less, they were using external band brakes. The objection made in England to the use of the external brakes is that they always drag. Their engines are very much smaller, for fuel-economy reasons; any dragging of brakes uses up a greater proportion of available engine power than is the case with cars in the United States.

The usual English criticism of American brakes is that they are intolerable, although I found about as many unsatisfactory brakes in England as here, on from one to five makes of car, depending upon the locality in which they are used. The brakes on city cars generally are maintained better than the average brake used in the country districts. Probably more attention has been given in England to brake design and the construction of internal brakes than here; in fact, that is true throughout Europe. The main objection to the usual internal brake is that wear is confined to a very much smaller area than is the case with external brakes. To compensate for this, the English designer arranges the adjustment feature so that brake adjustment can be made much more easily, often without getting out and going under the car. In the English models that have been produced within the past two years, the ease of brake adjustment has received great attention. In some cases large hand-wheels are placed so that one can adjust the angle of the cam with but little more difficulty than that of getting out of the car.

I had been led to believe that metal brakes were always noisy, but I found this to be an exaggeration. Except for the metallic grind, these internal brakes were only slightly more noisy generally than the American conventional type of external brake. I attribute this to better or closer fitting and to the elimination of points where vibration may occur. One finds noisy brakes in England, the same as here; but, in general, they are not much worse there. Brake-drums are very much smaller in England. To cite one case, an American car weighing approximately 2100 lb. was equipped with 31 x 4-in. tires and 14-in. brake-drums. The English car, having exactly the same size of engine, weighed 2800 lb., had 30 x 1½-in. tires and the diameter of the brake-drums was 10 in. After looking over the field, I am sure that the tendency in this direction is to increase the diameter of the brake-drums. Probably tire sizes will always be kept smaller, for the sake of economy and first cost, but the brake-drum diameters are certainly growing.

The English clutch situation was interesting in that it showed so many cone clutches on cars of the higher grades. The single-plate clutch also is used, but not to the same extent as here. There is a noticeable tendency to consider the use of the plate clutch, instead of the cone type. It will be surprising to see the great number of plate clutches that will be used in the years to come. There is less need for large-capacity clutches in England than there is here. That has a bearing on the use of the multiple-disc clutch. It is interesting to note some of the tendencies at the automobile show held in London in

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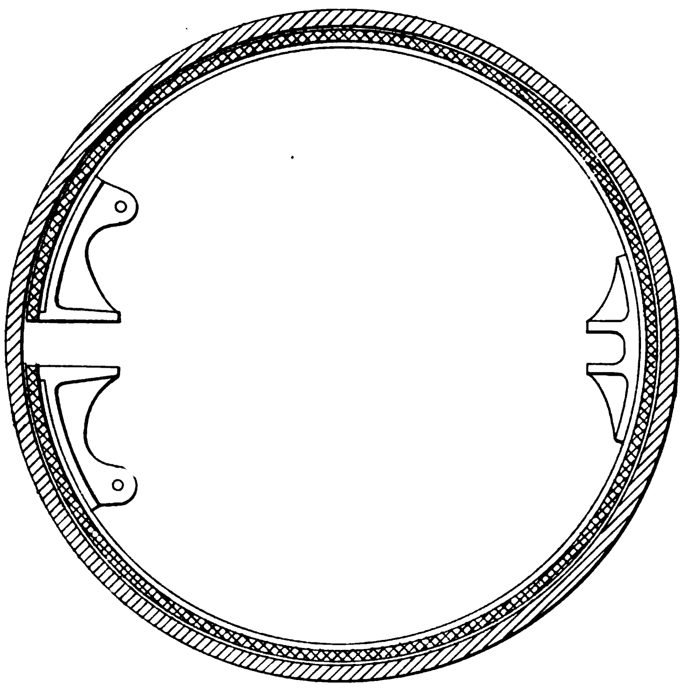

FIG. 1—CONTINUOUS-BAND TYPE OF BRAKE

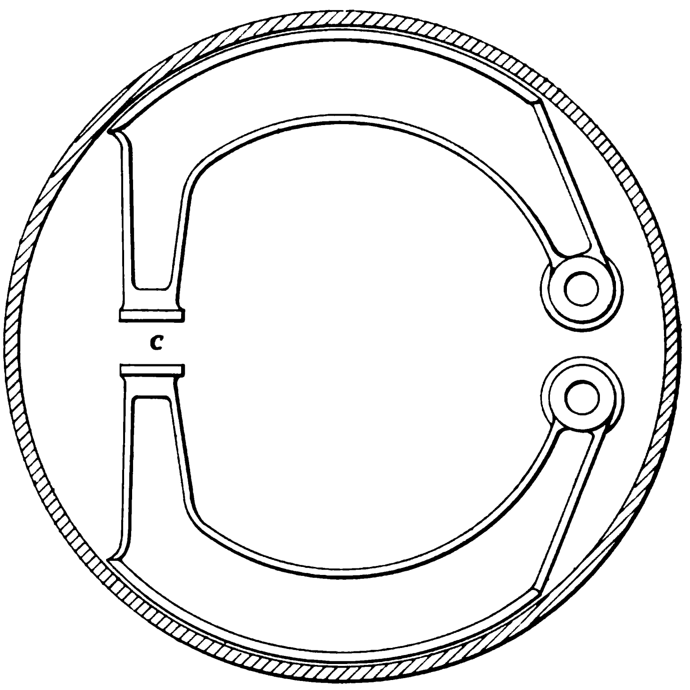

FIG. 3—A EUROPEAN INTERNAL BRAKE WITH TWO OPERATING PIVOTS

November, 1920. I have not checked these figures with those recently published; they are simply the result of my personal observation.

Seven nations were represented by 165 exhibits, Switzerland had 1; Holland, 1; Belgium, 3; Italy, 7; France, 28; the United States, 30; and England, 95. Of these 165 exhibits 49, or practically 30 per cent, were equipped with clutches of the single-plate type. There were 28, or 17 per cent, with clutches of the multiple-disc type. Only 4, or 2.5 per cent, had friction drives; that is, having the disc bearing on the ring, about 1 to 1½ in. wide. This leaves 83 cars, or just over 50 per cent, that were equipped with cone clutches and with linings and leather, cotton or asbestos. Of the 95 English cars 56, or 59 per cent, had cone clutches. There were 11, or 12 per cent, with clutches of the multiple-disc pattern and 24, or 25 per cent, had clutches of the single-plate variety. The only friction-drive clutches of the 165 exhibits were of English manufacture.

Of the French cars 61 per cent had cone-type clutches, 18 per cent were of the plate and 18 per cent of the disc type. The only band clutch was of French make. Of the 30 American cars, 50 per cent were equipped with single-plate, 27 per cent with multiple-disc and 23 per cent with

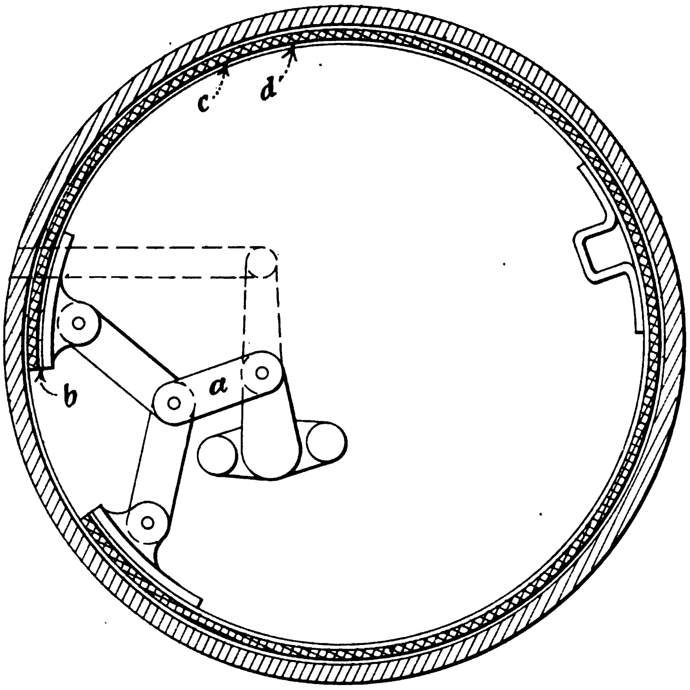

FIG. 2—ANOTHER INTERNAL BRAKE THAT IS OPERATED BY TWO LINKS

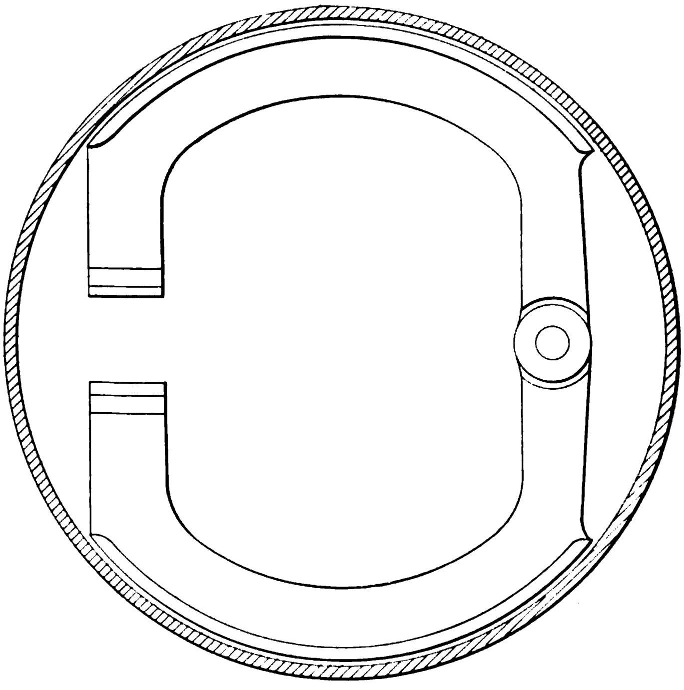

FIG. 4—IN THIS EUROPEAN BRAKE ONLY ONE PIVOT IS REQUIRED FOR OPERATION

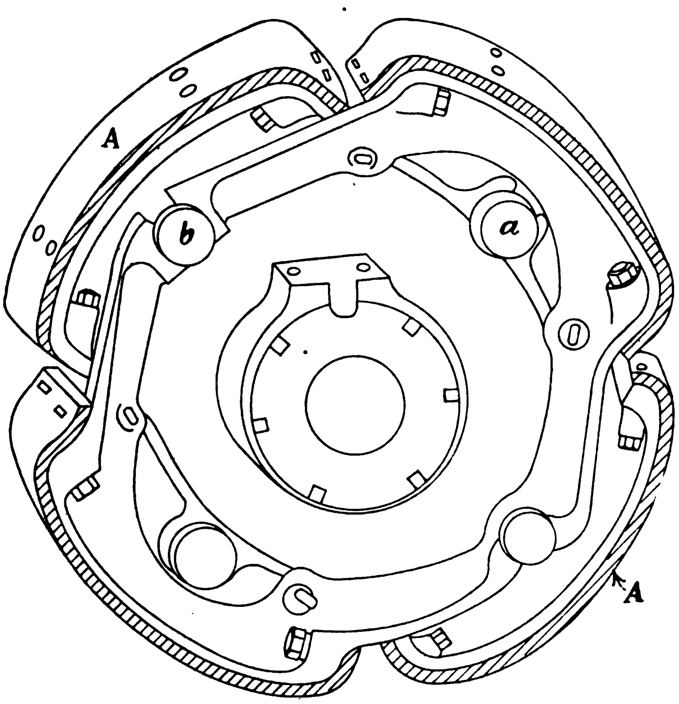

Fig. 5—A Double Internal Brake in Which Not More Than One-Quarter of the Drum Circumference Is Covered

cone-type clutches. With one or two exceptions, all the 30 American cars used the external brake for the service or foot-brake. Perhaps four or five used a transmission brake but, in general, the usual external and internal sets were used. With two or three exceptions, the English cars used internal brakes; most of them had trans-

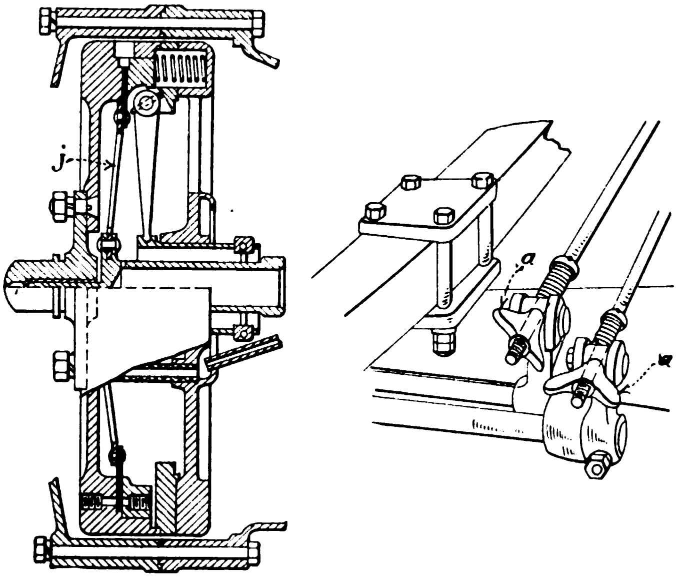

Fig. 6—Two Examples of European Design
The View at the Left Shows a Clutch Having the Facing on Both Sides and the Springs at the Periphery While the Mechanism Employed for Operating the Brakes on a Belgian Car Is Illustrated at the Right

mission brakes also. The Continental cars stand about the same as the English with respect to brakes, but we often find the external brake used on the transmission.

Continental Countries

The brake situation in Belgium is very similar to that in England; the internal expanding type is used almost universally and generally the brakes are metal-lined. There are cases where an asbestos liner is used and usually the brakes are fitted so that it can be used if so desired. Here also we find a considerable interest in the disc type of clutch. Of some eight or nine firms, there is a greater proportion that builds an engine that rates over 15 hp. than in England, although the actual number is really less. While those who use a cone clutch declare that they have no trouble with it, they are still a bit more than willing to try a clutch of the plate or multiple-disc type.

In Belgium we find one of the strong proponents of brakes on four wheels. The operating practice of one firm varies in a very interesting way from some others, as will be illustrated later. We find the internal brakes on the front-wheel drums usually operated by cams at the tops of the drums. This operating cam is on one end of the shaft that carries the lever and is fastened to the side-member of the frame. This shaft is furnished also with a flexible joint which allows for the steering movement of the wheels and carries the cam ring of the shaft. In the Belgian construction mentioned the brake is oper-

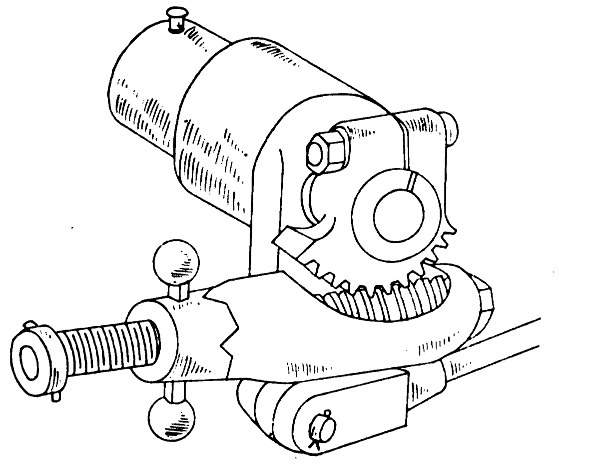

Fig. 7—A Form of Brake Adjustment in Which the Angularity of the Camshaft Is Changed

ated from the bottom and, by placing a specially shaped cam in the actual line of the steering-knuckle, carrying the operating shaft around with the wheel during the steering movement is avoided. I think there are only eight or nine cars built in Belgium, so that the field and the differences of practice are somewhat limited.

In Italy we find the internal brake and a rather larger proportional use of the transmission brake than in some of the other countries. The center of the industry in

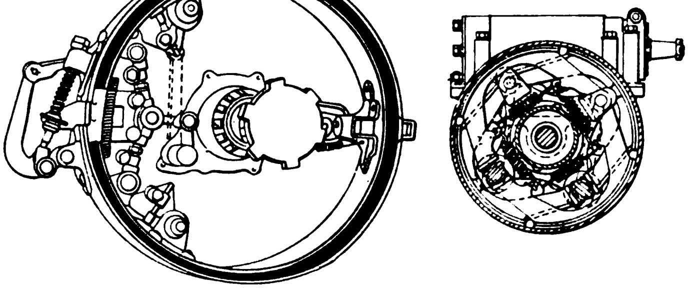

Fig. 8—Two Forms of American Truck Brake
At the Left Is a Simple Type of Toggle Brake and in the Brake at the Right a Certain Amount of Movement of the Cam Lever Is Necessary before the Lost Motion between the Shoe and the Drum Is Actually Taken Up

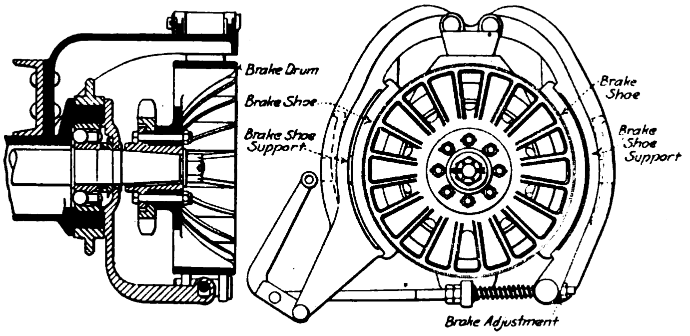

FIG. 9—AN EARLY FORM OF BRAKE IN WHICH AN EFFORT WAS MADE TO AIR-COOL THE DRUM

Italy is in the north, around Milan and Turin. Perhaps on account of the physical characteristics of the country, their engines are larger on the whole and their cars heavier and more rugged. Brakes are considerably more closely watched there than in England. Metal liners are generally in use. Brake-drums follow the other practice closely. We find brakes on four wheels in two or three cases. This feature is often made optional with the purchaser, at a price. The cone clutch is surely going out and more and more of the multiple-disc type are being used. An interesting feature of the multiple-disc type of clutch in Italy is that the diameters of the plates are considerably greater than those of the American clutches of the same type. The general practice in multiple-disc work here runs between 7 and 9 in., while the multiple-disc clutches of the Italian cars will run between 8½ and 10 in. Most of the multiple-disc clutches have liners of some kind. In one case cotton was used.

France is one of the most interesting centers of the automotive industry as regards new applications of ideas. We find the use of the internal brake on the rear wheels almost universal there; and the drums are being increased in size considerably. There is a peculiar intermixture of the use of internal and external brakes; internal brakes on the rear wheels and an internal brake on the transmission, or an external brake on the transmission and internal brakes on the rear wheels. Some of the larger cars have brakes on four wheels. Usually they are listed as "extra." The matter of applying brakes and their operating connections to the steering-knuckles offers some peculiar difficulties. The one interesting application is on the Hispano. The brakes on the four wheels are operated by an auxiliary clutch that is brought into action by the foot clutch rather than by spring action. This auxiliary clutch operates on a shaft that is driven by a worm and gear from the transmission shaft. Considerably greater attention is given to the equalization of brakes, as a rule. Anyone who has used or driven a car equipped with brakes on its four wheels cannot help but be impressed by the ease with which a car is controlled and the quickness with which it can be brought to a sudden stop. By "sudden" I mean a stop that will bump one's head against the windshield.

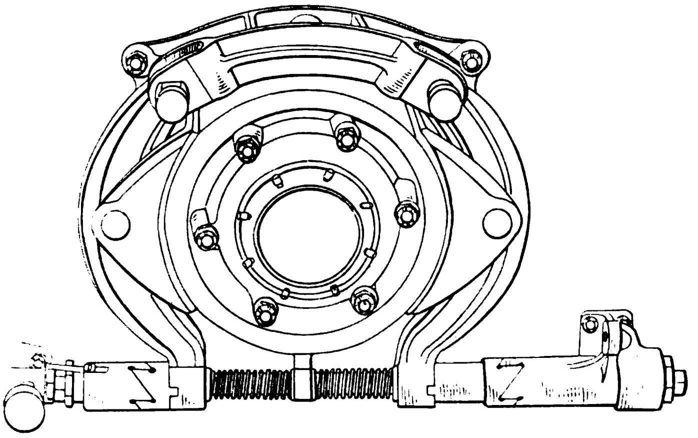

FIG. 10—THE PIERCE-ARROW BRAKE USING A RADIAL CAM

France still uses many cone clutches. Several multiple-disc and some single-plate types are used. One very interesting clutch is of the single-plate type; the clutch facing is required to do double duty, both sides being used. The facing is fastened to the metal plate that fits snugly inside the center hole of the facing. Another clutch that shows some ingenuity is one in which the shaft on which the driven member moves has been eliminated. The requisite motion for release is obtained by the buckling or dishing of the metal driven plate.

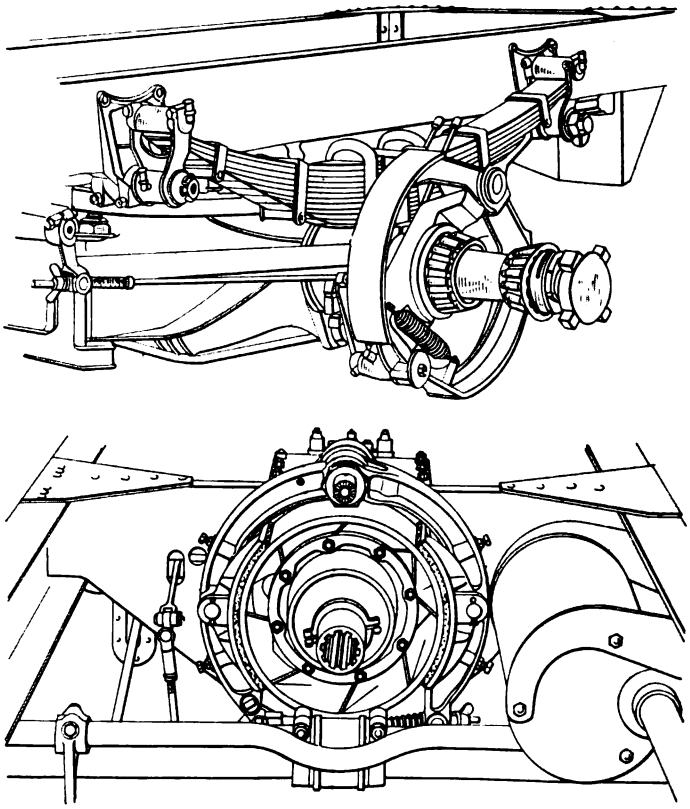

FIG. 11—TWO VIEWS OF THE BRAKE USED BY THE PACKARD COMPANY

European practice is so different from our own that we cannot criticise; it is based on economy, speed and comfort. The designers in France, England and Europe in general have just as much reason for their designs as we have for our own. French engineers tell me that French automobile body lines today are following those of American cars more than ever before. They also are following our practice of unit powerplant construction in some cases, but we find that the physical characteristics of the country and the gasoline price have a bearing upon European design and, when we criticise, we should take those factors into consideration.

BRAKES

Fig. 1 shows a continuous-band brake. One of the great difficulties with this type is the excessive wear on the cam ends. The back usually becomes clogged with mud in spite of the fact that it is an internal brake, and

it often refuses to operate. Fig. 2 shows another brake operated with two links, and this practice is subject to the same trouble. The pressure at *a*, instead of being entirely tangent at *b*, is nearly radial. The wear is on the facing at *c* and there is practically no wear at *d*.

Fig. 3 shows one of the internal brakes used in Europe, and unusual in American practice; it differs in having the two pivots at *a* and *b*, with the usual cam-operating

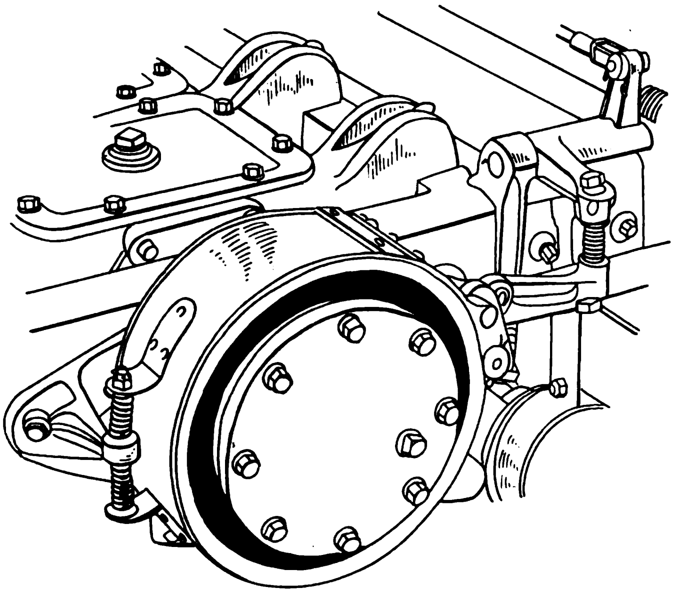

FIG. 12—BRAKE EMPLOYED ON A FOUR-WHEEL-DRIVE TRUCK

mechanism at *c* except that it is brought back from the circumference of the drum. Fig. 4 shows an internal brake of the same general type, except that it has only one pivot at the back. In two or three cases the brakes are operated by a wedge instead of a cam. Fig. 5 shows a double internal brake having only about one-quarter or less of the circumference of the drum covered. One set of brakes, *A*, is operated from the cam *a* and the other set from the cam *b*. The view at the left of Fig. 6 is a diagram of the clutch, in which the facing is used on both sides, with springs on the periphery. It depends simply on the buckling of the plate *j* to give the necessary motion for release. It is a De Dion clutch. Some of the mechanism for the operation of the front-wheel brakes on a Belgian car is shown at the right of Fig. 7. The cam *a* is in actual line with the steering-knuckle. The cam-operating shaft is brought back and operated

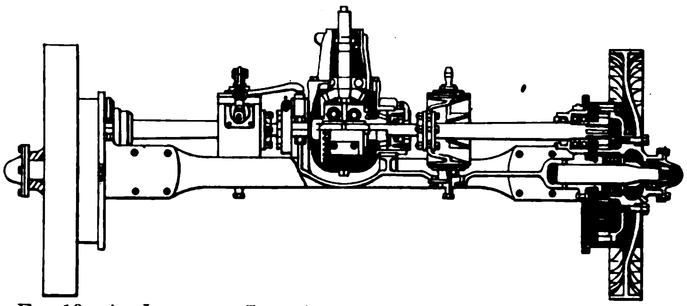

FIG. 13—AN INTERNAL-GEAR AXLE HAVING SUPERPOSED AUXILIARY OR PINION SHAFTS

from the front axle. The cam *a* is spherical, so that it allows the steering of the wheel without interfering in any way with the motion of the operating shaft. The Delage and one or two of the others operate their front-wheel brakes by a cam and a flexible connection from the frame. A universal-joint connects the cam with the operating shaft, the other end of which is held on the frame by a ball-and-socket joint. This construction allows the shaft to take any angle within the limits of the springs and still give no rotating motion to the cam.

In one of the English cars the thumb or wing nuts are placed at the back of the axle so that one has access to them without very much difficulty. Fig. 7 shows another one of the brake adjustments; it is a sector of a worm wheel and worm. The worm is turned by the ad-

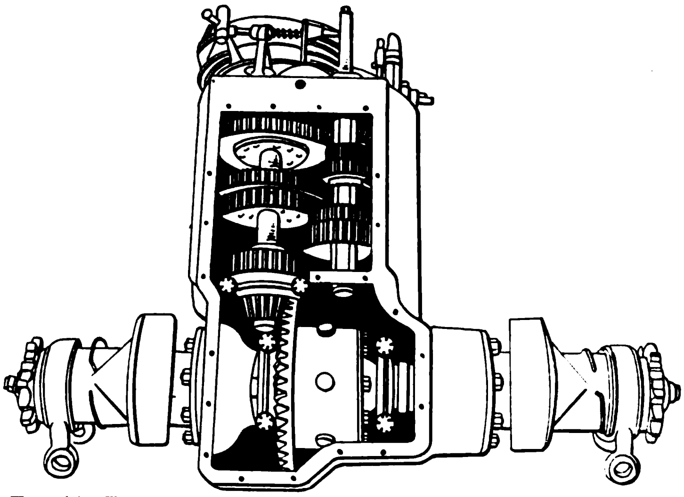

FIG. 14—TRANSMISSION FOR THE GARFORD TRUCK IN WHICH THE BRAKE BAND IS LOCATED AT THE FORWARD END OF THE COUNTERSHAFT

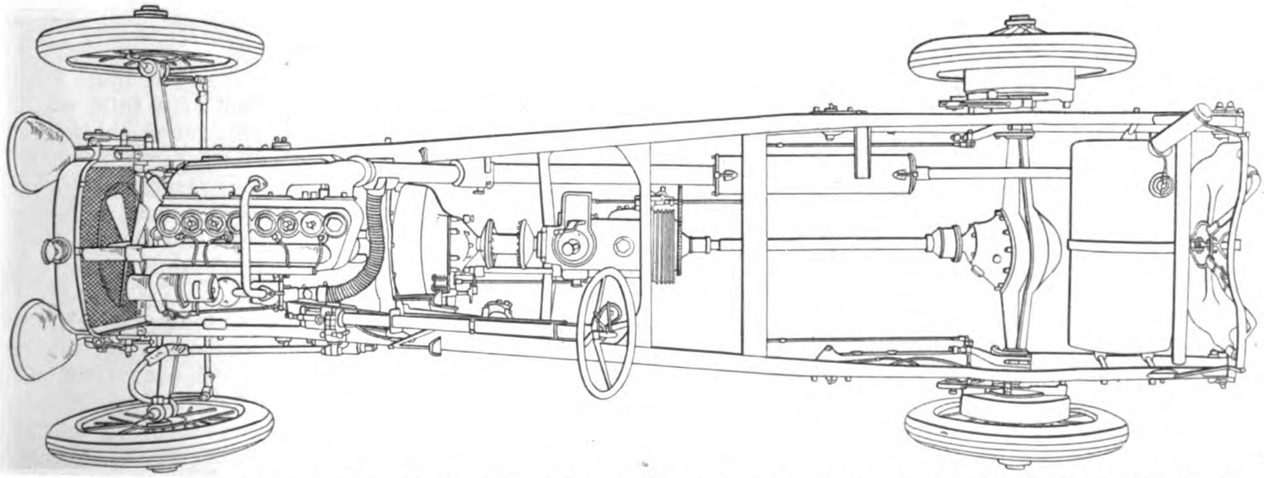

FIG. 15—A PASSENGER CAR CHASSIS HAVING A VERY SIMPLE BRAKE LAYOUT IN WHICH THE BRAKE CAMSHAFT PASSES THROUGH THE TRANSMISSION HOUSING AND OPERATES AN INTERNAL CAM ON THE BRAKE BACK OF THE TRANSMISSION

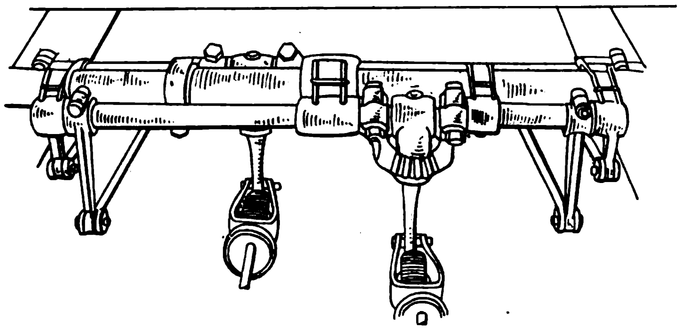

FIG. 16—A TRUCK-BRAKE CONTROL EQUIPPED WITH A DIFFERENTIAL EQUALIZER

justment *a*, drawing the rod along and so changing the angle of the camshaft *b*. The main objection to all these devices for changing the angularity of the cam is that, by changing the leverage, a point is reached where the wear of the brake is increased and this method is no longer effective.

Fig. 8 shows a simple type of toggle brake at the left and at the right is a Timken truck brake. In the latter it is interesting to note the position of the cams; there is a certain amount of movement of the cam lever before the cam actually takes up the lost motion between the shoe and the drum.

In an internal-gear axle of the Torbensen type one of the brakes is mounted on a pinion shaft. There is a cam action that operates the brake rather than the toggle. The large steel wheels provide a good anchorage for the brake-drums.

The Russell internal-gear axle is interesting in that it represents an attempt to enclose a brake-band. It is an external brake and has an enclosure plate that comes partly around the external brake-band. This is used also to keep dust out of the internal gears, but the same company also uses this axle on a passenger-car type. Fig. 9 illustrates the old Knox tractor, showing the attempt made toward air-cooling of the brake-drum. Fig. 10 shows a Pierce-Arrow brake, using a radial cam, and Fig. 11 a Packard brake.

Fig. 12 shows a four-wheel-drive truck. It seems odd to have a brake-drum upon one side with nothing back of it. The chain case is immediately ahead of the brake unit. Fig. 13 is an internal-gear axle with superposed auxiliary shafts or pinion shafts; the brake units are easily accessible and are mounted adjacent to the differential case. This same construction is used on some of the Kelly-Springfield trucks. Fig. 14 is a transmission from a large Garford truck, with the brake-band up at the front end of the countershaft. There is no direct drive on this transmission, the gearing and brake being similar to those on the old Mercer car.

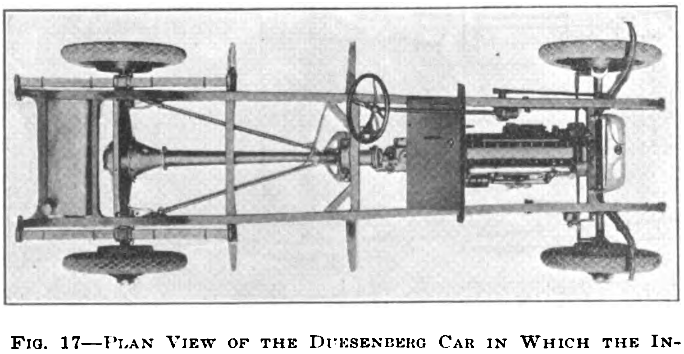

FIG. 17—PLAN VIEW OF THE DUESENBERG CAR IN WHICH THE INTERNAL BRAKES ACT ON ALL FOUR WHEELS AND ARE ACTUATED HYDRAULICALLY

Fig. 15 shows the Mercer chassis, which has a very simple brake layout. It has a brake camshaft that passes through the transmission housing and operates an internal cam on the brake back of the transmission. A wheel projects out from that shaft and the connection with the clutch pedal is made to the projecting lever by a small interconnecting link. A small thumb-screw at the top forms a very satisfactory adjustment. Fig. 16 illustrates the Atterbury brake-control unit, showing the differential type of equalizer and a method of keeping lubricant on the brake shaft; it has a very accessible hand-wheel brake-rod adjustment. The Lafayette equalizer is of the differential type on both sets of brakes and is immediately back of the gearbox. The designer of this chassis prides himself on not having a crooked rod on the car.

In the Panhard chassis a steel strip, having a series of holes in its rear end, is used instead of a rod or cam. This affords a very easy adjustment at the front end. The Lancia chassis for years has had the roller-chain links around the brake-drum to secure as nearly as possible an equal contraction all the way around. This principle is employed also on the Sunbeam car which has a pulley on the end of the lever. The brakes are operated through a cable that passes around one pulley, up and over another pulley. The cable running to the other side goes around the pulleys in the same way and back. By turning a thumb-screw all slack can be taken out of the brakes; this provides an equalizing effect. The Sunbeam builder developed this method through racing. Mr. Resta told me that it was found necessary to adjust the brakes during races and, to provide a scheme whereby the mechanic could do this quickly without getting out of the car, this method was the result.

The brake lever in the Delage car comes up in the center of the car and goes forward to the front brakes. The brake cam operates at the top and the shaft is flexibly mounted on the frame and universal-joint. The Renault company has brought out a differential brake-equalizer for two brakes on the rear wheels; as well as an equalizer in which there is a worm adjustment. The brake-shaft continues through and the clutch-tube floats on it. A sector of a bevel gear that merges with another bevel gear on the brake camshaft is mounted on one side. This brake camshaft is similar to that on the Mercer car. Fig. 17 shows a plan view of the Duesenberg car. This is another method of operating brakes, in the form of a flexible tube which has hydraulic actuation. These brakes are all of the internal type; they are four-wheel brakes.

THE DISCUSSION

W. D. REESE:—Innumerable problems must be solved in connection with the production of a safe, economical and efficient form of motorbus. Among these problems the question of brake design is certainly the most formidable. Brake failures, irrespective of vehicle type, must be vigorously guarded against, but, of course, in such cases with a bus the potential hazard is much greater on account of the larger number of persons carried.

Mr. Farwell has intimated that comparatively few improvements in brake design have been made during the past decade and that while brakes are fairly efficient, improvement in design has not kept pace with the other units in the automobile. In a general way, Mr. Farwell's statements appear to be correct. At the same time, we believe that the design of brake employed on our buses is extremely satisfactory. But this does not mean

that we are unwilling to admit the possibility of improvement. As we see the situation, the fundamentals of good brake design from the standpoint of public service requirements are as follows:

(1) Safety
(2) Simplicity of adjustment
(3) Maximum service between adjustments
(4) Low upkeep-cost
(5) Freedom from loose parts and consequent rattle
(6) Ability to readily dissipate heat

During 1920 the Fifth Avenue Coach Co. carried approximately 50,000,000 passengers, equivalent roughly to half the population of the United States, and operated buses traveling 9,000,000 miles, which represents a daily mileage sufficient to encircle the earth. According to the statistics of our transportation department, this necessitated approximately 36,000,000 brake applications for passenger and traffic stops, or an average of four applications per mile. This does not include the applications made while running down grades, which would increase the total number of applications by several million. Approximately 10,000 ft. of fabric brake-lining supplied by various manufacturers was used during the year.

We have tested a very large number of different brakes in various ways and excellent results have been obtained from those now standardized on our Model A bus. For example, tests were made with buses weighted with sand-bags to the equivalent of a full passenger load to determine the maximum braking that could be obtained with normal effort on the part of the driver. Many tests were conducted on Broadway, New York City, between 136th and 150th Streets, making runs in each direction with dry road-surface conditions and accurate data were arrived at by a recording device consisting essentially of four electromagnetically operated pointers, a time-marker clock and a contact-making device mounted on the hub on one of the rear wheels of the bus. One of the pointers was actuated by the time-marker clock to indicate 1-sec. intervals, another by the contact-making device on the wheel to indicate the number of revolutions and two other pointers by push buttons to indicate the length of the braking period on each brake. All stops were made without skidding the rear wheels. The results of a large number of tests showed that with normal effort on the part of the driver the deceleration obtainable was 3.75 m.p.h. per sec.

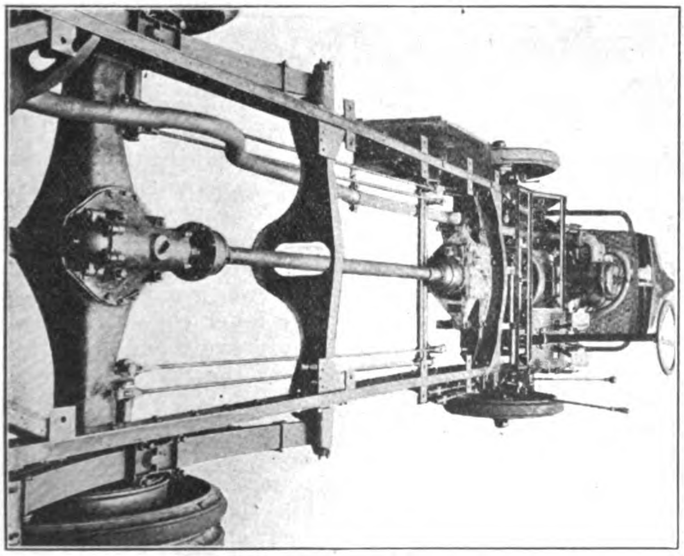

FIG. 18—BRAKE USED BY THE FIFTH AVENUE COACH CO. ON SOME OF ITS BUSES

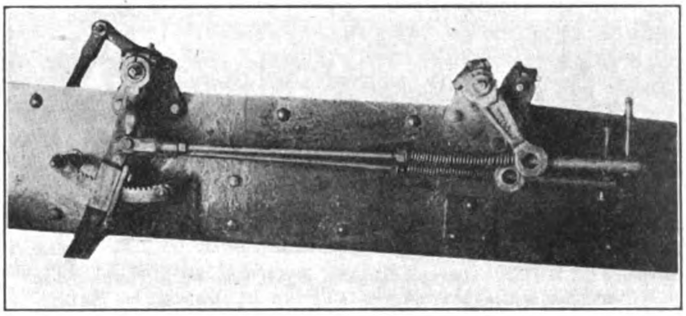

FIG. 19—A TYPE OF BRAKE ADJUSTMENT THAT ELIMINATES THE USE OF TURNBUCKLES

Fig. 18 shows the brakes on the Model A bus. The brake-pedal is operated in a conventional fashion by the right foot and the hand-brake by the right hand. Pressure applied to these members is converted by suitable linkage through a pull on the rods leading back along the side of the frame to the cross-shaft. At this point there are hooked up four rods running to the cam-actuating levers on either side. No equalizers are used since they are unnecessary when only one point is made use of for service adjustment.

It will be realized readily that in bus work, especially with a Hotchkiss type of drive, a rather difficult problem confronts the engineer who attempts to design a brake-operating mechanism, especially when we consider the tremendous deflection and consequent axle movement that are necessary if we are to have a vehicle that rides with the minimum amount of discomfort to the passengers and at the some time assures a perfect-acting brake under either the full or unloaded condition. To take care of the deflection we use a center cross-shaft which permits of a comparatively long rod to the cam operating lever. The position of the cross-shaft and the length of the levers used have been determined as being the best combination of theory and practice obtainable after a vast amount of experimental work. To eliminate spring trouble we find that it is necessary to test all of our springs on a spring-testing machine at regular intervals. It will be appreciated that one cannot get a perfect brake action with a weak spring on one side and a comparatively stiff spring on the other.

The actual wear on the band is taken care of by setting the levers, which are placed on serrated shafts throughout the entire mechanism, and also by shims which are placed on the top of the brake anchor-plate. The brake-band is made up of a single piece of spring steel to prevent deformation. There are no joints and the ends are made perfectly symmetrical so that they can be turned upside down when the upper part is worn, this being of course the first part to wear in the wrap-up type of brake. The band is hung on a spider attached to the axle and is perfectly free to rotate, its movement being limited only by the cam. A 20-in. diameter is used with a 2½-in. width, which gives a total braking area of about 600 sq. in.

The cam used for actuating the brake is flat on top and has a radius on the bottom such that the movement of the brake-band anchor is just proportional to the pedal or lever travel. As the cam nose moves downward, it drives the band against the drum and the remainder of the braking action is accomplished by the dragging effect of the lining, which tends to intensify progressively the pressure around the surface of the band. This is proved by the fact that the greatest amount of wear comes at a point about 6 in. back from the upper brake-band anchor-plate.

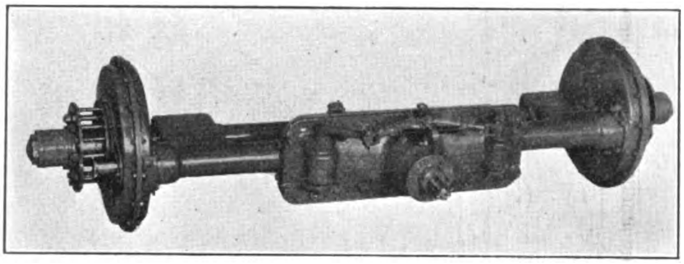

FIG. 20—A 1-TON INTERNAL-GEAR AXLE HAVING A SOLID STEEL CARRYING MEMBER WITH THE DRIVING MECHANISM IN FRONT

The drum we are using at the present time is of a special-alloy cast-steel. We have made extensive tests with pressed-steel drums, but these have always proved unsatisfactory. At present we are experimenting with a heat-treated forged-steel brake-drum having a high carbon-content, which has uptodate given extremely satisfactory service.

It is interesting to note in passing that the handbrake is arranged so that it pushes forward for application, which is just the reverse of conventional practice. The object of working the lever in this manner is that the hand has a shorter distance to travel for starting braking than it would have if one had to reach for the handle and then pull it back. This saving in time is often enough to avert a serious accident.

Fig. 19 shows a type of adjustment that eliminates the use of turnbuckles and permits road adjustments to be made rapidly and without getting under the vehicle. The tube shown, which acts as a nut, takes the ends of the rod and shortens or lengthens it as desired. It is designed so that the threads cannot possibly be damaged through carelessness. The pin, which is used as a lever, makes the tube unbalanced and consequently has no tendency to turn or change its adjustment through vibration.

The major portion of the brake adjustment is made in the garage after every 2000 miles of operation. At this time the wear on the lining is compensated for by shimming up under the brake anchor-plate so that the distance between the cam and this point of contact on the brake anchor-plate remains approximately the same throughout the life of the lining.

R. W. HASTINGS:—The firm I represent was organized to produce an improved truck-axle. To accomplish this the three factors selected for improvement were (*a*) increased accessibility, (*b*) proper enclosure and lubrication and (*c*) adequate brakes. In considering the most important features in the design of the axle, we have defined the term "adequate," as applied to our brakes, to mean a brake of large capacity, designed to deliver dependably uniform service without replacement for a period equal to the average life of the vehicle itself. Such a brake as compared with one of the band or shoe type must present an opportunity for greatly increased frictional area, an evenly distributed pressure to utilize this area fully and a complete enclosure of the mechanism as a protection from the abrasion and unreliability resulting from the introduction of foreign matter. The disc or clutch type of brake seems to fulfill these conditions best. It can be enclosed readily and, when properly designed, seems to possess qualities making it almost indestructible.

Fig. 20 shows our standard 1-ton internal-geared axle, with the usual solid-steel carrying member and the driving mechanism in front of that member. Our internal gear is enclosed in an oil-tight case, just inside of the wheel. The pinion has jaw engagement with the drive-shaft, providing for the removal of the drive-shaft without disturbing the gears, the wheel hub and bearings or the case surrounding them.

The application of the brake, Fig. 21, to our axle has been accomplished within the enlarged differential housing, it being applied in two similar units, one of which is located on either side of the differential and attached directly to the drive-shaft. The construction of the brake parallels closely that of the multiple-disc clutch, Fig. 22. A set of stationary plates of molded asbestos is slidably held within a housing that is secured to the front cover-plate. Steel rotating plates of special double construction are placed alternately with these friction plates

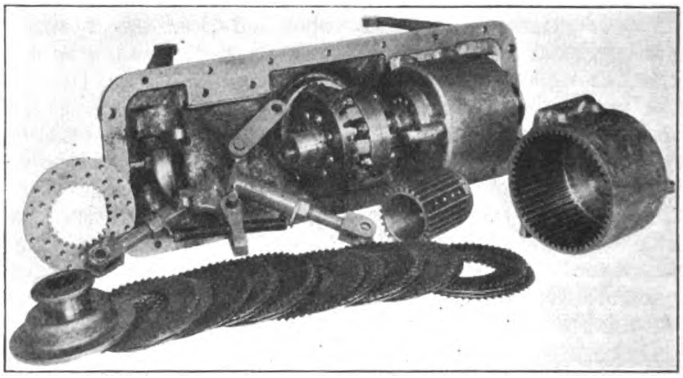

FIG. 22—THE CONSTRUCTION OF THE BRAKE IS VERY SIMILAR TO THAT OF A MULTIPLE-DISC CLUTCH

and are slidably mounted upon the hub member, which rotates with the drive-shaft by virtue of a splined engagement thereto. End-pressure is applied to the plates by a bell-shaped pressure-plate that is actuated by forked arms attached to two vertical shafts extending through the top of the cover-plate and terminating in lever arms which carry the toggle equalizing members. The toggle mechanism, Fig. 21, consists of two members having cam-shaped ends so that as their inner ends are pulled forward, thus separating the lever arms and applying the brake, the point of contact at the center between these cams does not travel forward but remains stationary, maintaining a constant angle of toggle action. This feature allows us to take advantage of the powerful toggle action and at the same time maintain a constant multiplication of leverage. While the multiplication of leverage due to this toggle construction increases the effective pressure on the brake, an item of probably greater interest and value is the automatic equalizing of

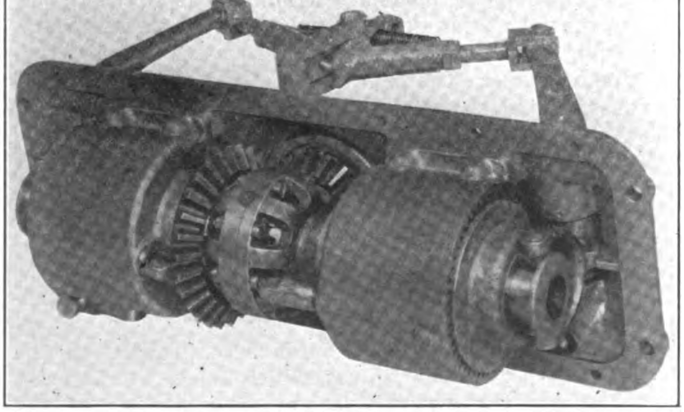

FIG. 21—THE APPLICATION OF THE BRAKE TO THE AXLE SHOWN IN FIG 20 IS IN TWO SIMILAR UNITS, ONE ON EITHER SIDE OF DIFFERENTIAL

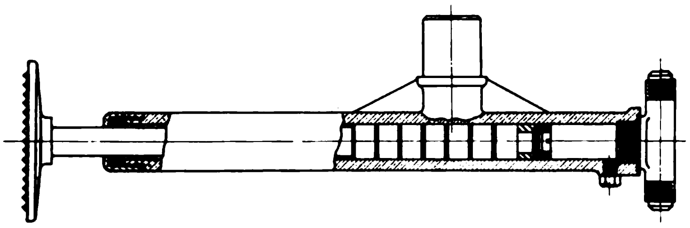

Fig. 23—Pedal Used on Pierce-Arrow Equipment in Connection with a Hydraulic Brake

the brake pressure thus accomplished. With the brake released, the toggle members are held by the return spring against a locating seat provided on the face of the cover, thus maintaining proper and equal clearance or opening for each of the brake units. Depression of the pedal draws the toggle forward, releasing it from this locating seat and thus leaving the entire system of levers free to swing to either side and so compensate for uneven adjustment. With the mechanism in this free position, it is evident that the reaction from the pressure upon one brake unit finds no resistance except that

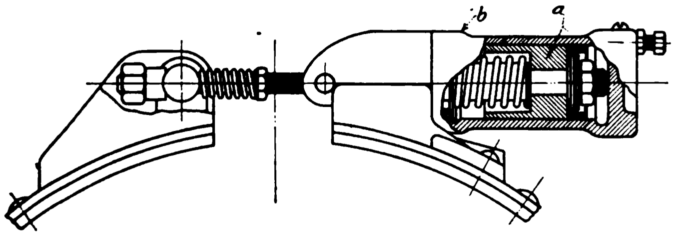

Fig. 24—Construction of the Band Mechanism Employed on a Hydraulic Brake

of the pressure upon the other unit, for which reason an absolutely even pressure upon each brake is guaranteed.

The question of oil circulation has been given considerable study. We have provided a means for introducing oil at the center of the brake that allow it to flow out through the specially spaced rotating plate, thus carrying away the heat that develops in the brake and giving it ample chance to radiate from the large surface of the axle housing. This provides an unusually cool brake and we have found almost no condition under which it is not possible to place one's hand upon the axle.

Proper adjustment of the brakes is made by releasing the outer nut on the toggle cams, following this with a similar manipulation of the inner nut, setting the brake arms over and moving in the pressure plate. Enough clearance is allowed to wear out the brake without any other adjustment. We have run one job about 19 months and still have the same plates; they show almost no sign of wear.

A Member:—I will relate my experience with the brakes on the Delage car from a sales standpoint. There are various advantageous factors about using brakes on all four wheels. One of the first and most important is comfort. We find that, no matter how suddenly the brakes are applied on all four wheels, there is less tendency to throw the passengers forward and out of the seats. Instead, we find a tendency of the entire chassis to sink into the wheels. In fact, one can see the hub cap sink an inch or two, as the brakes are applied harder. Another very important feature of four-wheel brakes lies in the increased safety they afford. These brakes can be applied on wet days, in snow and on ice, without chains, just the same as would be done on a dry pavement. The effect seems to be the same, with the possible exception that the brakes sometimes cause all four wheels to slide. But during an experience of 18 months they have never caused side-skidding. The general economy on brake-linings is another important item. There seems not to be the same amount of jar, because the brakes are seldom put on hard. A very slight pressure of the foot will stop the car. The economy on tires is very marked, due probably to the fact that the rear wheels do not drag. The tires seem to give much greater mileage than when using brakes on two wheels only. We have never determined the increase in mileage, but it is surprisingly large. I do not know whether that is due to the brakes or to the

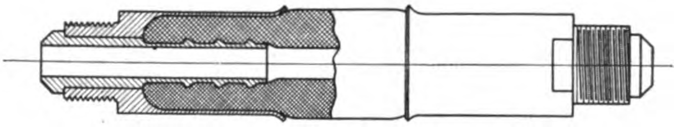

Fig. 25—Special Hydraulic Hose and a Somewhat Unusual Fitting Are Employed to Overcome Motion between the Frame and the Axle

car, but I feel that the brakes are responsible largely. These four-wheel brakes make it very easy to drive safely in traffic. One can run up closer to the car ahead. Last and most important from a sales standpoint, we find that when a man acquires the habit of driving a car with four-wheel brakes, he is less inclined to buy one having less braking power.

On the Delage car, we find that the brakes fulfill all our requirements under all conditions. The adjustments are very simple. There seems to be no wear. With four brakes, we have double the braking surface, with half the braking effort. Altogether, we find that four-wheel brakes afford very comfortable riding and are very

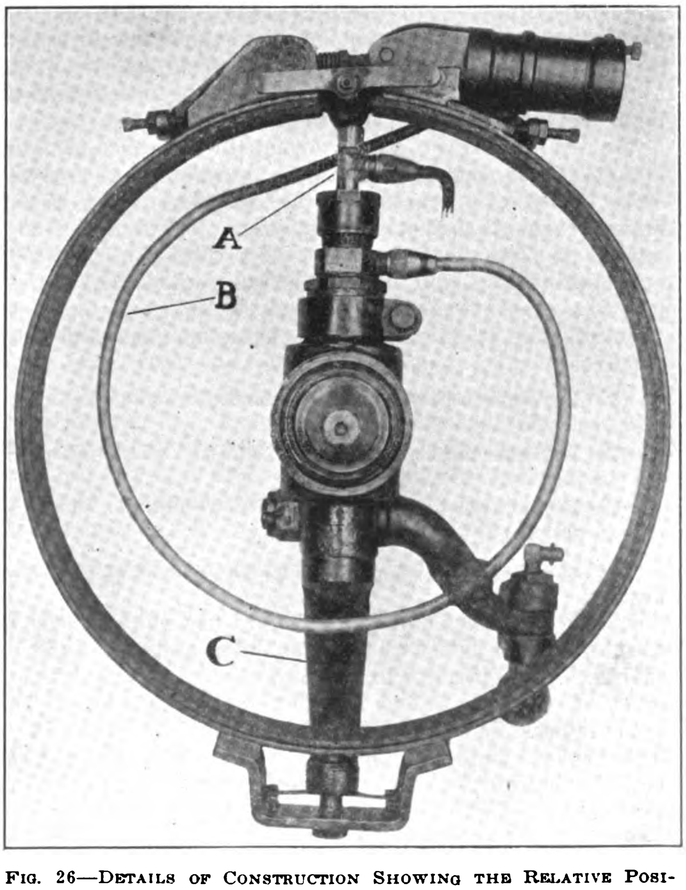

Fig. 26—Details of Construction Showing the Relative Positions of the Swivel Joint, the Knuckle-Pin and the Coil That Takes Up the Band Movement

much favored by the public. The people who have driven cars equipped with the four-wheel brake in Europe are very enthusiastic regarding their operation.

MONTGOMERY MAZE:—The four-wheel hydraulic brake that we are now manufacturing can be considered only in the light of an accessory. Whatever we accomplish in the way of replacement equipment can be viewed only from that standpoint, for it has been necessary to work around existing conditions of design which are far from being uniform or desirable from our point of view. Future factory equipment design can readily excel in looks, efficiency and cost.

The main items increasing our efficiency are

(1) Perfect equalization, automatically obtained since the application is through fluid pressure only
(2) Complete freedom from mechanical linkages
(3) Complete absence of any effect on braking arising from relative frame and axle movement
(4) Use of 100 per cent of the car weight as a source of road friction instead of a fraction of the normal weight on the rear wheels

All four wheels are operated simultaneously by a single pedal in the conventional manner and the degree of retardation is entirely within the control of the operator. The system is purely a displacement proposition, and an emergency stop can be obtained only by the application of the emergency pressure.

Fig. 23 shows the pedal we use for Pierce-Arrow equipment; on the Cadillac the standard pedal is not disturbed. It has a normal travel of approximately 3 in., the reserve being sufficient to wear out a third of the lining without adjustment.

Fig. 24 illustrates the construction of the band mechanism. The piston *a* is attached to one end of the band and the cylinder *b* to the other end. Pressure on the pedal displaces liquid from the master cylinder into each of the wheel cylinders alike, drawing all of the bands together with an equal force. It will be noted that no band can grip until all are in contact.

We are operating with a normal line-pressure of from 100 to 150 lb. per sq. in., but it is possible on extreme stops to set up a pressure of 750 lb. The liquid is conducted through a special, soft copper tubing, well supported at frequent intervals. This tubing is drawn to our specification and has an ultimate strength equivalent to a pressure of 13,800 lb. per sq. in.

In addition to this rigid line, three compensations are required

(1) For motion between the frame and the axle (spring action)
(2) For motion between the axle and the wheel (steering)
(3) For motion between the ends of the bands (brake action.)

The last named does not occur in our internal-brake design.

It has long been our intention to use rubber hose to counteract (1) but we were unable to secure a suitable fitting. This difficulty has been overcome by the development of a special fitting of exceptional merit which has made it possible to use a length of special hydraulic hose. The exposed length of hose shown in Fig. 25 is 1 in. and the flexibility is ample to compensate for any spring movement. This hose is built to our specification and each unit is tested to 3500-lb. pressure under vibrating conditions.

A swivel joint *A*, Fig. 26, is mounted directly over the knuckle-pin and in line with it to permit perfect steering. Braking cannot affect steering in any way, nor can steering affect braking. A coil pipe *B*, 40 in. long, takes up the band movement on the external brakes. This movement has a maximum of 3/8 in. The method of mounting the front-wheel equipment on an Elliott type axle is shown in Fig. 26. The swivel is mounted on the knuckles and the knuckle-pin *C*, being stationary, carries the lower half of the band. On the reversed-Elliott type of axle, the swivel is mounted on the knuckle-pin and the lower half of the band is supported from the steering arm which we replace in this case.

There are three points of closure in our system; (*a*) the copper-tube connections which are S.A.E. Standard flared-tube fittings, which are satisfactory in every way (we have never had a leak at this point); (*b*) the swivel packing boxes (these hold a maximum of special packing; because of their design that permits the operating pressures to compress the packing further, we have yet to encounter a single failure); and (*c*) the cylinder cup leathers. The last named has been a source of great difficulty but we have located a cup that has failed to show any leak during a 4-year test; with the stabilizing of the leather market, we are now able to secure a uniformity of material that obviates any further leaks at this point.

To make our system complete, however, we have adopted a standard reserve tank; it will be necessary every few months, due to slight seepage, to draw liquid from this tank. By keeping the tank full and filling the line only in this manner, all chance of drawing air into the line is done away with.

Brakes are uppermost in the mind of any owner in hilly country and he grasps at any remedy for his constant worry; but, regardless of experience, it takes but a single demonstration under any conditions to impress upon the operator that brakes have at last reached the plane of present-day engine design.

In addition to the safety factor given by the brakes, the owner also gets an actual monetary return. Tires can be made to show a 30-per cent increase in mileage from the standpoint of tread wear. Adjustments are not required at less than 10,000 miles, and relining of brakes is unnecessary under 30,000 miles. These figures are an average obtained from our test-cars; it is very doubtful if any owner will ever subject his car to the extreme and continued operation that they received.

HERBERT CHASE:—Brakes must absorb power quickly when they are applied but at other times they ought not to absorb power. One common fault with both American and foreign-built cars is that brakes do drag more or less. Possibly the external-band brake drags more than the internal. There is room for better construction in brakes in general. The average brake does not compare well with the other parts of the car in the quality of its construction. British criticism of the American car with respect to the brake construction is rather caustic and, in some cases, probably is justified. It should be borne in mind, however, that British cars cost more than American cars on an average, and more expense can be put into their brake construction.

What conclusion has Mr. Farwell reached about the metallic brake-lining that is generally used abroad? There are several different kinds, I believe. I wish to know how they compare with the fabric generally used here in regard to the value of the friction coefficient. Also, will Mr. Carson describe the pressed lining that I understand he has been working with? Possibly he will describe also the testing apparatus that is being developed by the Bureau of Standards for determining the

relative merits, including the wearing qualities, of different linings.

H. G. FARWELL:—With reference to the relative coefficient of friction of the metal brake-lining and the fabric lining, there seems to be considerable discussion and difference of opinion. The coefficient of friction will run approximately 0.4; with a bronze shoe it will run approximately 0.2. That has been corroborated by engineers from abroad. They use more encased drums abroad than we do here; pressed-steel drums are used almost altogether.

V. W. PAGÉ:—I witnessed some tests of the multiple-disc brake. I was afraid there would be considerable drag and attendant heating. After a number of tests down a steep test-hill, I was able to put my hands on that casing without any discomfort; a standard touring car that accompanied the test car stuck on this same test. I could not place my hand anywhere near the brake-drum on account of the heat. Then we tried some coasting tests. We find fully as good results with that form of multiple-disc brake as would be obtained with the conventional band-brake well adjusted, and considerably better results that one would get with a band-brake ordinarily adjusted.

C. CARSON:—Mr. Chase has requested information regarding the testing apparatus at the Bureau of Standards. The Parts and Fittings Division of the Society's Standards Committee was assigned the subject of brake-lining and it undertook to develop some standard method of testing brake-linings in collaboration with the Bureau of Standards at Washington. The results so far obtained have not entirely solved the problem, but very gratifying progress has been made.

A pressed-steel drum was mounted on the shaft of a dynamometer and a skeleton frame of the general form of a prony brake was built around it. A spring-balance was installed between the arms, to adjust the load on the brake-shoes. Two short flexible bands were used instead of a complete encircling band, as is commonly found on wheel brakes. These carried linings about 11 in. long, 2 in. wide and ¼ in. thick, and the pressure was applied so that a nearly uniform pressure per square inch would result. First, we tried to find what pressure per square inch could be carried and what velocity should be used. To accelerate the test, an attempt was made to run the apparatus with a water-cooled drum. That was provided by putting a plate equipped with the usual tube for introducing water in the open end of the pressed-steel drum. But when using high pressures and high velocities, heat is generated so much faster at the point of contact with the lining on the drum than it can be transmitted through a steel drum ¼ in. thick that, even with a drum containing water, the surface of the drum will fuse while it is running. So, we were forced to abandon the theory of running at high pressures with a water-cooled drum, because the water dissipated only a limited amount of the heat generated. The tests are not yet completed. We have made a long series of tests using both water-cooled and dry drums. Apparently, a character of lining that will give excellent results at moderate loads will break down and give very unsatisfactory results when high pressures and continued brake applications are given to it.

It seems unfortunate that as yet, all through the industry, there has been no standard method of testing such materials. Some firms have tested brake-lining by making what might be called a skid test. They apply a piece of friction material to a rotating drum, hang a certain weight on it, make it turn for a certain number of hours at a certain number of revolutions per minute and record the result. The material that endured the longest was given the credit for being the best, without taking into consideration the power absorbed by the brake during the run. We have tried to eliminate such a condition in the apparatus we use. At present, the doing of a uniform amount of work is taken as a basis for the test. The pressures per square inch are varied to make the power consumption constant at all times. We keep the revolutions of the dynamometer and the power consumption constant and change the pressure on the lining. We feel that, if the linings are doing the same amount of work, we can then approximate a fair comparison of their life.

Among other interesting results, we found that some of the yarn in the linings had been made with brass-wire cores and that the surface of the lining became covered with copper plating; to a certain extent the steel drum was coated likewise. Investigation indicated that the heat generated was sufficient to drive the zinc out in the form of vapor or dust. We have found that the degree of vulcanizing in the rubberized linings seems to have a very pronounced effect on their wearing quality. Linings made from the same fabric and having practically the same rubber mixture, but different degrees of vulcanizing, will vary in their performance under the same load conditions from 20 min. for one sample to 16 hr. for another.

A recommendation probably will be made to the users of brake-linings, asking them to modify, if possible, their method of installation. A prevalent method of installing lining requires the strip to be very flexible as the usual practice is to rivet it at the ends first and then press out the kink left in the center to make the lining hug the band tightly. That requires a considerable flexibility in the material. We believe that in adhering to that flexibility the users of lining are sacrificing much of the life of the lining. That is indicated by the change in wear according to the degree of vulcanization. With a vulcanized or with the woven type of lining, hard pressed and impregnated with a hard compound, this kinking method of installation could not be used and there would have to be a change in the method of application.

We find that they are using very hard cured lining in European practice. The Ferodo lining, which has a corrugated shim between the lining material and the band to circulate air and dissipate the heat, is an example of such material. This lining is an asbestos woven fabric; it is very hard, compressed to a very high degree and almost lacking in flexibility. It is installed usually in comparatively short curved pieces, because it is not flexible enough to bend around a band. My opinion is that, for long life and maximum service, the present method of installing brake-lining and its degree of hardness must be changed.

COEFFICIENT OF FRICTION OF BRAKES

The determination of the coefficient of friction is perhaps the most elusive problem we deal with. We have not even been able to determine it as a constant on any one particular sample during a run. We doubt very much if a really fixed coefficient of friction can be maintained with an impregnated, woven, or folded and stitched fabric, or for any fabric composed of yarns interlaced. It may be reached in some new form, such as unwoven or pulp lining commonly called molded material, used in some types of clutch-facing. A rough value for this coefficient would be about 0.40, but I think possibly that it should be modified to 0.36 for a woven and 0.42

or 0.43 for a rubberized lining, the latter having a slightly higher coefficient. Almost any coefficient desired can be obtained by changing the compound. One can make a lining having a severe grip or, changing the compound by introducing certain other ingredients such as waxes in one form or another, secure a low coefficient.

From the tests, we believe that the reason for the variation in the friction coefficient is that at no two times during the wear of a piece of lining is there the same condition of surface contact. Consider a piece of folded and stitched, laminated lining with a rubber compound. At the beginning, there is a veneer or surface of rubber that has a certain coefficient of friction on the steel drum. As the wear progresses, the coefficient changes because the surface contact is composed of a certain area of asbestos fiber, metal and rubber. The areas of the three materials in contact change continuously and the coefficient of friction changes correspondingly. While there is no uniform cycle of performance, there is a fluctuation and continual variation, even in the same piece of lining.

L. G. NILSON:—Who has had any experience with the metallic brake-lining that is a composition of lead and copper?

MR. CARSON:—We have not tested any of that material. I saw some clutch-rings that were made of that material recently. The engineer who conducted the test of the rings eliminated that material on account of its high cost and because the coefficient of friction was so low that it would have been necessary to increase the area of the clutch to a prohibitive amount to use it interchangeably with asbestos materials.

W. C. MARSHALL:—What tests have been made to show which type of brake is freer from oil, the internal or the external? The efficiency of the brake depends largely on whether the oil gets in on the brake-band and the drum. In some cases the oil might be thrown off. In other cases it might hold.

MR. FARWELL:—Our experience shows that oil gets in on both types. Probably a greater amount of the oil will be retained on an internal than on an external type. We find, in some cases, oil or grease on both the internal and external bands. It seems to involve choosing the lesser of two evils.

N. G. BERGENHOLTZ:—In regard to the brakes on the buses of the Fifth Avenue Coach Co., it seems that many of these are not shoe brakes and do not act equally in either direction. On this particular brake, our attention was called to the fact that the cam was not of the same shape on both sides. How does that brake act in going in a reverse direction? Does it give an equal braking effect?

MR. REESE:—No, it does not. The brake that we use is known as the "wrap-up" type. The efficiency is the greatest when going in the forward direction. When going in the reverse direction, it is much harder to apply. I should judge that if one could apply the necessary pressure the efficiency would be the same.

MR. BERGENHOLTZ:—Is the object of that cam shape only to take up the slack first, before the pressure is applied, rather than to try to equalize the pressure in both directions?

MR. REESE:—Yes.

M. C. HORINE:—A self-wrap brake is essentially a one-direction brake. Brakes have been developed which are known as the double-wrap type but a double-wrap brake, either external or internal, is practically not a wrap-type brake at all. That is, it is not a snubber; it does not work on the principle utilized when several turns of rope are taken around a capstan, as is the case with the ordinary self-wrap brake, because the self-wrapping on one side is compensated for by the unwrapping of the other. This matter of having a brake act equally in either direction is important, particularly in motor-truck work; great weights must be considered and gear-changes are not so certain on a grade, because the truck is going at a slower speed and its inertia will carry it forward a very much shorter distance. The experience of the company I represent was such that, previous to the time that type of brake was abandoned, it was necessary to redesign the cam so that it had equal action on both ends of the band. The only advantage of the flexible band in a double-wrap brake is to give a more or less equalized pressure.

The effects of pressure and speed on the wear of a brake have been suggested. The amount of wear on a brake should be roughly commensurate with the amount of energy dissipated. It seems that a brake might be designed with a small drum operating under very high speeds. This would mean lower friction at higher speeds and no more wear per square inch than with a larger brake operating at lower speeds at higher pressures. It seems to me that the wear would be dependent upon the amount of area. There is considerable buncombe with regard to braking area. It is possible to design a brake with a great braking area, much of which is worse than parasitic. Taking the shoe illustrated in Fig. 27 as an example, it would be possible to line it up to the tip where the cam contact is and down to the hinge. Such a brake acts as a lever. The portion of the shoe at *a* is on the wrong end of the lever. One can get very little pressure at that point, whereas at *b*, close to the fulcrum, one can get a great amount of pressure. If the portion *a* is lined, the lining will act as a spacer. It is impossible to get sufficient pressure on it to make an effective brake, and yet it acts as a spacer to prevent the portion *c d* of the lining which produces effective braking, from making contact. The portion of the lining at the point *b* does not reach the drum after a certain amount of wear, because it is so close to the hinge; so, it is largely parasitic area. On a rigid drum of this character I believe it is not necessary to have a lining for more than about the distance *c d*. I do not believe that any more lining on that shoe will give braking effect. In a case where it is close to the tip, it may prevent effective braking.

In regard to having internal and external brakes on the same drum, which is the conventional practice on touring cars and the cheaper kinds of truck axle, it does not seem right to me to put two brakes of any type on the same drum. Asbestos is hardly a good conductor of heat, whereas iron is a very good conductor. Since the drum is the brake member that contains most of the heat, it seems reasonable that the binder in the lining should burn because the drum gets hot. If one could always apply a brake to a cool drum the lining would not burn. Suppose we have two sets of brakes which we apply alternately in descending grades to avoid burning either set. If we apply the alternate sets of brakes to the same drum, which has already become heated by the application of the preceding set, the second set will burn almost immediately. Hence, the ideal brake arrangement would be for each set of brake shoes to act on a separate drum, which is the condition we have with four-wheel brakes and with shaft brakes.

Another consideration in connection with shaft brakes is that of equalization. It is possible to equalize the pressure on two brakes in a number of ways, such as using a simple cross-tree, a differential arrangement,

pulleys, fluids or other means. But that only equalizes the pressure and does not equalize the braking effect. It appears to me that the only way to equalize the braking effect is through the differential, inasmuch as two tires will never follow exactly similar tracks and it is the tire on the road that actually gives the braking effect. That seems to me a strong argument in favor of the shaft brake. However, the shaft brake has some defects which I think have not been given sufficient attention.

The greatest complaint against shaft brakes is that they chatter. Chatter in the shaft brake is due to many causes, chief among which is the fact that the brake itself is not supported firmly enough. Most shaft brakes overhang on a propeller-shaft and, naturally, since there is no propeller-shaft that remains concentric, there is a slight wobbling of the drum which ordinarily is supported separately from the shoes. Another cause of chatter is the looseness with which the actuating means and the shoes themselves are attached. If the shaft brake is properly designed with rigid shoes and a drum that is mounted between bearings, as one would hang a grindstone, and if the entire brake and its mechanism is supported by one rigid frame, there will be no chatter. I know this because I have experimented with brakes of that character.

The ordinary method of mounting a shaft brake is to have it operated by a pedal. A number of cars now obsolete had brakes of that sort, and it was characteristic of the operators of those cars that they almost never used the foot-brake. The operator always used the hand-brake because the shaft chattered. If it must chatter, the shaft brake should be operated by the hand-lever, because ordinarily the hand-lever is the one which is used to lock the car when parked. This means that it is generally applied when the car is stationary. The foot-brake ordinarily is used only when the car is moving; therefore, if we must have a chattering brake, let it be the hand and not the foot-brake. Another reason why the shaft brake should be operated by hand is that spring action does have the effect of shortening and lengthening brake-rods that are connected to the rear axle, and that it is a very common experience when the brake is applied with the car loaded to have it release itself when the car load is taken off. With very light cars, where the hand-brake acts on the rear axle, that is a common experience; when the brake is applied while the passengers are in the car, it releases itself when they get out of the car. That is experienced to a much greater extent on trucks. It is not exactly a common experience, but it does happen occasionally that a truck releases its brakes when the load is taken off, the brakes having been proprerly locked while the load was on. Another common result is that with the hand-brake applied with the truck empty it becomes impossible to release it after it is loaded. Naturally, a shaft brake, fixed to the frame cannot be affected by spring deflection and hence is the ideal hand brake.

A certain amount of prejudice against the shaft brake originates from the fear that, acting through the drive-shaft, universal-joints, drive gears, differential and axle shafts, it is less reliable and that these parts will be subjected to an abnormal stress from a sudden brake application. Experience shows, however, that failure of rear-axle brakes due to crystallized brake-rods, stuck and rusted pins and burned-out linings is more common than failure of driving parts. The strains to which the driving parts are subjected from shaft brakes, furthermore, are not so severe as is commonly supposed. It can be demonstrated easily that the shock on these parts pro-

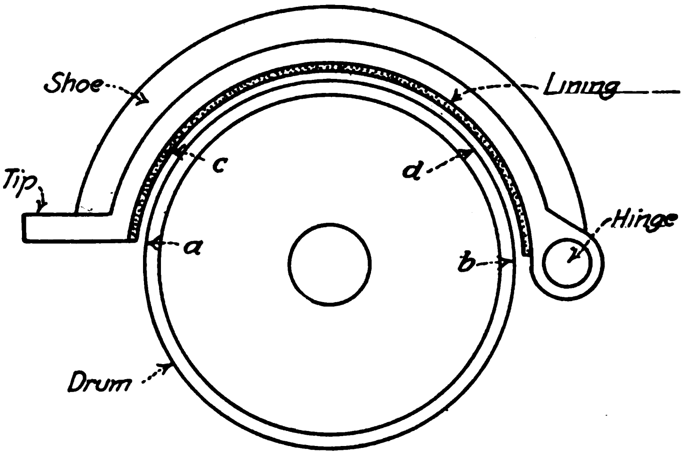

FIG. 27—AN EXAMPLE OF FAULTY BRAKE DESIGN

duced by a sudden application of the clutch at high engine-speed with the gearshift in low or reverse of our modern large-range gearboxes greatly exceeds the brake torque at which the wheels will slide.

A solid rod is apt to vibrate and crystallize. Cable is one of the first means we used for applying brakes to automobiles, but this has never been entirely satisfactory because cable is apt to fray. The flat-steel strap seems to have a certain amount of promise, except that it is a single strand and, as was found in airplane practice with streamlined solid wire, it is hardly safe. Strap of that sort is brittle and, if it fails, it fails clear across and the rod is broken. Some time ago I had a different style of brake-rod or cable on a small car, that consisted of a form of chain made up of flat brass-links, each link being folded so that the holes in the free ends registered and permitted the next link to be folded through these openings. The experience I had with that substitute was very encouraging. I think a little experimenting in the use of these folded sheet-metal chains will show that they have real possibilities as substitutes for brake-rods. They are extremely flexible, can be made very strong, and offer a very ready means of adjustment.

With regard to the disc brake housed in the differential housing mentioned by Mr. Hastings, granting that it does have a very powerful effect and very long wear and that the brake keeps cool, is not that effect at the expense of the oil? It is not true that the oil in that housing deteriorates very rapidly because it is being burned between the brake surfaces?

MR. HASTINGS:—We have operated this brake about 19 months and it was our practice to remove the oil frequently for inspection. We found that it did not deteriorate. During the last part of the run we have had one supply in the housing for 4 or 5 months and we find it in good condition today.

MR. HORINE:—Can you account for that?

MR. HASTINGS:—It is because there is a very large area and the heat is distributed over that area.

MR. HORINE:—Will not particles of brake facing, dislodged by friction, be circulated with the oil and do considerable mischief in the gears and bearings?

A. M. WOLF:—It is one of the first fundamentals of brake design to have them absolutely free when they are in the "off" position, and a correct brake should absorb absolutely no power when not in use. Dragging brakes are a prevalent failing and obviously a large factor in fuel-consumption.

To mention a few other means of braking, we can use air when coasting downhill, with the switch off and

working the engine as a compressor. However, the ordinary engine is not a very efficient brake under these conditions. The Saurer engine is built with a sliding camshaft that modifies the valve action so that the engine will absorb a maximum amount of power while turning over. The Saurer people also tried out a fan brake. The fan was mounted under the center portion of the chassis where it had plenty of room and, when it was thrown into action, its resistance was similar to that of a fan dynamometer absorbing energy.

An hydraulically actuated brake was mentioned, and attention has been paid also to hydraulic means of absorbing power. A car was developed in Europe in which the constant-mesh gears of the transmission were encased so that they would form an oil gear-pump. To cause braking effect, a pipe through which the oil circulated was blocked by closing a valve interposed therein. I understand, however, that this was not a success, due to the excessive heating of the oil and the very high pressures encountered; but with modern methods this idea might be revived. Most hydraulic transmissions function as a brake when the control valve is shut or set in a position corresponding with a speed slower than the prevailing one.

The airbrake that uses air as an actuating medium is somewhat old, having been applied on the first Northern four-cylinder car. This car had an air clutch, as well as brakes applied by air-actuated pistons operating the brake-rods. The clutch consisted of a large leather disc forming part of a bellows and was mounted so that, when air was admitted behind it, it would extend slightly forward and come into contact with the rear finished face of the flywheel. The airbrake is now being applied to trailers; it seems that the trailer application will cause both hydraulic and air actuation to become popular. It will be recalled that the Knox-Martin tractor was brought out with an hydraulically actuated brake.

Small reversible high-speed electric motors, acting through a large worm-gear reduction, have been used to actuate the brake-rods. A small button or lever switch-control makes the operation extremely simple; but such a system involves many complications in performing an operation that can be accomplished by very simple means. Unusually large vehicles might be an exception.

With reference to brake adjustments, I mention a device which automatically tightens the brake-rods when their travel is too great. This is done by a ratchet that is held in fixed relation from a cross-member. Movement of the brake-rod beyond a predetermined limit causes the ratchet to rotate a member which is threaded over the brake-rod. This device is borrowed from railroad practice; the slack adjuster, in this case air-actuated, is located on each brake cylinder.

The mounting of the brake cross-shafts deserves consideration, so that they shall be free from binding due to distortion. It was interesting to see how the Renault design obviated any such tendency by its universal mounting. There is one truck on the market that has a cast cross-member which also forms an anchorage for the front end of the rear springs. All the brake cross-shafts are mounted on this member and, due to its unit assembly, there is very little or practically no chance of binding occurring in service.

It is interesting to note the disappearance of the one-time long equalizer bars. They often exceeded the frame width, so that the rods running to the drums would be outside of the frames. Today, the rods are being kept within the frame side-rails. Truck design also is reverting to this method to a large extent. It allows a more substantial brake rigging on the axle.

In the Hotchkiss drive, due to the displacement of the axle under torque, driving and braking stresses, some builders allow for a certain amount of lost motion between the pedal and the final brake-rod. Considering the internal brake, I believe that we should design cams with a small circumferential section or base circle before the shoes are expanded. It naturally would be a more costly cam. This is not necessary, of course, when radius-rods or an anchored torque-tube is used, if the clevis-pin of the brake-lever has the proper location.

Regarding the lubrication of the brake rigging, we see cars today with grease or oil-cups in places that the owner or driver will never bother to reach. In fact, some cannot be reached without crawling under the car on one's back. This refers to brake cross-shafts on the frame, and also to brake shafts on the axle. I am a firm believer in the oilless bushing, of any of the several types, for such locations and I am surprised that all companies do not use them.

PROGRESS MADE IN GARAGE EQUIPMENT

(*Concluded from page* 66)

in that part of the Country, some trucks and plenty of cars. This shop got all the tractor and all the motor-vehicle business, by virtue of its excellent equipment.

Most dealers will say that garage equipment costs too much. Some of the men in a shop are pretty handy. I have seen some fine engine-stands and other equipment that the mechanics built themselves. If the shop is well tooled-up, one hardly can blame the proprietor if the men make their own equipment. However, the repairmen say they will buy equipment if it is priced reasonably. I saw a portable hoist in a corner of one garage and asked if it ever was used. I was told that one objection to its use is that most cars are fitted with bumpers that prevent this particular hoist from reaching over far enough to lift the engine out.

In regard to the flat-rate system of selling service, suppose I have a shop that is not well tooled-up and some other person has a shop that is very well equipped, and that we both are handling the same make of car. If I use old-time methods in taking a cylinder-head off and removing the valves, and the competing shop has the necessary equipment and can do the job in half the time, it can establish a flat-rate system of selling its service that will appeal far more to the car-owner than my service does. Hence, the equipment plays a direct part in the selling of service by the flat-rate method, and it is one of the essentials that certainly must be considered. I know that the car builders are beginning to take more interest in this. For instance, in laying out the cars they are beginning to plan equipment to go with each operation in service before they settle upon a design. They consider the steps that will be taken in servicing that particular unit. Then they design equipment that will go with that car. That goes to show the amount of thought that is being given to this very important question of garage equipment.

Malleable-Iron Drilling Data

By H. A. Schwartz[1] *and* W. W. Flagle[2]

Cleveland Section Paper *Illustrated with* Photographs *and* Charts

After commenting upon the two contradictory attitudes toward malleable iron in the automotive industry and outlining its history briefly, the authors discuss the differences between malleable and ordinary gray-iron and supplement this with a description of the heat-treating of malleable castings.

Five factors that influence the machining properties of malleable-iron are stated. These were investigated in tests made with drills having variable characteristics that were governed by six specified general factors. Charts of the results are presented and commented upon in some detail, inclusive of empirical formulas and constants and deductions made therefrom.

The machining properties of malleable-iron is a new subject in engineering literature. C. F. Kettering, past president of the Society, once expressed the belief that the future of engineering would consist of a careful study of all the materials of construction and the selection of the material for a given purpose the properties of which most nearly correspond to the ideal. The department over which I preside was organized with the idea of furnishing the automotive or any interested industry authentic information as to the properties of malleable-iron castings.

Apparently, there are two well-defined and contradictory attitudes toward malleable-iron in the automotive industry, one being decidedly unfavorable. Having had occasion recently to buy a car, I inquired regarding a certain make of machine. The salesman told me that no malleable castings were used in that car. The first thing observable on raising the hood was a malleable casting and, as a matter of fact, the car in question had 31 parts made of malleable-iron. It is unfortunate that anyone should wish to conceal the use of so valuable a material in parts for which it is suited. The opposite viewpoint is held by certain manufacturers. They seem to feel that malleable-iron should give satisfactory service for any purpose whatsoever, and attempt to make everything of it regardless of its suitability.

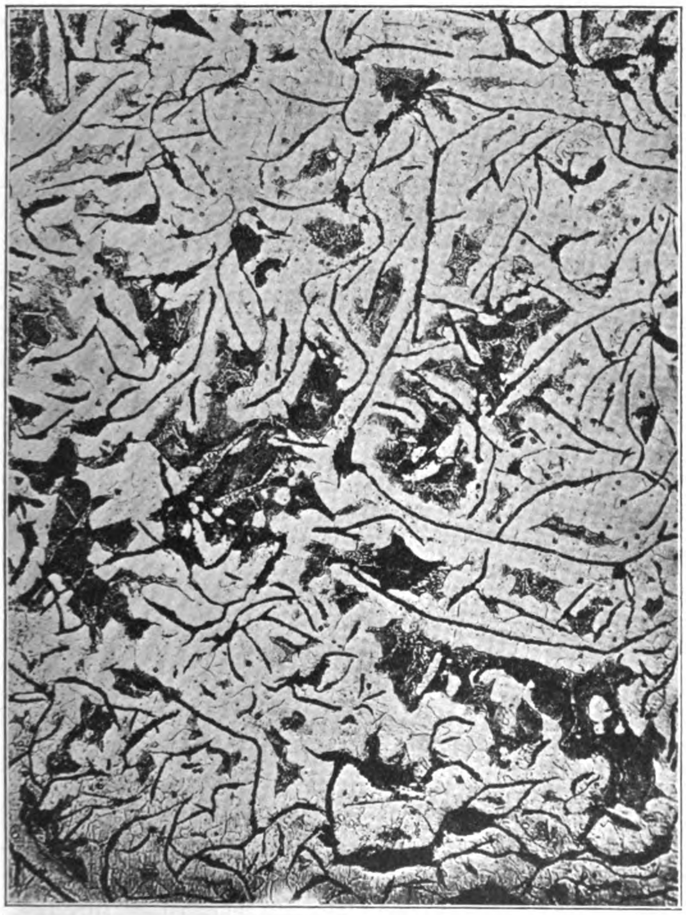

Fig. 1—Photomicrograph of Gray Iron Magnified 100 Times

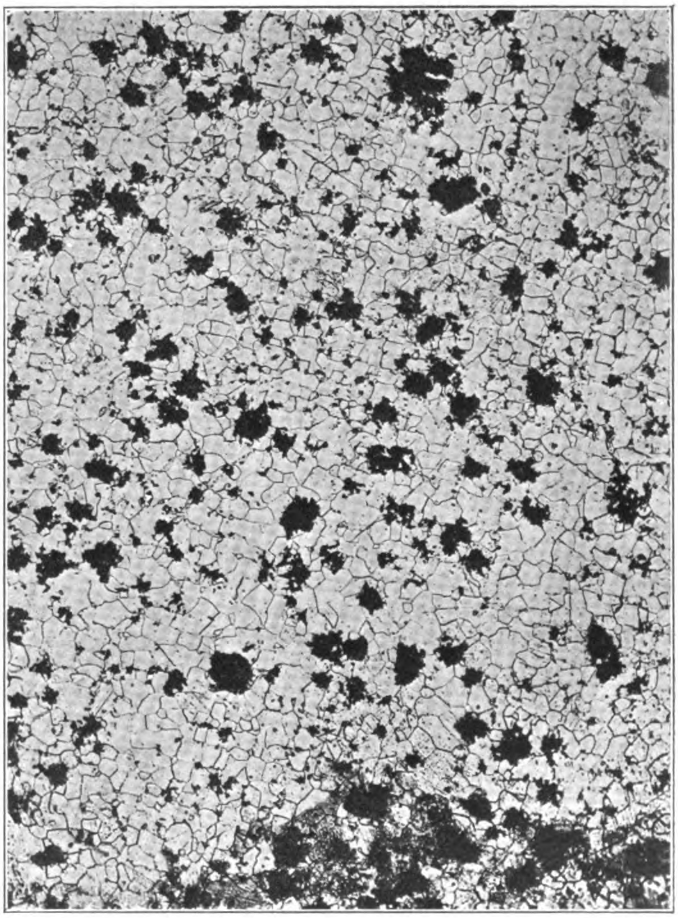

Fig. 2—Photomicrograph of Malleable Cast Iron Magnified 100 Times

The history of malleable-iron dates back to Reaumur, who published a description in 1722 of a system of making it by decarburization. In trying to practice the art of making malleable-iron castings, an American named Boyden discovered a different product in 1826 which is

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[2] Engineer of tests, research laboratory, National Malleable Castings Co., Cleveland.

now known as black heart malleable. The most recent and accurate figures at my disposal indicate that there are 176 producers of malleable castings in the United States. Others estimate that there are more than 200 producers, but the important ones number between 75 and 80. The principal plants are located in the territory north of the Ohio and east of the Mississippi rivers.

Malleable and Gray-Iron Differences

Malleable-iron consists of a mass of nearly pure iron or ferrite, through which some 2.0 to 2.5 per cent of carbon is scattered in a spheroidal form in the free state. This particular form of carbon is characteristic of this product only and is known as temper carbon. Gray-iron consists of a metallic mass composed of a mixture of ferrite and pearlite, through which some 3.00 to 3.25 per cent of free carbon is scattered in the form of flaky graphite crystals. It is obvious, as shown by the photomicrographs reproduced in Figs. 1 and 2, that the former conditions produce much less interruption of matrix than the latter. A further consideration is that pure iron is much more ductile than pearlite, which has a corresponding effect upon the two cast products.

The carbon in malleable-iron exists in the geometric form that characterizes it because that carbon is liberated at a temperature when the metal is nearly solid, but the graphite of gray-iron is liberated at a temperature but little below that of the melting point. The fact that it must grow in a nearly solid medium rolls or crushes up the free carbon of malleable-iron into the spheroidal form that characterizes temper carbon. The process of manufacture is first to produce a casting that contains no free carbon and then to heat-treat that casting so as to break up the combined carbon into iron and free carbon.

The Heat-Treating Process

Two fallacies are encountered frequently with respect to the annealing or heat-treating process. The first is that the process is conducted for the purpose of eliminating carbon and that, therefore, the surface of the metal must differ widely in properties from those of the center. The elimination of carbon from the surface metal is a mere incident in the process and affects the metal but slightly, increasing the ductility of the product a little. The primary purpose of the heat-treatment is the separation of cementite into ferrite and carbon, and this process does not proceed more rapidly or more completely at the surface than within. Malleable castings, when machined, therefore possess properties that are comparable with those of unmachined castings.

The second fallacy is that the annealing reaction is similar to that used for the annealing of steel and, therefore, the malleable-iron manufacturer is taking too much time for this process. An automotive engineer of my acquaintance once insisted most strongly that we were wrong in taking 9 days to heat-treat castings. He said he would prove this to us by taking a casting in the evening and returning it completely annealed in the morning. That was more than a year ago and he has not yet returned. Steel can be annealed in a few hours, but the various stages in the graphitizing heat-treatment require definite and specific times and cannot be executed in a shorter time interval. All malleable-iron producers would arrange to graphitize the carbon completely overnight if that were possible. The process constantly is subject to study and experiment with a view to decreasing the time involved. So far, however, no great reduction has been found possible.

An attempt to hurry the annealing process results in the user obtaining an inferior product which impairs the reputation of the producer. Those who purchase material should remember that long annealing processes are executed at the expense of the manufacturer; obviously, they would not be carried out if they were not essential to the satisfactory completion of the product. It is absolutely necessary that demands for malleable-iron products be adequately anticipated to allow sufficient time for this process of manufacture.

The physical properties of normal malleable-iron under various circumstances are covered in my previous papers entitled Malleable Iron as a Material for Engineering Construction[3]; Some Physical Constants of American Malleable-Iron[4]; and The Effect of Machining and of Cross-Section on the Tensile Properties of Malleable-Iron.[5]

Machining Properties

The machining properties of malleable-iron are the subject of experiment in the research laboratory of the National Malleable Castings Co. The behavior of twist drills on other products has been the subject of experiment more especially by B. W. Benedict and W. P. Lukens, who gave data obtained in drilling gray-iron in their report entitled An Investigation of Twist Drills.[6] The only data on malleable-iron of which we have knowledge were secured in a very short investigation by Edwin K. Smith and William Barr, and reported in a paper on The Relation Between Machining Qualities of Malleable Castings and Physical Tests.[7]

Feeling that further work was requisite on drilling stresses when cutting malleable-iron, it was decided to begin the study of machinability in general by an investigation of these stresses. The variables, the effects of which are to be studied, include the following, those numbered from 1 to 6 being in reference to the drill and those from 7 to 11 in regard to the properties of the material.

(1) Diameter
(2) Rate of feed
(3) Speer
(4) Point angle
(5) Clearance angle
(6) Helix angle
(7) Chemical composition
(8) Tensile-strength
(9) Elongation
(10) Brinell hardness number
(11) Shore number

Since the life of the drill is not under observation, the chemical and physical properties of the drill steel were not significant and were assumed to be constant throughout. The effect of cutting compounds was not within the scope of the investigation.

The experimental procedure was to determine on the Olsen universal efficiency machine the torque and thrust of drills operating under various predetermined conditions. Reference is made to a paper by T. Y. Olsen on An Efficiency Testing-Machine for Testing Taps and Dies.[8] All the drills used were of high-speed steel, made and ground by the Cleveland Twist Drill Co. No drill

[3] See *Transactions of the American Foundrymen's Association*, vol. 27, p. 373.

[4] See *Proceedings of the American Society for Testing Materials*, vol. 19, part 2, p. 247.

[5] See *Proceedings of the American Society for Testing Materials*, vol. 20, part 2, p. 70.

[6] See Bulletin No. 103, Engineering Experiment Station of the University of Illinois, 1917-1918, vol. 15, No. 13.

[7] See *Transactions of the American Foundrymen's Association*, vol. 28, p. 330.

[8] See *Proceedings of the American Society for Testing Materials*, vol. 14, part 2, p. 541.

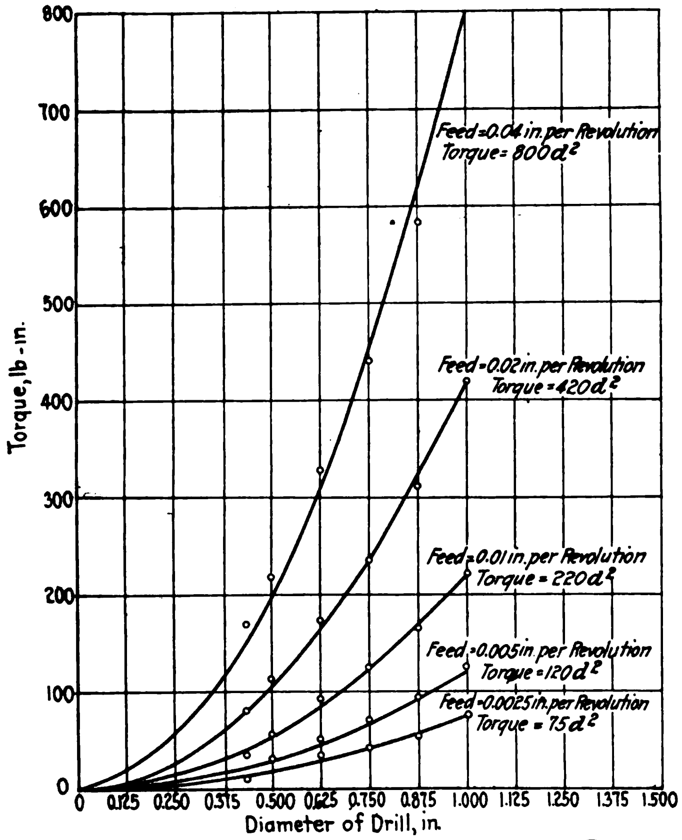


FIG. 3—VARIATION IN THE TORQUE WITH CHANGES IN THE DIAMETER OF THE DRILL AND THE RATE OF FEED

was used to a point where a change in stress that could be detected was produced. This factor was checked by drilling a standard malleable-iron piece from time to time, under standard conditions, as the investigation progressed.

For the investigation of items Nos. 1, 2 and 3, a large amount of malleable-iron from a single heat and a single annealing pot was available. The investigation comprised the drilling of this material with twist drills of standard form having diameters of 7/16, ½, ⅝, ¾, ⅞ and 1 in. at feeds of 0.0025, 0.0050, 0.0100, 0.0200 and 0.0400 in. per revolution at a speed of 240 r.p.m.; and the drilling of the same material with a standard ¾-in. twist-drill at the same range of feeds at speeds that were, as nearly as practicable, 40, 80, 160, 320 and 640 r.p.m. A few runs were made with drills of other diameters to corroborate the conclusion that the effect of speed did not vary greatly with drill diameter.

For the investigation of items Nos. 4, 5 and 6, a new lot of malleable-iron was employed which, from tests made with standard twist-drills, was known to be identical with the first in resistance to drilling. Instead of using the standard drill, which has a point angle of 118 deg., a clearance angle of 12 deg. and a helix angle of 27.5 deg., nine special drills, each ¾ in. in diameter, were provided, that had point angles of 98, 118 and 138 deg. and clearance angles of 5, 10 and 15 deg., respectively. The helix angle was 27.5 deg. in each case. A straight-fluted ¾-in. drill having standard point and clearance angles was provided also.

For the investigation of items Nos. 6 to 11 inclusive, 179 specimens of regular and special malleable-iron were available. These were made by the several plants of the National Malleable Castings and the Eastern Malleable Iron companies and by the Dayton, the Northern, the Erie and the Trenton Malleable Iron companies. Our thanks are due these organizations for placing at our disposal material of divers origins and processes of manufacture. This material represented practically the complete range of quality commercially attainable in the product. Some low-grade specially made material was also used. The metal was all completely graphitized. The chemical composition and tensile properties were determined by the foundry. The Brinell hardness and Shore numbers were obtained in our own laboratory.

Each material was tested with two standard ½-in. drills at 240 r.p.m. and a 0.005-in. feed. The material was drilled also by each of two ½-in. drills running at 240 r.p.m. with a constant pressure of 220 lb. on the drill point; the torque and the penetration per revolution were recorded. It is obvious that the detailed results of such an investigation are too voluminous to be given and the essential data have therefore been reduced to graphic form.

GRAPHIC TEST-RESULTS

Fig. 3 shows the relation of torque to diameter and feed as a series of parabolas, one for each rate of feed, correlating the diameter and the torque.

Fig. 4 shows the relation of the thrust to the diameter and the feed as a series of flat curves, one for each rate of feed, correlating diameter and thrust. The thrusts developed by the smaller drills, especially at low rates of feed, are so low in value as to render the data somewhat uncertain.

Fig. 5 shows the relation between the torque and the thrust of a ¾-in. drill as related to the speed, a separate line being shown for each rate of feed. A few tests for ½ and 1-in. drills were conducted, tending to correlate

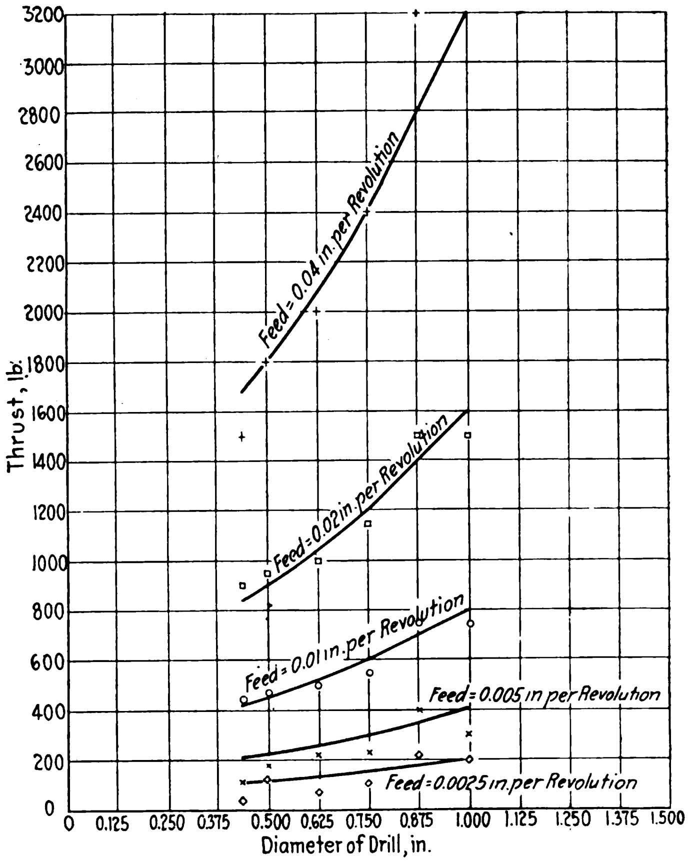


FIG. 4—VARIATION OF THE THRUST, WHICH IS ASSUMED TO BE A CURVILINEAR FUNCTION OF THE DIAMETER OF THE DRILL, WITH CHANGES IN THE RATE OF FEED OF THE DRILL AND ITS DIAMETER

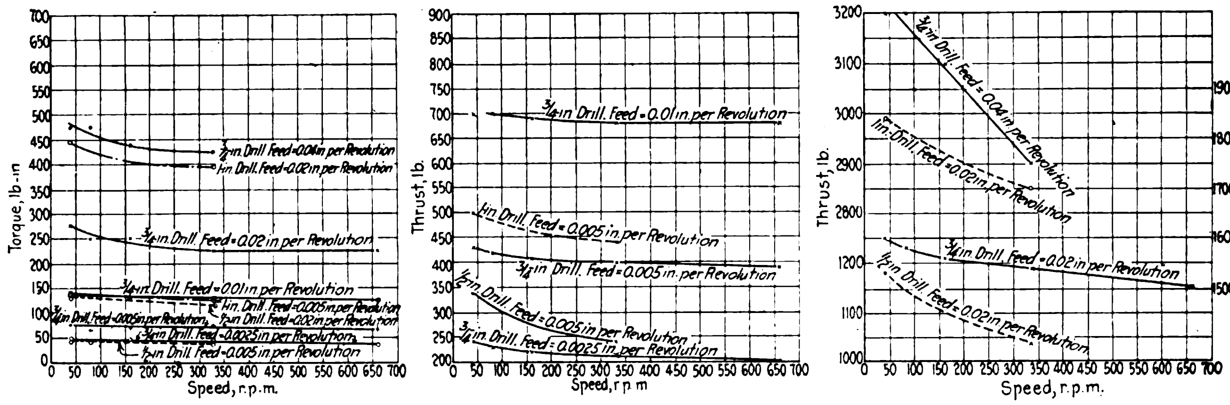


FIG. 5—TORQUE AND THRUST OF A ¾-IN. DRILL AT DIFFERENT SPEEDS AND FEEDS
Results of Some Tests Made with ½ and 1-In. Drills That Were Conducted To Correlate and Corroberate the Conclusions Based on the Tests of the ¾-In. Drill Are Also Presented

and corroborate the conclusions based upon the ¾-in. drills.

Fig. 6 shows the torque and the thrust of drills of various point and clearance angles as related to speed. Each chart applies to a drill of a given point-angle, the thrust or the torque being plotted against the speed. A different symbol was used for each of three clearance-angles.

Fig. 7 shows the torque and the thrust of the straight-fluted drill plotted against the speed at two different rates of feed and, for comparative purposes, the same data for a standard twist drill. Thus, Figs. 3 to 7 record the available data on the drilling conditions as a variable and permit fairly accurate conclusions as to the load conditions on drills of known diameter and form, working on the standard material under known conditions of feed and speed.

EFFECT OF THE MATERIAL BEING MACHINED

In considering the effect of the material being machined as comprised in items Nos. 7 to 11 of the program of investigation, it must be borne in mind that these five factors are not entirely independent variables. Assuming complete graphitization, which is justifiable with good commercial metal and known to apply to our present material, the physical properties of the product are determined primarily by its carbon-content. Were it not for the minor effects of variations in other chemical elements present and in thermal history, the last four variables would be functions of the carbon-content alone and bear definite relationships among themselves.

The tensile-strength, elongation and Brinell hardness number increase as the carbon decreases regularly, even under commercial conditions. The Shore number is practically constant for all samples investigated. Therefore, it was to be expected that graphs plotting drilling stresses against items Nos. 7 to 10 would be geometrically similar, and the Shore number would find no application in determining the physical properties of the material. Our tests were all made under such conditions that only material at least ⅛ in. from the surface was examined. Under

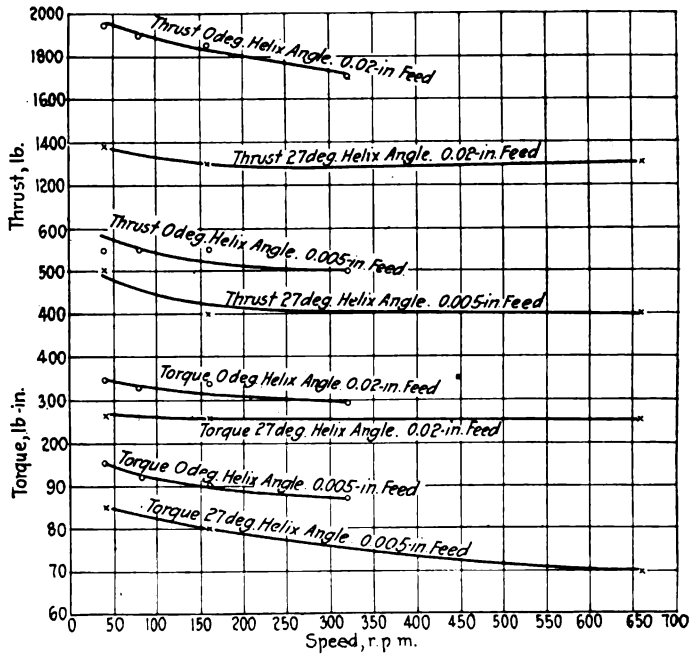


FIG. 7—TORQUE AND THRUST OF STANDARD TWIST AND STRAIGHT FLUTED DRILLS IN MALLEABLE IRON

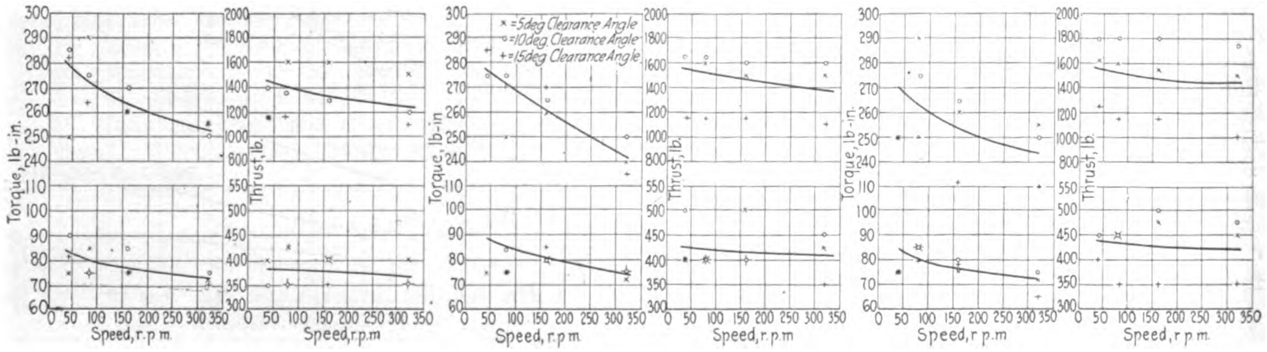


FIG. 6—RELATION BETWEEN THE TORQUE, THRUST AND SPEED OF DRILLS HAVING VARIOUS POINT ANGLES
The Point Angle in the Two Sets of Curves at the Left Was 98 Deg., for the Middle Pair It Was 118 Deg., and for the Curves at the Right It Was 138 Deg.

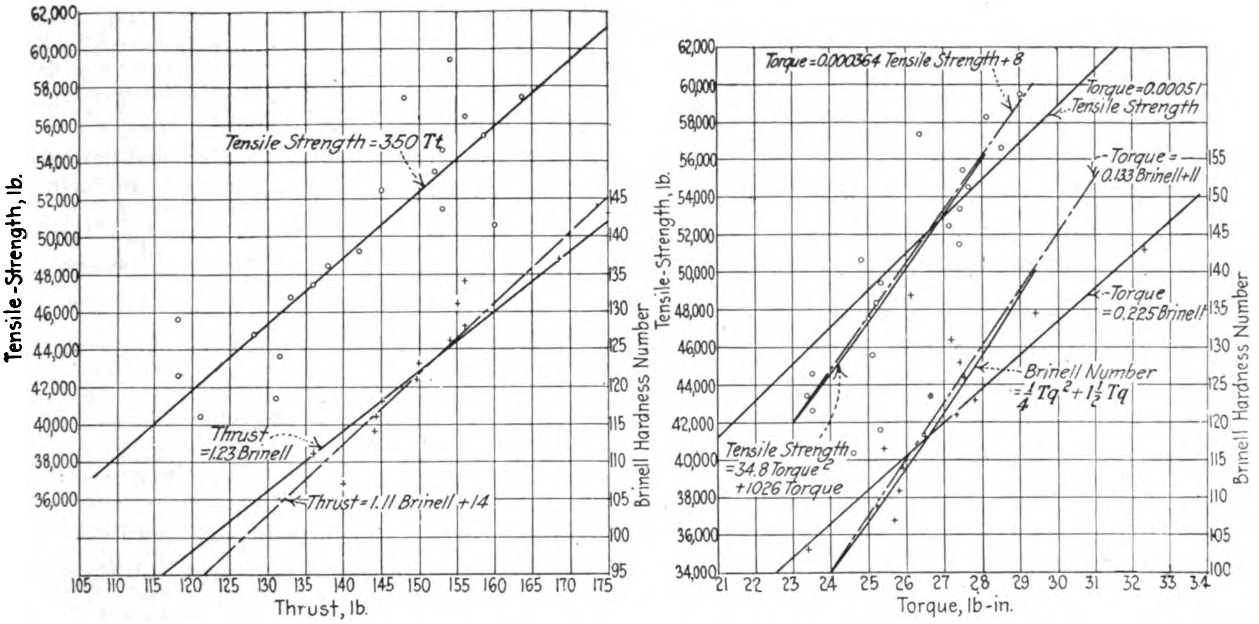


Fig. 8—At the Left Curves Showing the Relation Between the Thrust of a ½-In. Drill and the Brinell Hardness Number and the Tensile-Strength of the Iron and at the Right Similar Curves for the Torque

these circumstances, no connection between decarburization and machinability could be traced, even if it existed. Item No. 7 can thus be dismissed with the statement that items Nos. 8 to 10 are expressions of this variable. Item No. 11 can be discarded on account of the following considerations affecting the Shore number. The usual physical properties of malleable iron are determined primarily by the relative amount of ferrite and temper carbon in the mass. The Shore test, however, is made upon an almost microscopic area comprising only ferrite, and it is therefore a measure of the properties of the ferrite only. These properties would be affected but little by the presence of the usual amounts of alloy.

In Fig. 8 the thrust and the torque of the ½-in. drill running at 240 r.p.m. are plotted against the tensile-strength and Brinell hardness number, each point representing the average drill-stress for a group of specimens of constant strength or hardness as the case may be. A similar compilation against elongation is omitted as superfluous, since elongation and tensile-strength are interdependent and the latter property is measurable more accurately. Omitting the derivation of the several formulas in the interest of brevity, the data of Figs. 3 to 8 lead us to the following general conclusions that apply to completely annealed malleable-iron castings.

Let

- a = A constant depending upon the feed
- B = The Brinell hardness number of the malleable iron
- b = A constant depending upon the diameter of the drill
- d = Drill diameter in inches
- f = Rate of feed in inches per revolution
- P = The thrust in pounds
- s = Speed in revolutions per minute
- T = The drill torque in pound-inches
- U = The ultimate-strength of the malleable iron in pounds per square inch
- W = The work done in drilling, in foot-pounds per cubic inch

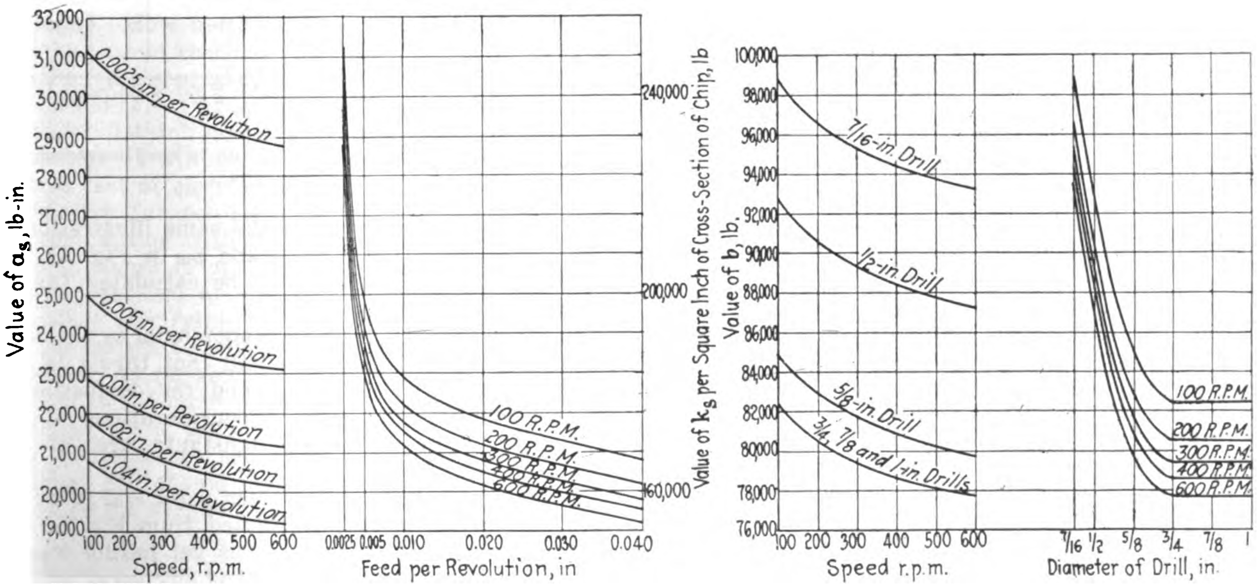


Fig. 9—Curves Giving the Values of a_s, k_s and b for Various Speeds

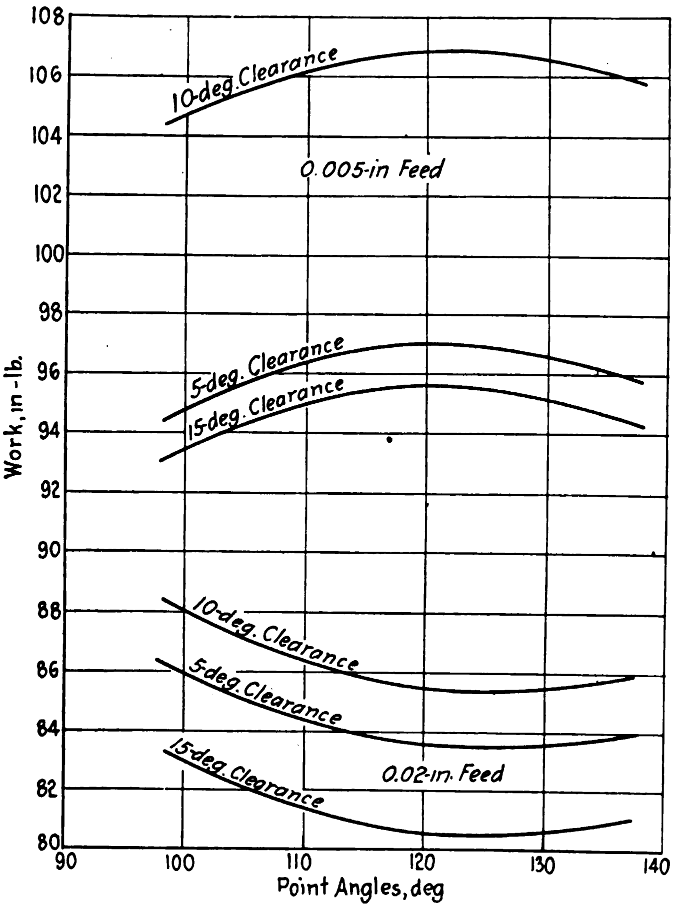


FIG. 10—CURVES GIVING THE AMOUNT OF WORK REQUIRED TO PENETRATE MALLEABLE IRON A DISTANCE OF 1 IN. WITH ⅝-IN. HIGH-SPEED DRILLS OF DIFFERENT POINT AND CLEARANCE ANGLES

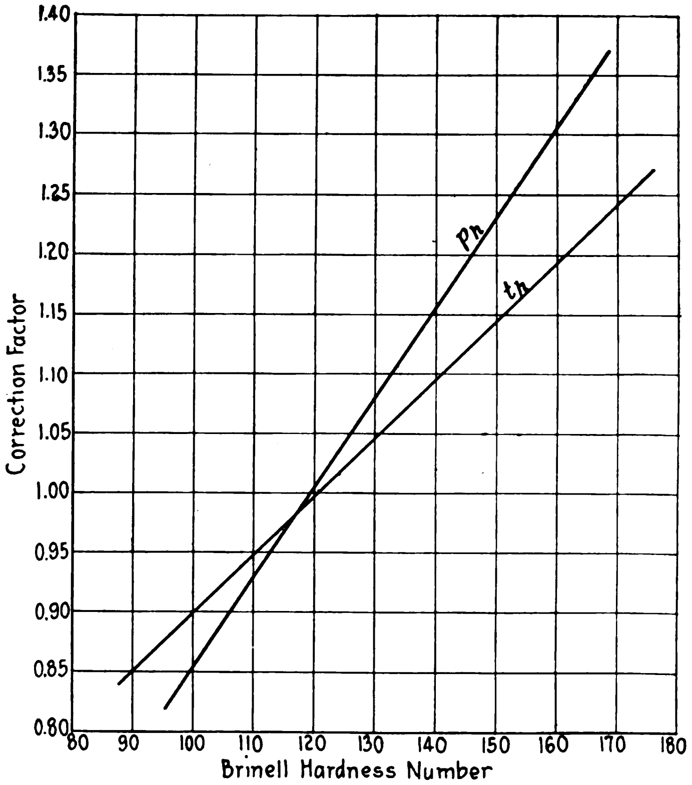


FIG. 11—CORRECTION FACTORS FOR p_h AND t_h FOR VARIOUS BRINELL HARDNESS NUMBERS

The values of a and b are shown in Fig. 9. Then, for drills of standard form, we have

$$T = (0.0049\,B + 0.409) \times (167.06/s^{0.04597})\ a \times fd^2$$
$$P = (0.00755\,B + 0.0952) \times (937/s^{0.03450})\ b \times fd$$
$$W = 8\ (0.0049\,B + 0.409) \times (167.06/s^{0.04597})\ a + (0.00755\,B + 0.0952) \times (937/s^{0.03450}) + (4b/\pi d)$$

where T, P and W are in terms of Brinell hardness number. Or, in terms of ultimate-strength, we have

$$T = (0.0000135U + 0.297) \times (167.06/s^{0.04597})\ a \times fds^2$$
$$P = (U/52{,}000) \times (937/s^{0.03450})\ b \times fd$$
$$W = 8\ (0.0000135U + 0.297) \times (167.06/s^{0.04597})\ a + [(U/52{,}000) \times (937/s^{0.03450})] + (4b/\pi d)$$

To simplify these equations, let

$$(167.06/s^{0.04597})\ a = a_s$$
$$(937/s^{0.03450})\ b = b_s$$
$$0.0049B + 0.409 = t_h$$
$$0.00755B + 0.0952 = p_h$$
$$0.0000135U + 0.297 = t_u$$
$$U/52{,}000 = p_u$$

Substituting these values, we have

In Terms of Hardness	In Terms of Ultimate-Strength
$T = t_h \cdot a_s \cdot fd^2$	$= \ t_u \cdot a_s \cdot fd^2$
$P = p_h \cdot b_s \cdot fd$	$= \ p_u \cdot b_s \cdot fd^2$
$W = 8t_h\ a_s + 4\ (p_h \cdot b_s/\pi d)$	$= \ 8t_u\ a_s + 4\ (p_u b_s/\pi d)$

For convenience, values of a_s are plotted at the left of Fig. 9; of b_s at the right of the same illustration; of t_h and p_h in Fig. 10; and of t_u and p_u in Fig. 11. From these values, T, P and W can be calculated for known values of the several constants.

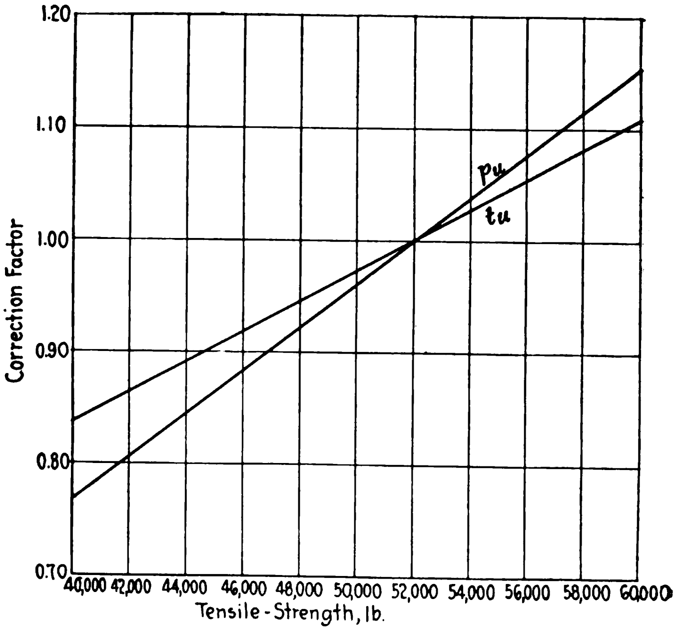


FIG. 12—CORRECTION FACTORS FOR p_u AND t_u FOR DIFFERENT TENSILE-STRENGTHS

That the torque should be proportional to the feed and the square of the diameter; and that thrust is proportional to feed and diameter; and the derivation of W from T and P, are based on considerations of applied mechanics. The numerical constants are, of course, purely empirical.

In Fig. 12 the relation between work and form of drill point is summarized, as calculated from Fig. 6. Generalizations as to drill form are as yet hardly warranted. From considerations of mechanical efficiency, an increase

in the helix and the clearance angles to the highest values consistent with drill wear and strength would seem to be indicated. The effect of the point angle seems unexpectedly small.

By substituting in the formulas for t_h, p_h, t_u and p_u, the extreme values of B and U likely to be encountered it will be observed that the difference in torque between the weakest and strongest irons reasonably likely to be obtained is approximately 30 per cent of the average value. The thrust may vary perhaps 40 per cent between the strongest and weakest commercial irons.

THE DISCUSSION

W. R. STRICKLAND:—What progress has been made in casting thin sections?

H. A. SCHWARTZ:—Malleable iron originally was developed for use in small, thin castings. The recent tendency has been toward heavier sections. I have seen castings where perhaps 20 would be required to weigh 1 oz.; however, such work is very unusual. To avoid cooling strains that may crack the casting, it is necessary to avoid abrupt changes from thin to thick sections.

A MEMBER:—The aluminum foundries have become expert in the use of chills to prevent the cracking of castings. Is there any such art in the making of malleable castings?

MR. SCHWARTZ:—Chills have been used for many years to keep out cracks and also to remove large shrinks. Perhaps I should have spoken of this more fully. One can make perfectly good malleable-iron and still make a very bad casting. For instance, in the automotive trade malleable-iron, when properly produced, is the best possible material for making hubs; but when improperly made, it is about the worst. The reason is that if the work is entrusted to a producer who desires to make the most castings for the least money irrespective of quality, a hub may be produced that is 95 per cent hub and 5 per cent air; the 5 per cent represents the volume of shrink. Being at the point of junction of the barrel of the hub with the flange, these shrinks weaken it at its most critical point and easily may be a cause of failure. The remedy is not in better iron, but in better foundry practice in avoiding shrinkage by the use of suitable feeders. This method, however, costs money and is not likely to be practised by those who feel it necessary on account of competition to cut prices to the limit.

A MEMBER:—Have you experimented with malleable-iron pistons? How much will electric-furnace practice increase the cost of castings? Is the element of manganese considered important?

MR. SCHWARTZ:—Our company has not experimented with pistons. I think malleable-iron possesses no particular advantage. I will not undertake any statement as to the relative cost of electric-furnace and air-furnace malleable. Electric-furnace malleable has been sold continuously in competition with the good grades of air-furnace metal. Manganese is of importance on account of the unavoidable presence of sulphur; but I warn any consumer against specifying the chemical properties of the material he desires to buy. The user buys the material on account of its physical or engineering qualities and is interested in the material possessing the qualities that are required. He is usually not well informed as to the method of manufacture, certainly not as well informed as the producer, and should refrain from telling the latter how to secure the results.

A MEMBER:—Is malleable-iron suitable as a bearing metal, for heavy loads?

MR. SCHWARTZ:—Malleable-iron is not suitable for bearings, but there is a specific case in the design of the trucks under a freight or passenger car where the journal-boxes ride against the truck column. In this construction malleable-iron is used. In other words, its resistance to wear is good enough so that its utility for purposes where wear resistance is an incident will not be destroyed. It should not be recommended purely as a bearing material, so far as we know now.

FERDINAND JEHLE:—A piston must be made of good bearing metal, and have walls of such cross-section that they will conduct away some of the heat. The mere fact that it might be possible to make it very thin would not mean that it would be a good piston.

MR. SCHWARTZ:—I do not see, offhand, how one could cast a thinner piston of malleable-iron than of gray-iron, but I think one would have no trouble with it.

E. T. BIRDSALL:—Does it take longer to anneal castings weighing 50 to 60 lb. and from ½ to 1 in. thick, than thinner sections? Would it cost more or less to machine malleable than steel castings?

MR. SCHWARTZ:—The heaviest casting of which I have knowledge weighed 896 lb.; it was a transmission housing on a military tractor. A thickness of ½ in. is not considered thick; about 1 in. is considered moderately thick, although much heavier sections have been made. Thickness, as such, has no connection with annealing time, although for other reasons fairly heavy castings take longer to anneal than small ones. I think that, as a general rule, a malleable casting of ordinary size, including machining, will cost less than the corresponding steel casting, the difference being largely in the cost of the machining.

A MEMBER:—You mentioned the wear of malleable axle-housings on railroad-car service. How does the wear compare with that of steel journal-box housings? Is cast-iron used for this purpose?

MR. SCHWARTZ:—Cast iron has been used, although malleable-iron makes the better journal-box. Cast-steel journal-boxes have been made by our company, but they are not common. I know of no figures showing the comparative wear of the several materials. We are now working along these lines.

A MEMBER:—Is there any way of filling porous places in malleable castings?

MR. SCHWARTZ:—They can be filled with compounds such as Smooth-On. They can be filled also by bronze welding or brazing with the acetylene torch. The producer can fill the pores by acetylene welding with white cast-iron and reannealing the castings. In view of the difficulty of producing perfect welds, it is doubtful whether this practice should be resorted to at critical points in a casting. Our company is opposed to it.

A MEMBER:—Does your company request manufacturers to state where they wish the sprue put?

MR. SCHWARTZ:—Not in exactly that form. As a rule, the buyer of a casting is not sufficiently well acquainted with foundry practice to have an opinion of value on this point, but we welcome an expression from him as to where the important points are from which it is necessary to exclude shrinkage.

A MEMBER:—Is it possible to nickel-plate malleable castings?

MR. SCHWARTZ:—That is a very common practice.

Research Topics and Suggestions

THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.

GEARS

THE subject of gears and gear performance is too broad to permit of more than a few suggestions as to some of the phases that appear to need further study from the research standpoint, particularly in view of the demand for a certain amount of standardization in gear practice in the automotive field.

For the purpose of this discussion, we shall omit all of the many questions of gear design, such as tooth contour, pitch and pressure angle, all of which have been dealt with at length in the literature and probably are well known to all who have made a study of the design and production of high-duty gears.

The topics that seem to have received the least attention in the literature of the industry, and probably have received correspondingly less attention from engineers, are those of gear performance. The performance requirements of gears for automotive use may be classed under the headings

(1) Strength	(3) Quietness
(2) Wearing quality	(4) Mechanical efficiency

The relative importance of these four qualities depends entirely upon the type of gear and class of service. For instance, in tractor gearing, wearing quality is likely to be of the greatest importance, whereas for a passenger-car transmission, adequate strength and reasonable quietness may be demanded, and for front-end gears, quietness alone may be the major requisite.

A review of the published material on the foregoing topics reveals a disappointingly small amount of important matter that deals directly with them.

Strength

Nearly all of the many formulas for computing the strength of gears are based on the formula published by Wilfred Lewis in a paper that appeared in the *American Machinist*, May 4, 1893. The author states that at that time there were some 48 different formulas for the strength of gear teeth, differing by factors of as much as 500 per cent.

The Lewis formula, as commonly used, consists of two parts, one of which applies to static loads, the other to the increase in tooth pressure with speed. The former part is an exact, or "rational" formula based on certain assumptions. If a given load is applied uniformly across the tip of a tooth, in a tangential direction, the maximum fiber stress is given by the formula with only the assumptions necessary in computing the stresses in a beam. However, these assumptions are open to question, and the formula cannot be more nearly correct than the assumptions on which it is based.

The second portion of the formula is a factor depending upon speed and is based on the assumption that the maximum tooth-pressure for a given average torque increases in proportion to the speed with an arbitrary factor of proportionality, the basis of which the author does not give. This portion of the formula is not exact or rational, but is supposedly based on the results of experiments that must have been made prior to 1893; probably, therefore, on gears of entirely different types from present-day automobile gears. It is hardly to be expected that the Lewis formula, including this factor, would apply to modern practice.

So far as we have been able to determine from published data, there are no formulas for gear strength in use at present that give entirely trustworthy results, and most of the gears designed primarily for strength seem to be laid out on the basis of experience alone. This is by no means a satisfactory condition, and it cannot be remedied until we can provide answers to the following questions:

(1) Are the assumptions on which the Lewis formula is founded well based, and what modifications are necessary to make it, or any similar formula that may be developed, more readily applicable to modern practice?

(2) What is the effect of speed in increasing the pressures on gear teeth beyond those due to the average torque? Is this increase proportional to the speed or to the square of the speed? To what extent does it depend upon changes in angular velocity, and on the moment of inertia of the rotating parts?

It seems rather surprising that so little experimental work has been done on the latter problem. While a few isolated experimenters have improvised means for measuring the relative and actual angular velocities of a set of gears, so far as we have been able to learn, no instruments or methods have been developed which would facilitate a general study of this subject Yet without such a study it is impossible to determine the effect of speed on the maximum stresses set up in gearing A few of the more recent articles dealing with this phase of the subject are

The Strength of Gear Teeth. Guido H. Marx, *Journal of the American Society of Mechanical Engineers*, Vol. 35, p. 109.

This article describes an elaborate series of tests of strength of cast-iron gears under load at various speeds. Results are tabulated at length, and causes given for the relation of load to breaking stress.

A peculiar result is that breaking stress is a minimum at a moderate speed, and increases with higher speeds on the two curves given. Comparing the actual figures of breaking stress for speeds of 0, 100, 300 and 600 ft. per min. with the Lewis formula, actual factors of safety run from 5.5 to 12.2. Static breaking-stresses were measured for one-tooth and for two-tooth engagement by loading the cast-iron gear against a steel gear with teeth cut away except for the number required for contact. Breaking stresses were found to be very much in excess of those to be assumed from the Lewis formula. The author criticises the Lewis formula, showing several important omissions, such as neglect of distribution of stress between different teeth and neglect of angularity of load application. This is an important and rather carefully prepared article, and should be of interest in connection with any study of gear strength.

Tests of Strength for Gear Teeth. Andrew C. Gleason, *Machinery*, January 1914, p. 382.

Mr. Gleason describes the results of a special test for the breaking strength of gear teeth, the gears being held in a special chuck and one tooth subjected to load. A special micrometer was used to detect distortion. Figures are given for a gear, 1-in. face, 2⅓-in. pitch diameter with 14 teeth of 6 pitch, loaded at the tip of the tooth. The breaking stress varies on six different materials with different heat-treatment from 9000 to 22,450 lb. per sq. in.

Strength of Gear Teeth. S. J. Berard, *American Machinist*, Nov. 27, 1919, p. 925.

This is a discussion of gear-tooth strength on the basis of the Lewis formula. The author points out that the strength

is not the same for gears of different sizes. He gives a chart of values of T and F in the Lewis formula, where T is the beam thickness and F is the face width. He uses Barth's formula for the effects of speed.

Approximate Method for Determining the Strength of Gear Teeth. Willard A. Thomas, *American Machinist*, Aug. 7, 1919, p. 273.

The author quotes the Lewis formula, and develops a more simple practical formula making some simplifying assumptions. The proposed new formula might be of value from its simplicity when applied to cast-iron or bronze gears used in general machinery practice. It is not capable of general application.

Convenient Forms of the Lewis Formula for the Strength of Gear Teeth. J. H. Carver, *Machinery*, October, 1914, p. 99.

This is a brief paper giving some examples and supposed simplifications of the Lewis formula.

Wearing Quality

For classes of gearing operating at anywhere near full load, for long periods, the rate of wear, rather than strength, may be the determining factor in design. It is, perhaps, natural that not much experimental work has been done on the subject of gear wear, since it is affected by important circumstances over which the designer has little control, such as the kind of lubricant and the amount of abrasive material present, as well as the sort of service required. Some valuable contributions on this subject have been made by Joseph Jandasek, in a series of articles entitled Gear-teeth Sizes from the Standpoint of Durability, published in *Automotive Industries* June 10 and 17, 1920, pp. 1305 and 1402.

In these papers the author points out some of the limitations in the computation of gear dimension on a basis of strength, as for instance the effect of tooth deflection on stress distribution, and maintains that unit surface-pressure is of equal importance with total pressure in determining rate of wear. Wear is not proportional to unit pressure, but is small up to a certain critical pressure beyond which it increases very rapidly.

The author calls attention to the various types of contact between curved surfaces that occur in machinery practice, such as plain and roller cams, ball and roller bearings and gear teeth, and taking as his basis the formulas of Hertz, develops an expression for the compressive stress at the contact surface for the several forms of contact.

He discusses at some length the relations between the results of the Lewis formula and practice with gears of different materials, and proposes a speed increment proportional to the square of the speed instead of to the first power as proposed by Lewis, and states his reasons for considering this a more rational assumption.

A discussion of the comparative merits of case-hardened versus tempered gears follows. Case-hardened gears are recommended for constant-mesh service, and tempered gears for change-speed. While case-hardened gears are less subject to pitting than tempered gears, pitting can be reduced in tempered gears by reducing the unit pressure and using a harder temper. The author develops formulas for unit pressure on gear teeth, showing that this pressure depends upon the elasticity of the materials. Gray-iron and bronze are cited as good gear materials for moderate loads owing to their high deformability.

An example is given to illustrate the calculation of maximum compressive stress in a set of gears.

A second series of papers on gearing, Gearing Calculations by the Compressive Stress Method, by Joseph Jandasek, published in *Automotive Industries* Sept. 15 and 22, 1921, pp. 512 and 564, is a further development of the subject of the relation between unit pressure and rate of wear. Resistance of metals to wear depends upon (a) hardness where the particles of metal are not readily displaced and (b) toughness where the particles of metal are not easily removed, even if displaced.

The remainder of the paper deals with one of the above-mentioned topics, the effect of speed on the strength of gears. The author discusses at some length the causes of increase in the load with an increase in the speed and proposes a new empirical formula based on his own observations of variation in angular velocity. Increments in load are due to

(1) Accelerations, both positive and negative, due to
   (a) Irregularity of tooth outline
   (b) Variable gear-tooth and surface deflection
   (c) Wear
(2) Accelerations, or deflections due to shock or oscillation (Resonance effects)

The author prefers his formula, in which the load is assumed to increase with the square of the speed, to that of Lewis, in which the load is assumed to increase as the speed, because

(1) Accelerations must increase *with higher power of speed*, about the square; moreover, no gears are safe at high speeds
(2) Weight of parts increases with load
(3) Deflection increases with load

He then develops a new formula for load capacity differing from one given in a previous paper, and based on the assumption of a maximum allowable comprehensive stress at the tooth face, and discusses the influence of

(1) Gear ratio
(2) Quality of material
(3) Application to helical gears

These papers are too long and too important to one interested in the subject of gear wear, to permit an adequate review here. They should be read with care.

Another paper dealing with this subject is An Investigation of Tooth Wear with Automobile Gear Steels, by E. R. Ross, in *Automotive Industries*, Nov. 3, 1921, p. 865, in which the author reports the results of tests of gears of three classes, tempered gears with two degrees of hardness, and case-hardened gears still harder than the former. He recommends a scleroscope hardness of 75 or over to secure minimum wear.

An article on Spur-Gear Erosion, by F. W. Lanchester, in *Engineering* for June 17, 1921, provides an interesting discussion of the causes of spur-gear erosion, and suggests remedies.

The author points out that rolling in place of sliding contact between spur gears has been considered the ideal condition. Contrary to a common misconception, the relative amounts of rolling and sliding contact cannot be modified. He refers to tests with the Daimler-Lanchester dynamometer, in which worm gears, with which there is only sliding friction, showed surprisingly high efficiencies.

This introduces a discussion of a type of abrasion observed on high-duty spur gears, which points to a conclusion opposed to the common belief. On examining a number of pairs of gears, where the commonly observed wear was in the initial stages only, he found that there was an area near the pitch-line where the surface had become concave. This area always occurred, not at the pitch-line, but slightly toward the tip of the tooth, so that the worn areas did not come together on the mating gears, but the shoulder of each wore into the other.

The author presents the explanation that this can result from actual metallic contact and abrasion between the gears at portions of the surface where sliding contact occurs at a very low rate of speed. He points out that this type of wear is particularly productive of noise, as it introduces decided irregularities in angular velocity. Since noise seems to be produced even by perfect gears, he suggests that it may result from the reversal of stress, as the sliding contact becomes rolling contact and then reverses its direction, thus giving rise to a definite periodic force that necessarily must be transmitted to the gears and shafts, and may produce resonance effects in any of these parts that have a similar natural period of vibration.

Two possible remedies for the observed type of wear are suggested. One is the adaptation of types of gearing in which there is only sliding contact. This does not seem practicable at present, on account of size limitation for transmission gears and the like. The other suggestion is the adoption of metals that show less tendency to adhere in contact. Metals of dissimilar character usually behave better in this respect and, while the author sees no likelihood of using dissimilar metals for transmission gears, he suggests that much might be gained by a study of the behavior of dissimilar steels for this purpose. He does not take up the subject of the quality of lubricant as affecting the problem, but this too might be a fruitful field for further research.

Some topics that call for further study in connection with the relation of gear wear to gear life are

(1) Gear material; hardness and structure
(2) Tooth form; radius of curvature and unit pressure
(3) Peripheral speeds; as they affect both the tooth pressures and the impact and rubbing velocities
(4) Lubrication; effect of lubricants on gear wear, and the theory of gear lubrication
(5) Abrasive wear; the effects of the usual forms of foreign matter are little known

The causes of noise and the question of gear efficiency are left for discussion at a later date.

(To be concluded)

MUTUAL BENEFITS OF FOREIGN TRADE[1]

BEFORE the war a great international system of beneficial trade had been built up, gradually and naturally; we scarcely realized how. The industries of the world had become in large degree interdependent. Western Europe was densely populated and highly industrialized. It was the focus of the world's exchanges. It was constantly sending out great quantities of manufactured goods in exchange for raw materials and foodstuffs. The war demoralized the industries of Europe and broke down the system of exchanges. Even within Europe the old channels of trade have been blockaded. The old Austro-Hungarian Empire, within which trade formerly was free and unrestricted, was divided into six independent states, each of which proceeded forthwith to erect high customs barriers against the others. Russia was formerly a great source of foodstuffs and raw materials for Western Europe, for which payment was made in manufactures; that trade has disappeared. The relations between Europe and the rest of the world have been interrupted in the same manner. India is a great tea-producing country, and Russia was one of the chief markets for tea. The inability of Russia to take its usual quantity of tea has prostrated the tea industry of India, and the inability of India to sell her products has cut down her purchases of cotton goods in England, and finally the inability of England to sell cotton goods has reduced her purchases of raw cotton in the United States. And so in this and other ways the European situation reacts upon us.

Our territory is so extensive and our resources so varied that we are in better position to live within ourselves than any other country. If our industries had been developed with that in view we might have been able to get along within ourselves, not completely, but more fully than at present. But our industries have been developed as a part of the industrial organization of the world. Although our exports are but a small part of our aggregate production, they represent great industries in which many millions of our people are engaged; and the price obtained for the portion that is exported affects the price for the whole production. Cotton mills of the United States have capacity to work up only about one-half of our cotton crop; the remainder must be exported. The lands that are growing cotton cannot be shifted to other crops without over-production of those crops, for we are exporters of all the principal agricultural products. In short, it is not practicable to shift the population into the new industries that would be necessary for the United States to live within itself. Furthermore, it is safe to say that no such policy would ever satisfy the ambitions of the American people or would be long maintained even if adopted.

A state of trade with balances running continually one way is abnormal and cannot possibly be permanent, because the debtor country cannot find the means of payment. Its efforts to procure the means of payment are certain to send exchange to a premium, and the premium will rise until it puts a check upon purchases in the creditor country and so brings the trade back into balance. Exchange rates act as an automatic governor. We are accustomed to say that the exchanges are in our favor when the dollar rates above other currencies; and they are in our favor for buying purposes, but not for selling. They help us to make purchases in other countries, but they do not help us to sell in other countries. On the contrary, they are a handicap upon sales.

The fact that international payments must be made for the most part in goods is very well illustrated by the difficulties that arise in the case of the German reparations payments. Most of us would like to see Germany pay, as far as is humanly possible, for the damages done by her armies and on the sea during the war; but many people do not understand the difficulty about paying great sums in another country and in another money. The reparations payments must be made in gold or in goods, and Germany has but little gold. The payments cannot be made in German paper money, for that is worthless outside of Germany and not worth much in Germany. If you were to visit Germany and see the many signs of wealth in the country, the fine cities, the great industries, the railroads, the forests and mines and farms, you might conclude that Germany was able to pay large sums; but none of these kinds of property can be transported out of Germany.

We have almost the same problem in the indebtedness running from the foreign governments to the United States Government. Some of our people are eager to have the payments begin; it has been proposed that the payments as received be applied to the payment of adjusted compensation to our ex-soldiers of the war. The people who urge this policy are apparently thinking that the payments will be made in money; but the payments cannot be made in money. The debtor countries are all collecting their taxes in depreciated paper currency; and even if they were able to collect from their taxpayers the sums nominally sufficient to make the payments to the United States, they could not make them in their paper currencies, because we have no use for those currencies. Those countries must export goods, which will create credits against which they must draw; and the very congressmen who have been urging that pressure be exerted upon the foreign governments to hasten payments, have been engaged at the same time in the preparation of a new customs tariff designed to reduce the offerings of foreign goods within this country.

As good merchants we are bound to give some attention to the means at the command of our customers for making payments. It is not enough that other countries shall want our products; they must be able to pay for them. The problem of payment is just as much ours as theirs, because they cannot pay without our help. The purchasing power of every

(Concluded on page 106)

[1] From an address delivered before the Export Managers' Club of New York by George E. Roberts, vice-president of the National City Bank, New York City.

Camber and Gather Relationships in Front-Wheel Alignment

By J. C. SPROULL[1] *Illustrated with* DRAWINGS

DEFINING the word "camber" to mean the tilting of the wheel spindles to make the wheels lean outward at the top, and the word "gather" to mean a second tilting of the spindles so as to bring the forward part of the wheels nearer together than the rear part, the author analyzes the effects produced by each of these operations and deduces mathematical formulas from which their proper values can be determined.

Tabular data and drawings are used in connection with the discussion of the effects of camber and gather on tire wear, and the use of the formulas and tables is explained in detail for an average case in which S, the slant height of the cone of which the wheel is the base, is taken as being approximately 500 in. In regard to misalignment, the author states that if this amounts to as much as ⅜ in., the life of a front tire may be reduced by many thousand miles.

THE wheels of ox-carts in olden times were "dished" for structural reasons. In order that the spokes between the hub and the road might stand perpendicularly to the road surface, the spindles were "cambered." This caused the wheels to lean outward at the top. It was discovered that both tractive force and tire wear of such a pair of wheels could be reduced materially by giving the wheels "gather"; that is, by again tilting the spindles so as to bring the forward part of the wheels nearer together than the rear part.

The front wheels of modern motor-cars have both camber and gather, although the wheels are of the artillery type without dish. Reasons for camber and gather of the front wheels of motor cars are given in books of instruction published by manufacturers, and elsewhere; and are well understood by motorists generally. While the reasons for camber and gather in the front wheels of motor cars are altogether different from the reason, mentioned above, for cambering the axle of an ox-cart, the relation existing between camber and gather is similar. It is the purpose of this article to discuss this relation.

ANALYSIS

A wheel that leans outward at the top, that is, a combered wheel, can be considered as the base of a cone lying upon its side as shown in Fig. 1. The "natural" path of such a wheel, therefore, when it rolls on a plane surface, is the circumference of a circle described about the apex of the cone as a center. Obviously, unless the front wheels have gather, there will be a tendency for them to separate when they roll on a plane surface; this will result in a mutual slight side-skidding of both wheels, the right-hand wheel skidding toward the left and the left-hand wheel toward the right.

It is seen that this constant side-skidding, even though comparatively slight, will, if continued indefinitely, prove to be destructive to both tire and road surface. Thus, if the tendency of a wheel to turn outward is represented by an angle of only 0.2 deg., which corresponds to a misalignment of ⅛ in. for a 35-in. wheel, or an error of ¼-in. in gather, the effect in traveling 1000

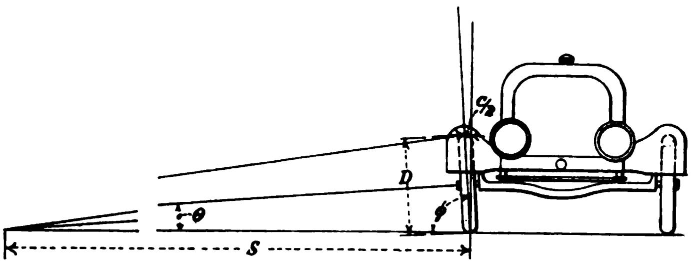


FIG. 1—DIAGRAM SHOWING HOW A CAMBERED WHEEL RESEMBLES THE BASE OF A CONE LYING ON ITS SIDE

miles will be equivalent to dragging the tire slowly sidewise under load a distance of more than 3 miles. If, however, the road surface at the point of contact with the tire is normal to the plane of the wheel, as might be the case when traveling upon a crowned road, then the wheel must be considered as a right section of a cylinder and its natural path will be a straight line. Therefore, the wheels should have no gather for this special case, even though they have camber. Where the road crown is excessive so that the angle a is less than angle b in Fig. 2, the gather becomes negative; that is, the wheels should "toe-out" rather than "toe in." Such a condition is extremely rare. The conclusion is that, for a given camber, gather should be less for vehicles that habitually travel over highly crowned roads than for those that commonly travel on comparatively flat roads.

Consider the case of a flat road and let

2α = Angle between two lines in the plane of the road intersecting at the apex of the cone and touching opposite ends of the area of contact between the tire and the road as shown in Fig. 3

β = Angle of gather for one wheel, in degrees

C = Wheel camber, equal to the difference of the distances between the tops and the bottoms of the vertical diameters of the tires, in inches

D = Diameter of the wheel, in inches

G = Gather, equal to the difference of the distances between the rears and the fronts of the horizontal diameters of the tires, in inches

L = Length of area of contact between the tire and the road, in inches

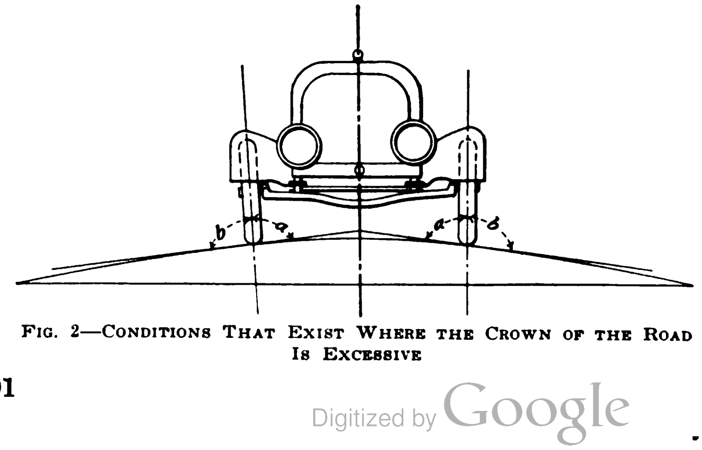


FIG. 2—CONDITIONS THAT EXIST WHERE THE CROWN OF THE ROAD IS EXCESSIVE

[1] Engineer of tests, B. F. Goodrich Co., Akron, Ohio.

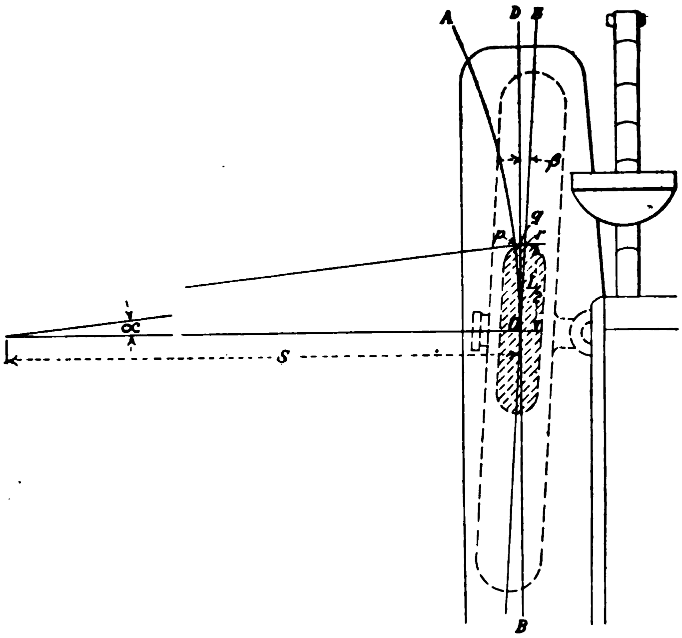


FIG. 3—DIAGRAM ILLUSTRATING THE RELATIONS THAT EXIST BETWEEN CAMBER, GATHER AND TIRE WEAR

Φ = Angle between the planes of the wheels and the road, in degrees
S = Slant height in inches of the rolling cone of which the wheel is the base, as shown in Fig. 1
Θ = Spindle camber, equal to one-half the apex angle, in degrees, of the rolling cone as shown in Fig. 1

Then, according to Fig. 1,

$$\cos \Phi = \sin \Theta = C/2D \qquad (1)$$
$$S = \tfrac{1}{2} D \sin \Theta = (D \times 2 \times D)/2C = D^2/C \qquad (2)$$

An examination of a number of representative motor vehicles indicates that the value of S can be taken as 500 in. In other words, the spindle camber Θ has been established by custom as approximately 2 deg. Accordingly we can write the simple relation $C = D^2/500$, which approximates the average case closely, and from which relation we compute the values shown in Table 1.

TABLE 1—CAMBER FOR VARIOUS WHEEL DIAMETERS

D, Wheel Diameter, in.	C, Camber, in.
30	1.80
32	2.12
34	2.31
36	2.59
38	2.88
40	3.20
42	3.52

CAMBER, GATHER, AND TIRE WEAR

To get a clear conception of the relation existing between camber, gather and tire wear, it is necessary to take into consideration the area of contact between the tire and the road, the length of this area being of greatest moment. This at once involves inflation pressure, as will be seen later. In Fig. 3, which is drawn in the plane of the road, the point O represents the center of the area of contact; the arc OA, the natural path of a cambered wheel that has no gather; the line BD, the direction of motion of the vehicle; and the angle β, the angle of gather. The slant height of the rolling cone is S, which we have seen can be assumed to be 500 in.

Without gather and without restraint, the point O presently will be at P. The restraint of the axle and the spindle will cause the point O to follow the path Oq, gather or no gather. Restraint exerted by the axle means tire destruction, since the restraining force must be transmitted to the road through the tire. If now we direct the wheel along the line OE so that the distance qr equals the distance Pq, then, for the region covered by the area of contact, no external restraint will be required since these two opposite tendencies are equal and their resultant is zero. Our problem then is reduced to the simple requirement of making $Pq = qr$. We have,

$$\tan \beta = (1 - \cos \alpha)/\sin \alpha \qquad (3)$$
$$\sin \alpha = \tfrac{1}{2}L/500 = L/1000 \qquad (4)$$
$$\cos \alpha = \sqrt{(1{,}000{,}000 - L^2)}/1{,}000{,}000 \qquad (5)$$
$$\beta = \tan^{-1}[\{1000 - \sqrt{(1{,}000{,}000 - L^2)}\}/L] \qquad (6)$$

Equation (6) is for values of L from 6 in. to 10 in., very nearly,

$$\beta = \tan^{-1} L/2000 \qquad (7)$$

Solution of equation (7) gives values for L and $\tan \beta$ as shown in Table 2.

TABLE 2—VALUES OF L AND TAN β

L, in.	Tan β
6	0.0030
7	0.0035
8	0.0040
9	0.0045
10	0.0050

Since the sine and the tangent are practically equal for the very small angle given in Table 2, we can also write

$$\tan \beta = \tfrac{1}{2}G/D = G/2D \qquad (8)$$

Hence

$$G = 2D \tan \beta$$

and we have the relations for L and G shown in Table 3.

TABLE 3—VALUES OF L AND G

L, in.	G
6	$0.006D$
7	$0.007D$
8	$0.008D$
9	$0.009D$
10	$0.010D$

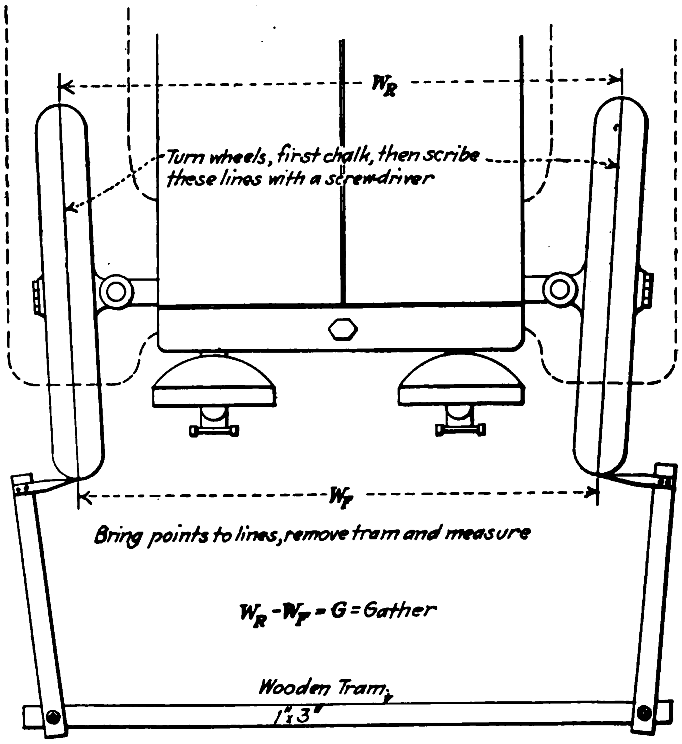


FIG. 4—TRAM FOR CHECKING THE ALIGNMENT OF THE FRONT WHEELS

(Concluded on page 116)

Variations in Modern Diesel-Engine Design

By THOMAS ORCHARD LISLE[1]

PENNSYLVANIA SECTION MEETING *Illustrated with* PHOTOGRAPHS AND DRAWINGS

SUBSEQUENT to an outline of the great number of different designs of Diesel engine and an indication of the many variations in their mechanical features, the author comments unfavorably upon the tendency of builders of Diesel engines to depart from the original design unnecessarily and believes the time has come to restrict this practice.

The illustrations include a number of the leading types of American and European marine Diesel engine and brief comment accompanies each one. The different modes of departure from original design are indicated. Combination and opposed-piston engines. mechanical-injection and double-acting Diesels, compound engines and engines representing the latest European practice are shown and described. Brief comment is made also on the subject of Diesel-engine fuel-consumption.

A SHIP owner who is not familiar with the motorships of today, and seriously considers the adoption of economical oil-engine power for his fleet for the first time, is confronted by an almost bewildering variety of Diesel engines from which he must make his selection, many of which differ radically in design from each other. All have certain advantageous claims, but there is little or no semblance of standardization in either framework construction or in methods of combustion.

There are more than 50 different designs of Diesel engine on the market. The designs include

- Two cycle, with
  - Air-injection
  - Airless injection
  - Scavenge valves
  - No scavenge valves
  - Combined scavenging and injection valves
  - Scavenging pumps driven by cranks at the end of the engine
  - Scavenging pumps operated by rocking levers from the crossheads
  - Stepped scavenging pistons
  - Separate electrically driven scavenging blowers
- Four cycle, with
  - Air-injection
  - Airless injection
  - Vertical inlet and exhaust-valves
  - Horizontal valves
- Variations, inclusive of
  - Trunk-piston
  - Crosshead
  - Low compression, with auxiliary combustion-starting system
  - Medium compression, with or without steam-heated cylinders
  - Medium compression with electric or red-hot point starting devices
  - Diesel combustion on one side and steam pressure on the other side of the piston
  - Single-acting
  - Double-acting
  - Opposed piston, and variations of this particular design
  - Use of cylinder liners
  - Non-use of cylinder liners
  - Detachable cylinder-heads
  - Cylinder-heads cast integrally with the cylinders
  - Heavy cast-iron frames
  - Steel columns and no cast-iron frame
  - Both frames and columns of cast iron
  - Cylinders cast separately
  - Cylinders cast in-block
  - V-type cylinders
  - Miscellaneous, some of which can be called Diesel engines only by stretching a technical point

All of these have variations in their reversing mechanisms. Some have fresh or salt-water-cooled pistons, others have oil-cooled pistons and some have no cooling medium other than the atmosphere. Lastly, there are advocates of the Diesel-electric drive; and of the Diesel reduction-gear drive, as opposed to direct drive.

Nearly every engineering company that recently has turned its attention to Diesel-engine construction has seemed to desire to produce an engine that is unlike anything previously invented, rather than to follow established successful practice conservatively, within reason but without unnecessary "plagiarism," or else to adopt a careful combination of the best designs in a businesslike manner. It generally is better to pay a royalty for a known good product, if necessary, than to try indirectly to obtain something that is only partly as successful and as practical commercially.

Is it any wonder that some ship owners acknowledge their uncertainty and turn in desperation to geared turbines and turbine-electric drives, enduring the consequent particular troubles and anti-economies that are attendant upon them? There appears to be an almost innumerable number of new designs steadily coming on the market; therefore, is it not time to call a temporary halt? Ultimately, of course, some designs will be eliminated, as has been true of other previous ones.

In the face of following this zig-zag course, with submerged dangers lurking ahead, to port and to starboard, the Diesel engines that have been adopted more extensively than any other type of engine for large merchant ships have been of simple and straightforward design and follow recognized steam-engine design as nearly as is practicable. On the other hand, a number of engines that differ radically have given excellent results in the limited number of vessels in which they have been installed; so, it is exceedingly difficult to draw a definite line of development, because the different models may give equally good results if well constructed. The illustrations show a number of the leading types of American

[1] Editor of *Motorship*, New York City.

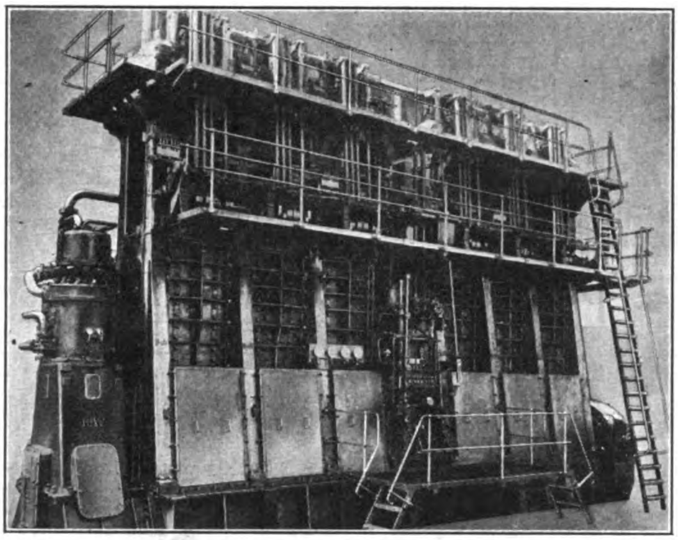


FIG. 1—THE BURMEISTER & WAIN LONG-STROKE DIESEL ENGINE

and European marine Diesel engine, and I will comment briefly upon each design.

Fig. 1 shows the Burmeister & Wain long-stroke design, which is one of the best known. Seven companies, located in Sweden, Norway, Denmark, Scotland, Germany and America build this engine under license, and all have made their constructions with practically no change in the design. This engine is of very robust and straightforward construction without having had any attempt made toward radical departure from known marine practice. I consider that this is responsible for its success, in conjunction with sound engineering. During the 10 years since the first Burmeister & Wain engine was completed, the design has not been without its numerous slight troubles due to minor parts but, although nearly all of these have been eliminated by modification, the present model is almost the same in appearance as the original design. The Cramp shipyard in Philadelphia is building four such 2250-i.hp. engines.

Fig. 2 shows an American version of the same engine, the Worthington, although it is not built under a license. This engine follows the earlier design more closely. It has cylinders that are cast separately, instead of the box type of cylinder cast three per set or in single boxes bolted together. Its main differentiation from the Burmeister & Wain design is that the front frames have their lower parts detachable to facilitate the removal of the crankshaft. There is also an auxiliary exhaust-port in the lower end of each cylinder that can be cut-out if desired.

Another well-known and successful marine Diesel engine is the Werkspoor engine shown in Fig. 3. It is

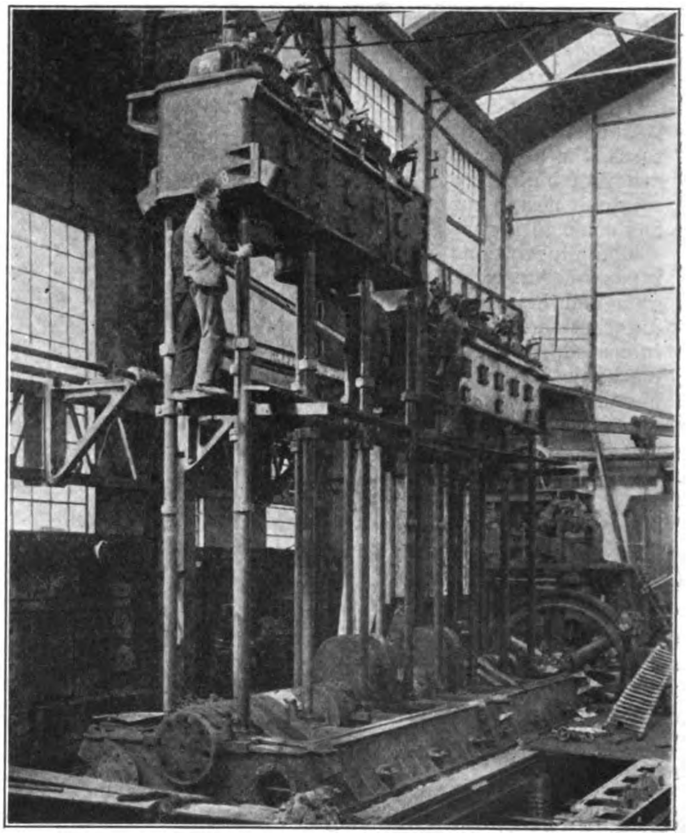


FIG. 3—THE WERKSPOOR MARINE DIESEL ENGINE IN COURSE OF CONSTRUCTION

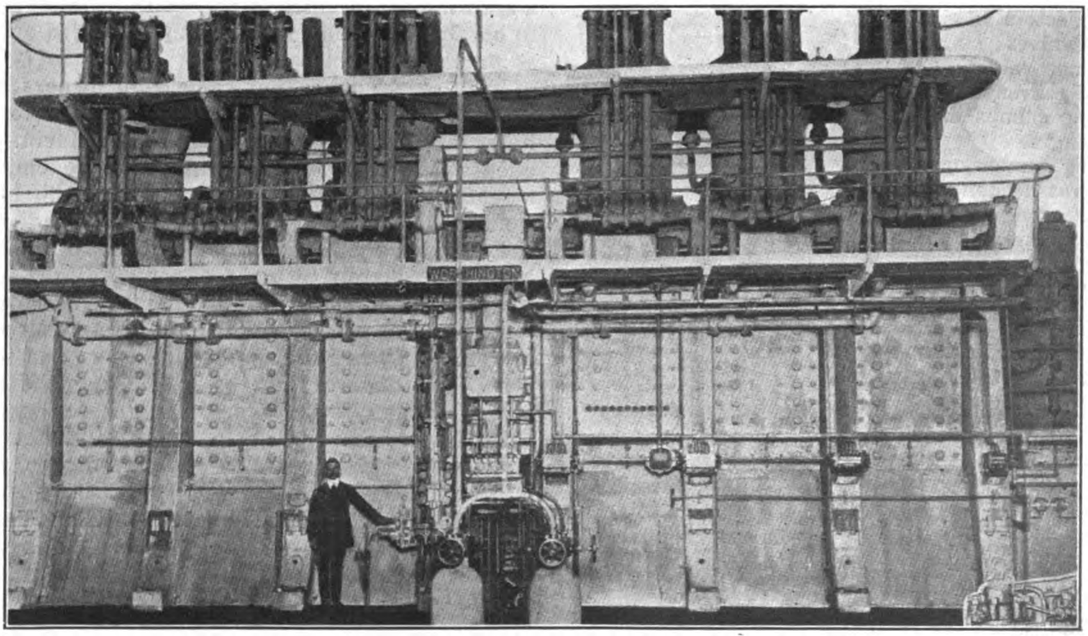


FIG. 2—THE WORTHINGTON, AN AMERICAN TYPE OF DIESEL ENGINE

constructed along lines that are very different from those of the Burmeister & Wain design and yet it follows recognized lines of marine steam-engine design. It, also, is of the four-cylinder single-acting type. Fig. 4 shows that the large cast-iron frames similar to those in the Burmeister & Wain engine are replaced by lighter steel columns, and that steel tierods run from the bedplate to the top of the cylinder box. The cylinders have no jackets in the ordinary sense or detachable heads, but are of mushroom construction and "dropped" into the box, the latter also acting as a lateral girder. The cylinders have detachable extensions for removing the pistons. The cast-iron columns at the back are used for carrying the crosshead guides and air intercoolers. These features of the construction are different from any other designs, although the cylinder-box system has been adopted by the Tosi, Krupp, North British, Holeby, Polar, Burmeister & Wain and several other companies in recent years. I believe it will come into more general practice later on. About 10 companies build the Werkspoor engine. Fig. 5 gives a better idea of the steel-column construction and shows more clearly the cast-iron columns at the back of the Werkspoor engine that are used

Fig. 4—A Completed Werkspoor Engine

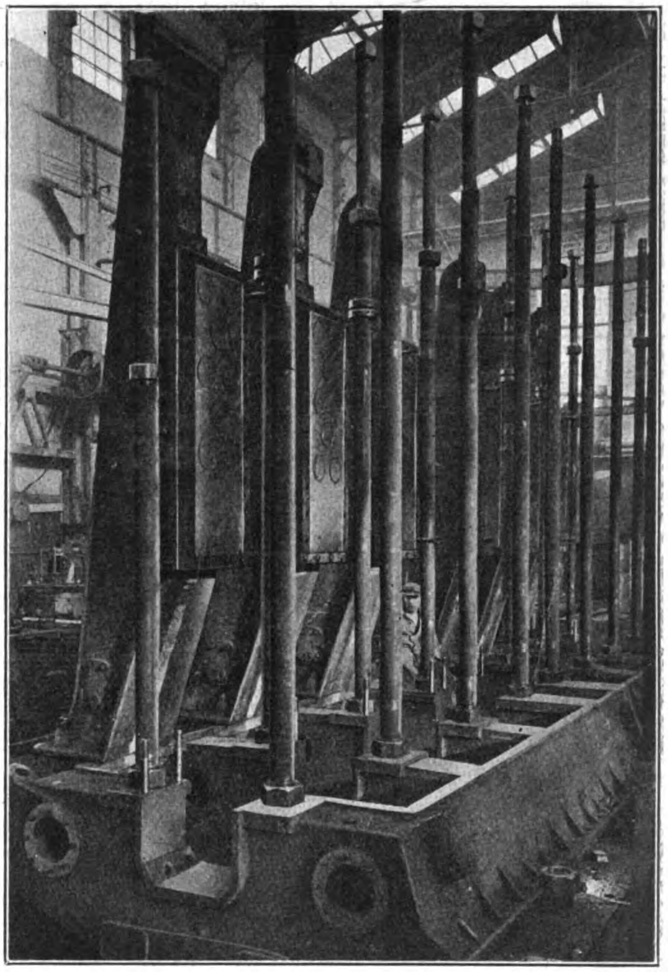


Fig. 5—In the Werkspoor Engine the Heavy Cast-Iron Frames Are Replaced by Lighter Steel Columns and Steel Tierods Extend from the Bedplate to the Top of the Cylinder Box

for carrying the crosshead guides and intercoolers for the air-compressors. The diagonal tierods are not shown.

Departure from Original Design

The question arises in regard to how far a licensee should depart from the original design to suit local conditions of construction and operation. Fig. 6 shows one of the Werkspoor engines constructed for the Standard Oil Co. tanker H. T. Harper by the Pacific Diesel Engine Co., Oakland, Cal. There are three firms building Werkspoor Diesel engines in the United States. All three are working along different lines and all have changed the parent design somewhat. It will be noted that the Werkspoor engine of the Pacific Diesel Engine Co. has retained the steel columns, but the large cast-iron columns at the back and the diagonal tierods are dispensed with. The company has introduced a cast-iron frame in their place, through which the steel columns run and, by cast-iron distance-pieces between the frames and the cylinder box, it has secured the effect of vertical tierods. However, somewhat similar construction was used by the Werkspoor company in the engines of the motorships Vulcanus and Sembilan about 12 years ago, which are still in service; it was also used in some of its stationary and submarine engines.

Fig. 7 illustrates a 2000-i.hp. Werkspoor engine now being built at the Camden plant of the New York Shipbuilding Corporation. It differs from both the Holland-Werkspoor and the Pacific-Werkspoor engines in that

cast-iron A-frames run from the lower sides of the cylinder boxes to the bedplate and instead of the steel columns and diagonal rods, or the combination frame-

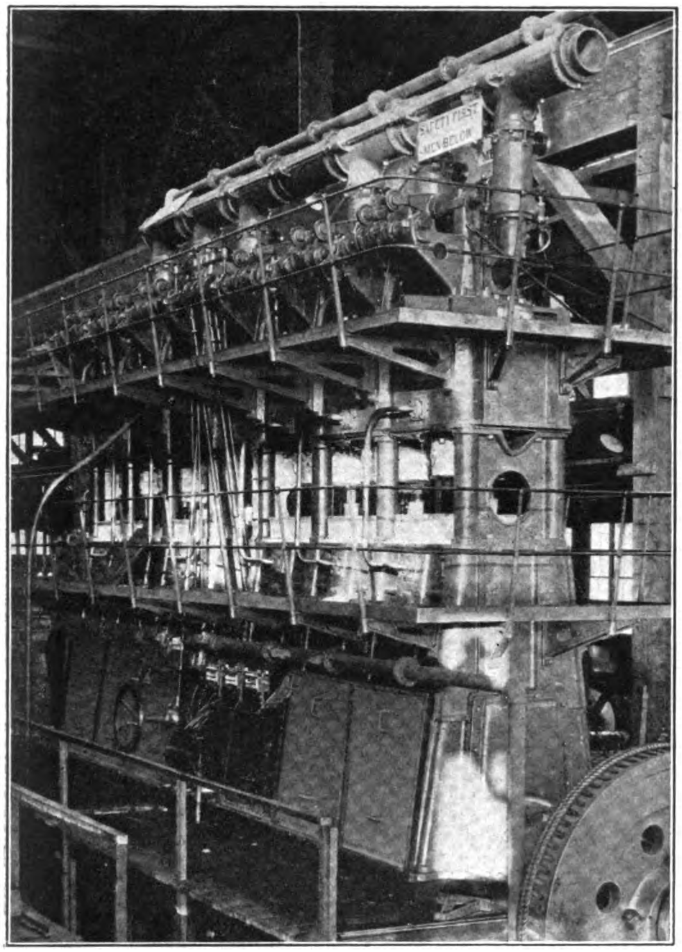


FIG. 6—A WERKSPOOR TYPE OF DIESEL ENGINE CONSTRUCTED ON THE PACIFIC COAST FOR AN OIL TANKSHIP

and-column arrangement. However, the upper part of the engine follows regular Werkspoor design. Another change is that the New York Shipbuilding Corporation

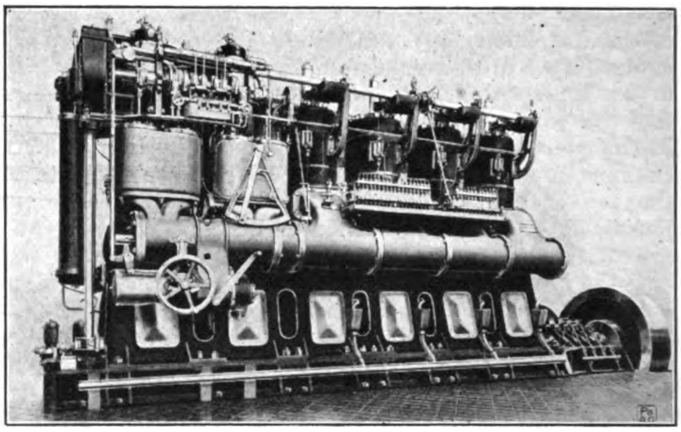


FIG. 8—A 360-B. HP. TRUNK-PISTON ENGINE OF THE TWO-CYCLE PORT SCAVENGING TYPE IN WHICH THE SCAVENGING CYLINDERS ARE ALSO EMPLOYED FOR AIR STARTING

has reverted to the stepped-piston type of air-compressor, instead of using the separate-stage compressor adopted by Werkspoor and others of its licensees.

Another case where licensors and their licensees make a number of distinctive types of engine is illustrated in Fig. 8 which depicts a 360-b.hp. trunk-piston type Polar Diesel engine of the two-cycle port-scavenging type in

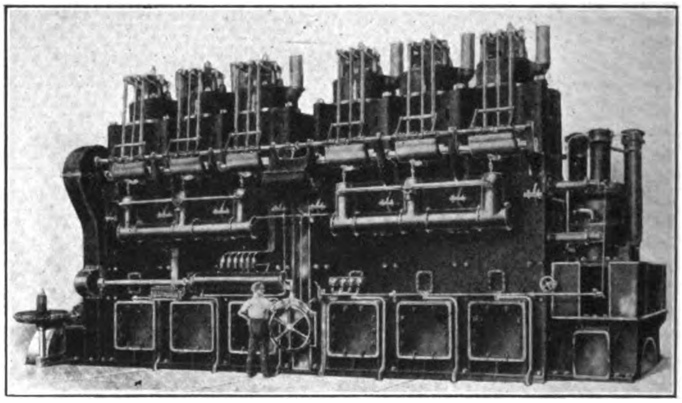


FIG. 9—A 900-B. HP. CROSSHEAD-TYPE ENGINE IN WHICH THE CYLINDERS ARE IN TWO BOX GROUPS AND BOLTED TO BOX-TYPE CAST-IRON FRAMES

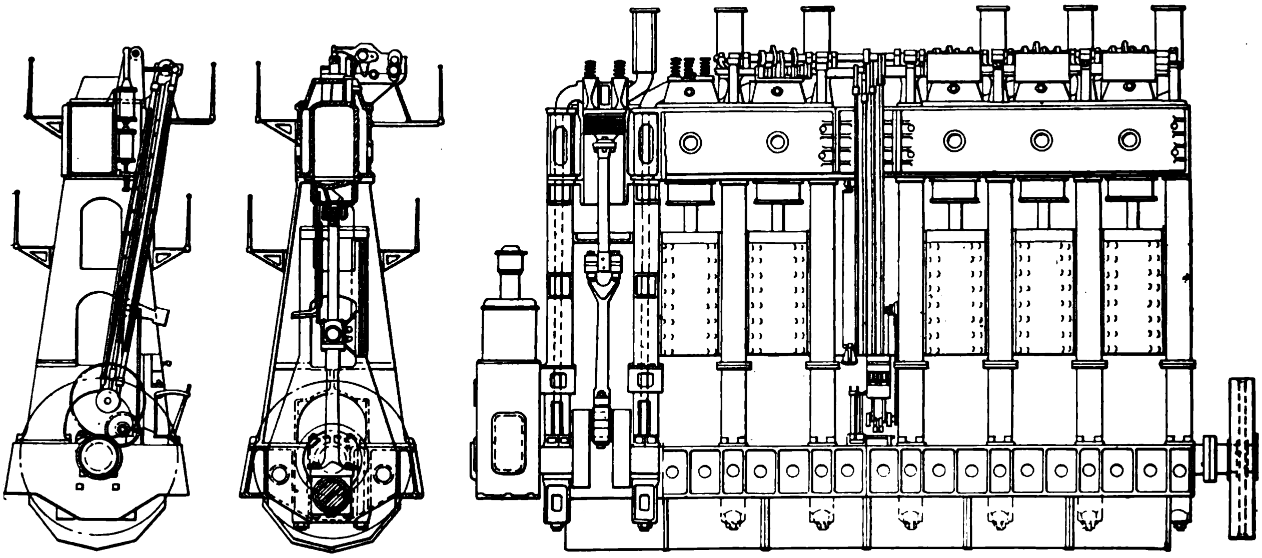


FIG. 7—A 2000-I. HP. WERKSPOOR ENGINE BUILT IN THE UNITED STATES AND HAVING CAST-IRON A-FRAMES EXTENDING FROM THE UNDERSIDE OF THE CYLINDER BOXES TO THE BEDPLATE INSTEAD OF THE STEEL COLUMNS AND DIAGONAL TIERODS OF THE ENGINE SHOWN IN FIG. 4 OR THE COMBINATION FRAME AND COLUMN ARRANGEMENT ILLUSTRATED IN FIG. 6

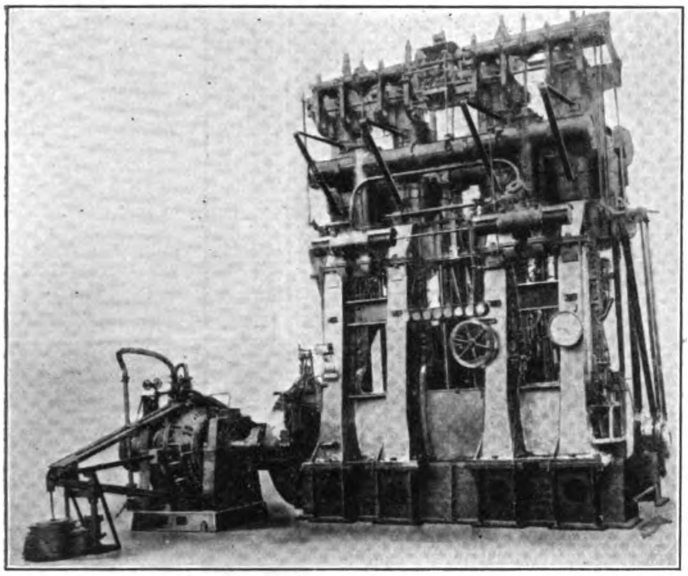


FIG. 10—A 600-B. HP. CROSSHEAD-TYPE ENGINE OF BRITISH CONSTRUCTION IN WHICH THE WORKING CYLINDERS ARE CAST SEPARATELY AND THE FRAMEWORK IS OF THE OPEN A-TYPE

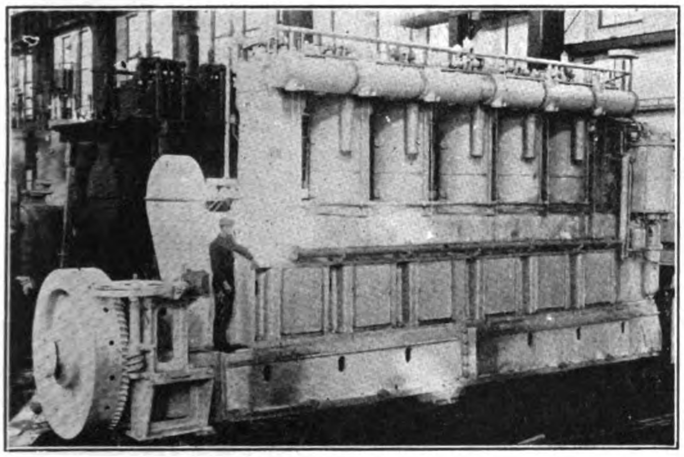


FIG. 11—AN AMERICAN TRUNK-PISTON TYPE 750-B. HP. ENGINE

which the scavenging cylinders are used also as air-starting cylinders. They can be seen at the forward end under the air-compressor stages. Fig. 9 shows a 900-

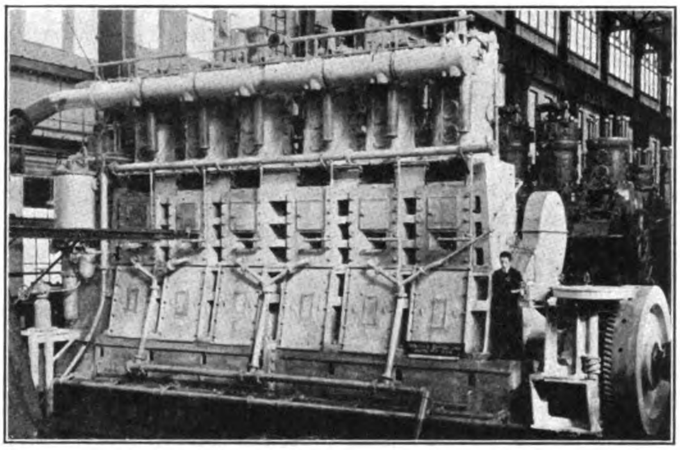


FIG. 12—A 900-B. HP. CROSSHEAD-TYPE AMERICAN ENGINE

b.hp. crosshead-type Polar engine designed for 135 r.p.m. that is built also by the Atlas Diesels Motorer and is of somewhat distinctive design. The cylinders are in two box groups, bolted on to box-type cast-iron frames. The cylinders for air-starting are below the working cylinders, compressed air being admitted to the under sides of the pistons by sliding valves. This is done to avoid the entrance of cold starting-air and its contact with the hot working piston. A somewhat similar ar-

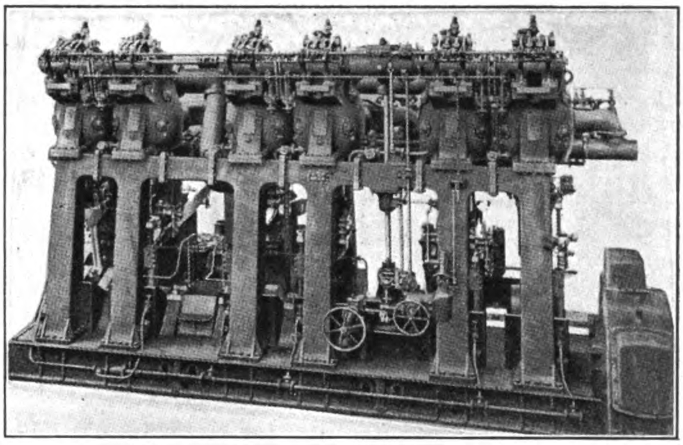


FIG. 13—A BELGIAN TWO-CYCLE CROSSHEAD-TYPE ENGINE EQUIPPED WITH VALVE-IN-HEAD CYLINDERS

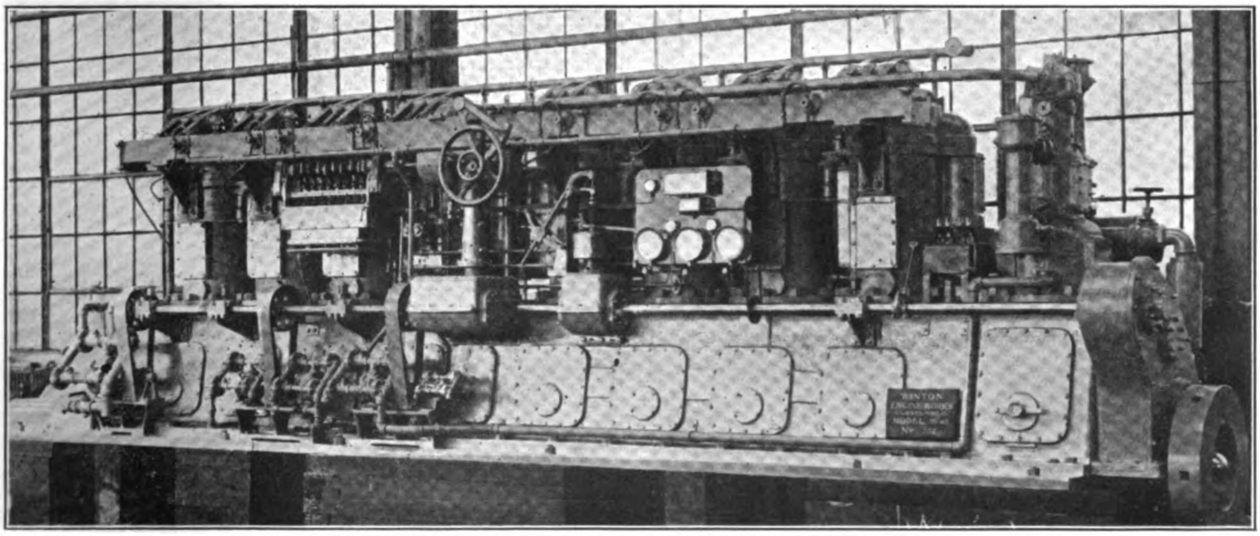


FIG. 14—A 400-B. HP. HIGH-SPEED TRUNK-PISTON FOUR-CYCLE ENGINE THAT IS USED EXTENSIVELY IN CONJUNCTION WITH THE ELECTRIC DRIVE

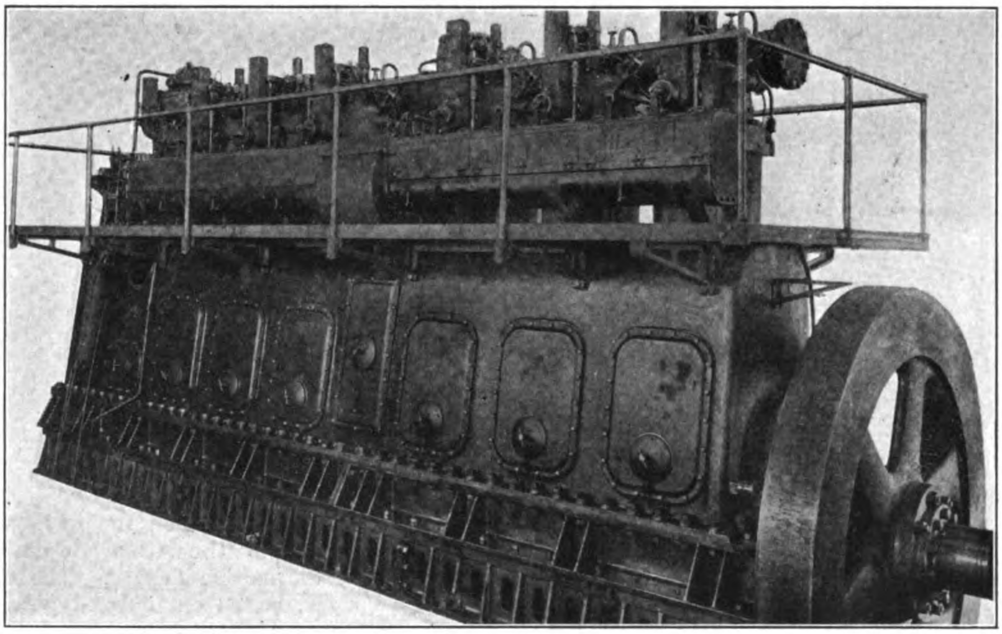


FIG. 15—A 700-B. HP. FOUR-CYCLE TRUNK-PISTON ENGINE HAVING THE INLET AND EXHAUST-VALVES HORIZONTAL INSTEAD OF VERTICAL

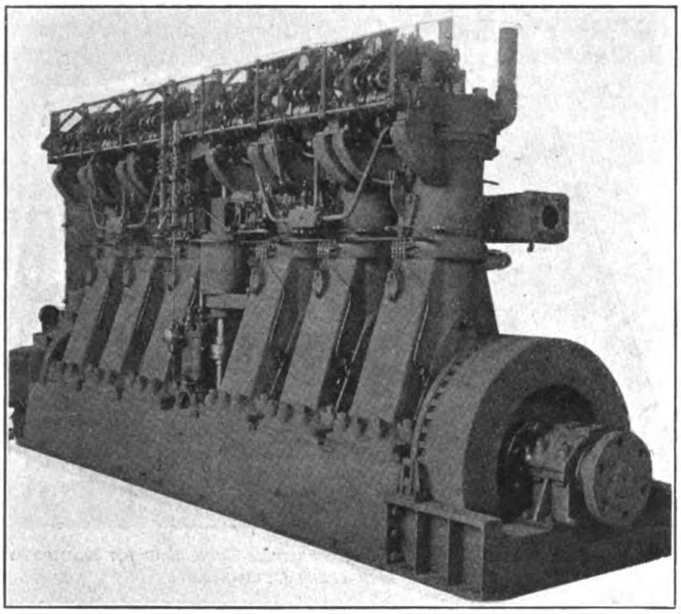


FIG. 16—AN AMERICAN TRUNK-PISTON FOUR-CYCLE ENGINE

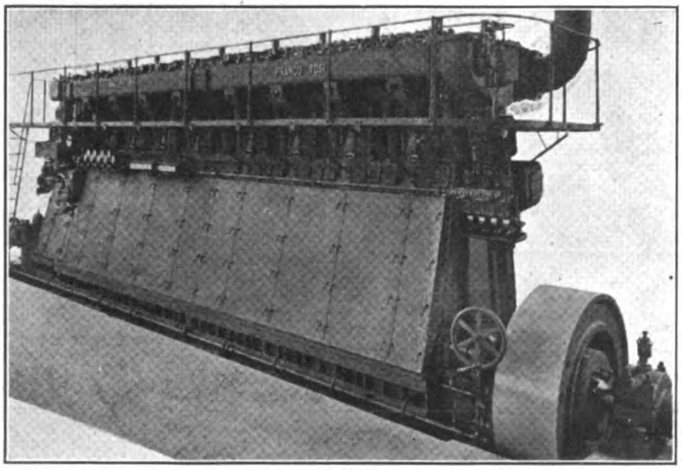


FIG. 17—AN ITALIAN FOUR-CYCLE CROSSHEAD-TYPE ENGINE THAT IS CHARACTERIZED BY SIMPLICITY OF DESIGN

rangement is shown in Fig. 10, which is a Neptune-Polar crosshead-type engine of 600-b.hp. that is built by the British licensee, Swan, Hunter & Wigham Richardson; but the general appearance of the engine is much different, it having working cylinders that are cast separately, and an open A-type framework.

Fig. 11 shows a trunk-piston type 750-b.hp. engine that is built by the American licensees, McIntosh & Seymour. Engines of this design and power have given splendid service in the Vacuum Oil Co.'s tankers Bayonne and Brammell Point. A crosshead-type 900-b. hp. engine is

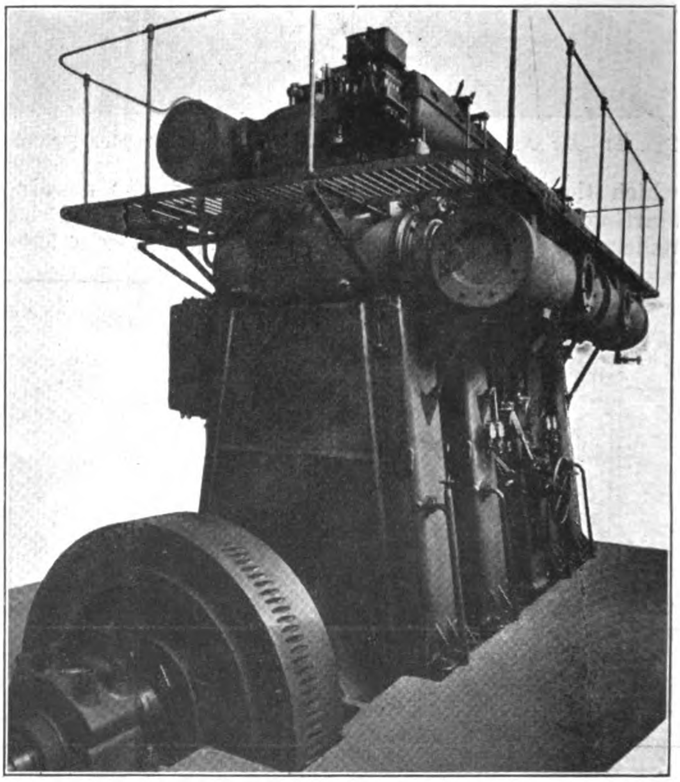


FIG. 18—A CROSSHEAD-TYPE TWO-CYCLE ENGINE OF ITALIAN CONSTRUCTION THAT WAS ABANDONED IN FAVOR OF THE FOUR-CYCLE MODEL ILLUSTRATED IN FIG. 17

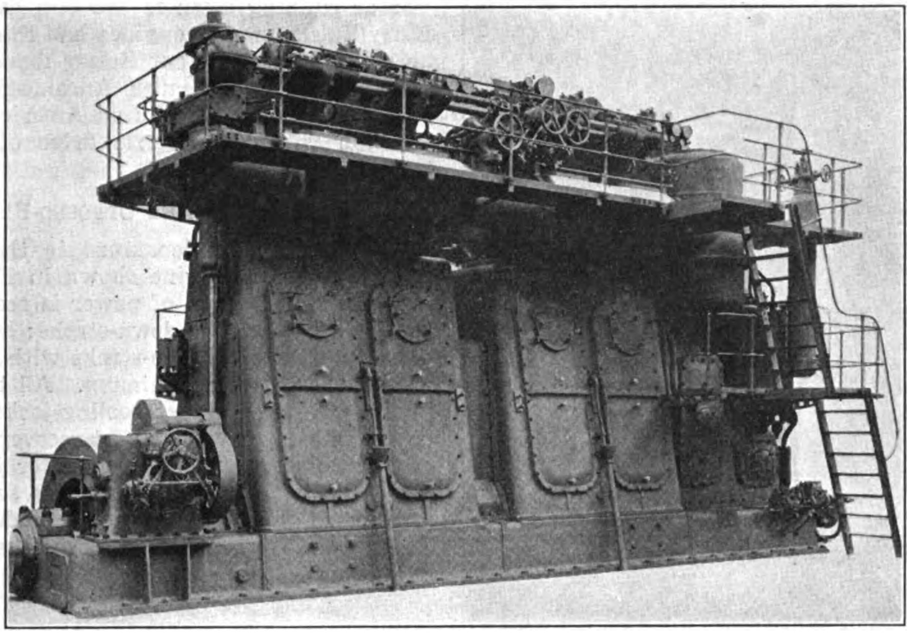


FIG. 19—A 1200-B. HP. PORT-SCAVENGING TYPE OF ENGINE HAVING THE SCAVENGING PUMP AND THE AIR-COMPRESSOR MOUNTED AT THE FORWARD END

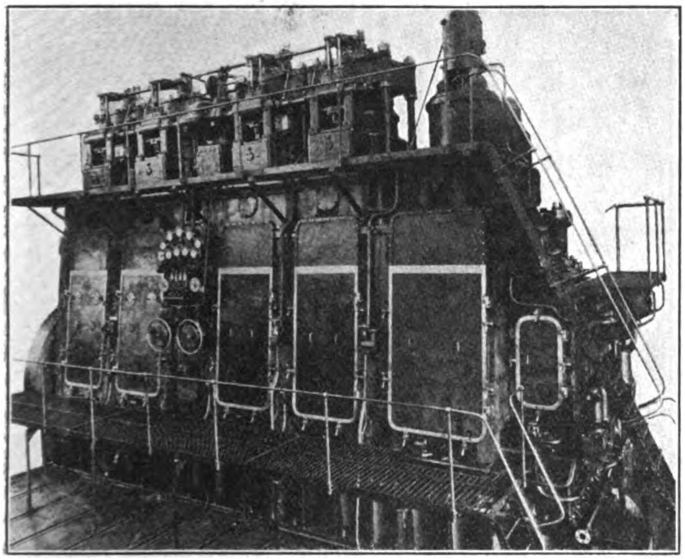


FIG. 20—A PORT-SCAVENGING TWO-CYCLE CROSSHEAD TYPE OF ENGINE IN WHICH THE CYLINDER-HEADS ARE BOLTED DOWN BY LONG TIERODS RUNNING DOWN TO THE BEDPLATE

illustrated in Fig. 12. This is installed in the American motorship Kennecott.

Fig. 13 shows a two-cycle crosshead-type engine of the valve-in-head scavenging type, having open cast-iron frames. It was built by Carel Fréres, of Belgium, and is constructed under license in America by the Nordberg Mfg. Co. of Milwaukee. The Schneider engine, which is constructed in France, is built along very similar lines. A much different type is illustrated in Fig. 14. This is the 400-b.hp. high-speed trunk-piston four-cycle Winton engine. It is built more along the lines of large marine gasoline engines than those of steam-engine practice and is used extensively in conjunction with electric drive, as well as with direct drive in vessels of moderate size. The United States War Department has installed 14 such engines in twin-screw concrete passenger-ships with great success. Another four-cycle trunk-piston engine, but one of lower speed, is the 700-b.hp. Nelseco engine shown in Fig. 15. This engine is original in that it has horizontal inlet and exhaust-valves instead of having the customary vertical valves.

A third type of American-built trunk-piston four-cycle marine Diesel engine is shown in Fig. 16. This is the Dow engine constructed under British license. There is only one American ship equipped with these engines and, although a wooden vessel, she has given splendid service.

For a simple design of the four-cycle crosshead-type engine, the Tosi Diesel engine shown in Fig. 17 has much to commend it. The Tosi firm also built a neat crosshead-type two-cycle Diesel engine such as is shown in Fig. 18, but this was not so successful and it was abandoned for the four-cycle model.

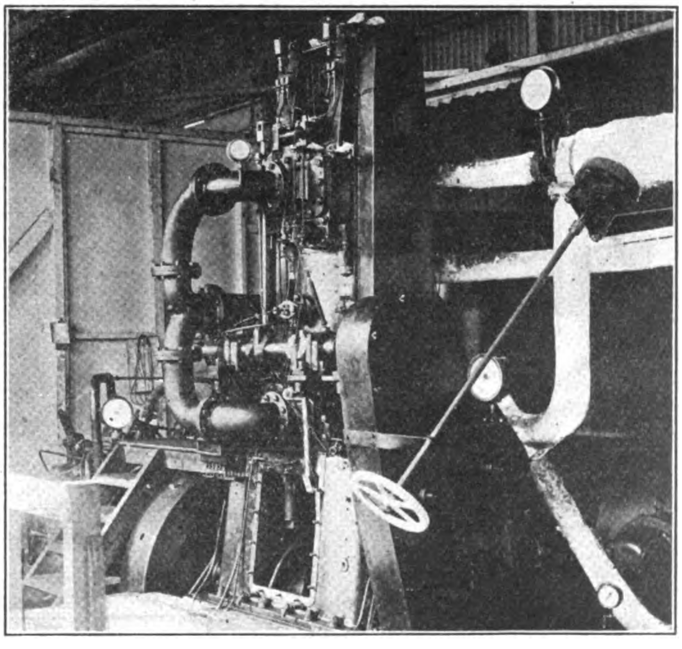


FIG. 21—A COMBINATION DIESEL AND STEAM ENGINE OF BRITISH CONSTRUCTION

In This Engine the Combustion of Oil Provides the Main Supply of Power on the Down-Stroke, While Steam Power Is Used on the Up-Stroke To Reduce the Heat Losses to a Minimum

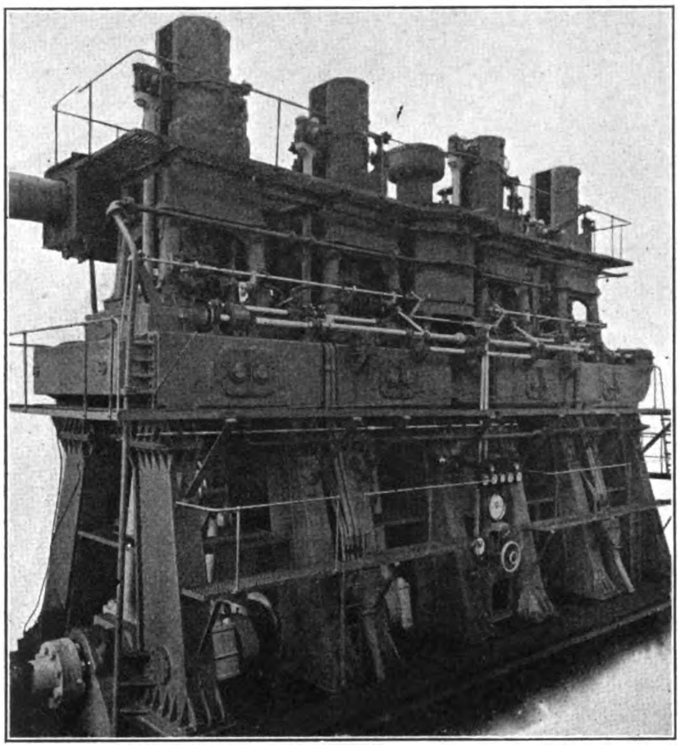


Fig. 22—A Four-Cylinder Crosshead-Type Opposed-Piston Two-Cycle 3000-I. Hp. Engine in Which the Pistons in Each Cylinder Move Apart and Combustion Takes Place between Them

Of all two-cycle crosshead-type Diesel engines, probably the simplest in design is the Sulzer engine, shown in Fig. 19, also built by the Busch-Sulzer Co., St. Louis. The model illustrated is a 1200-b.hp. engine of the port-scavenging class, and it is an exceptionally well-balanced job. The scavenging pump and air-compressor are at the forward end, but the scavenging pump is replaced by an electrically driven blower in new constructions. The Ansaldo-San Giorgio engine is of somewhat similar construction, yet different; it also is of the port-scavenging two-cycle crosshead type and is shown in Fig. 20. The method of bolting down the cylinder-heads by using long

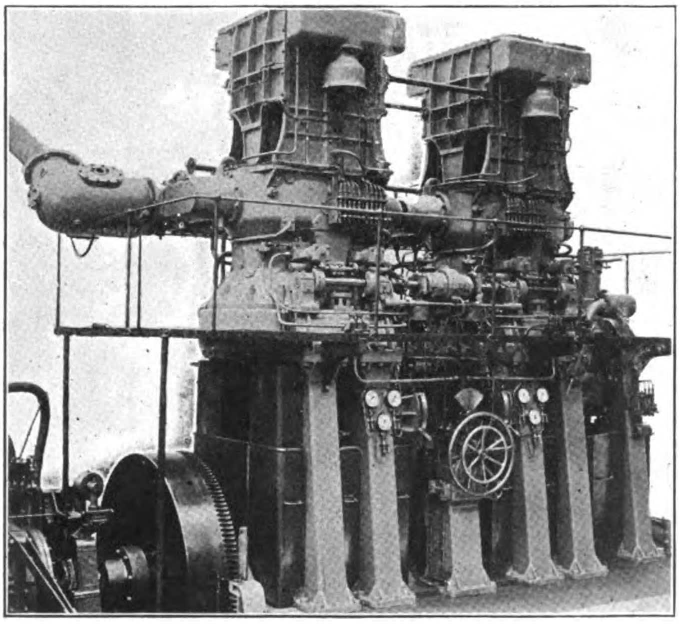


Fig. 23—Another Design of Opposed-Piston Engine

tierods running down to the bedplate should be noticed. Early Sulzer marine engines had this feature, but it was abandoned; the present Sulzer marine engines have no steel rods. Two smaller Ansaldo-San Giorgio engines were recently installed in an American freighter in conjunction with the electric drive of the General Electric Co.

Combination and Opposed-Piston Engines

Another radical departure is the Still combination Diesel-and-steam engine shown in Fig. 21. In this design the main source of power is secured from the combustion of oil on the down-stroke; supplementary steam power is used on the up-stroke with the object of reducing heat losses to a minimum. All the heat usually lost in the working-cylinder cooling-jackets is used for generating steam, as well as heat recovered from the exhaust gases by generators and feed-water jackets. Compounding in a single-cylinder engine is secured by using the lower ends of two combustion cylinders as high-pressure steam-cylinders and those of the other four working cylinders as low-pressure steam-cylinders. Overload is secured by oil-firing the steam generator. Oil-firing is used also for starting on steam. This furnishes initial heat to the Diesel cylinders and so enables the cylinder pressure to be kept down to 300 lb. per sq. in. as against 400 to 500 lb. per sq. in. for the straight Diesel engine. The single-cylinder unit illustrated develops 350 b.hp. at 350 r.p.m., or 540 b.hp. on overload.

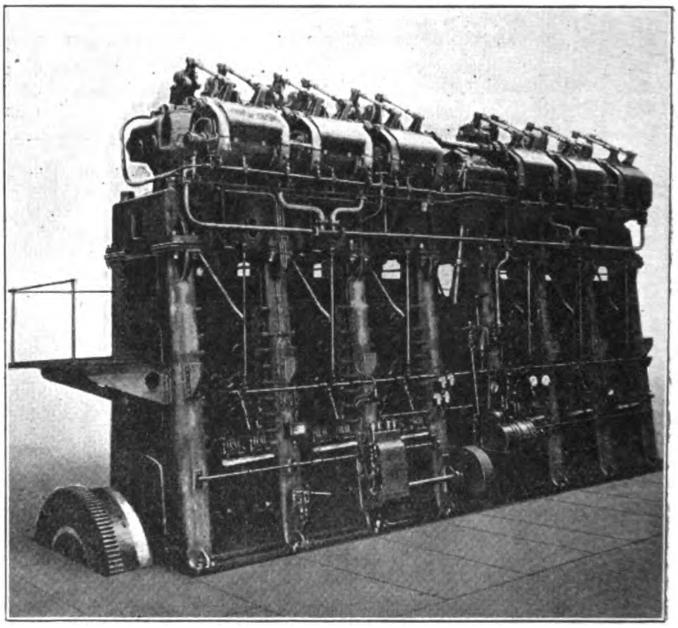


Fig. 24—A Four-Cycle Airless Injection Engine Developing 1250 B. Hp. at 115 R.P.M.

A large shipyard near Philadelphia is deeply interested in the opposed-piston type of Diesel engine. A Doxford four-cylinder crosshead-type opposed-piston two-cycle engine of 3000 i.hp. is illustrated in Fig. 22. The pistons in each cylinder move apart and combustion takes place between them on the Ochelhauser principle. The second vessel equipped with the Doxford engine was placed in service recently. Excellent results were obtained at the tests of both ships. Airless injection of fuel at high pressure has been adopted in conjunction with low cylinder-compression and, because of the latter, steam or hot water is used for warming the cylinder-jackets prior to starting from cold.

A different arrangement of the opposed-piston design is depicted in Fig. 23. This is the Cammell-Laird Fullagar marine Diesel engine, which, regardless of its radical departure from conventional design, is likely to come to the front, even if only because of the amount of engineering skill that is back of it. The firms having building licenses in England are John Brown & Co., Clydebank; Palmers Shipbuilding & Iron Co., Newcastle; Cammell-Laird & Co., Birkenhead; English Electric Co., London; Dunsmuir & Jackson, and David Rowan & Co., Glasgow. The Bretagne Shipbuilding Co., Nantes, holds a license in France. However, American firms do not as yet seem to have so much faith, although American engineers who have seen this engine have been very enthusiastic. It is an adaptation of the Ochelhauser system combined with bold defiance of the best engineering practice, which is to transmit force in a straight line so far as this is feasible. The upper piston in the Ochelhauser system is connected to a crosshead that carries two connecting-rods which pass to the sides of the cylinder to two crankpins outside and opposite to a central crankpin, to which the lower piston is connected by its rod, the three crankpins with their webs forming a single crank unit between a pair of bearings. In the Cammell-Laird Fullagar design, only a single crank is required per cylinder and two cylinders are required to form a complete unit. Instead of being an overhung arm placed over its crank in a fore-and-aft line, the crosshead of the upper piston is placed centrally in regard to its piston but in an athwartship line; and, instead of having a connecting-rod from each end directly down to its crank, there is a diagonal rod from each end of the crosshead to keep the strains central. These diagonal rods lead down to a crosshead from which a connecting-rod is attached to a second crank, the guide shoes being arranged to take the thrust of both the connecting and the diagonal rods. Thus, the upper piston of one cylinder is connected to the lower piston of the adjoining cylinder; and the lower piston is connected to the upper piston of the same adjoining cylinder. Another radical change in design exhibited by this engine is that the scavenging pistons and cylinders are of square section.

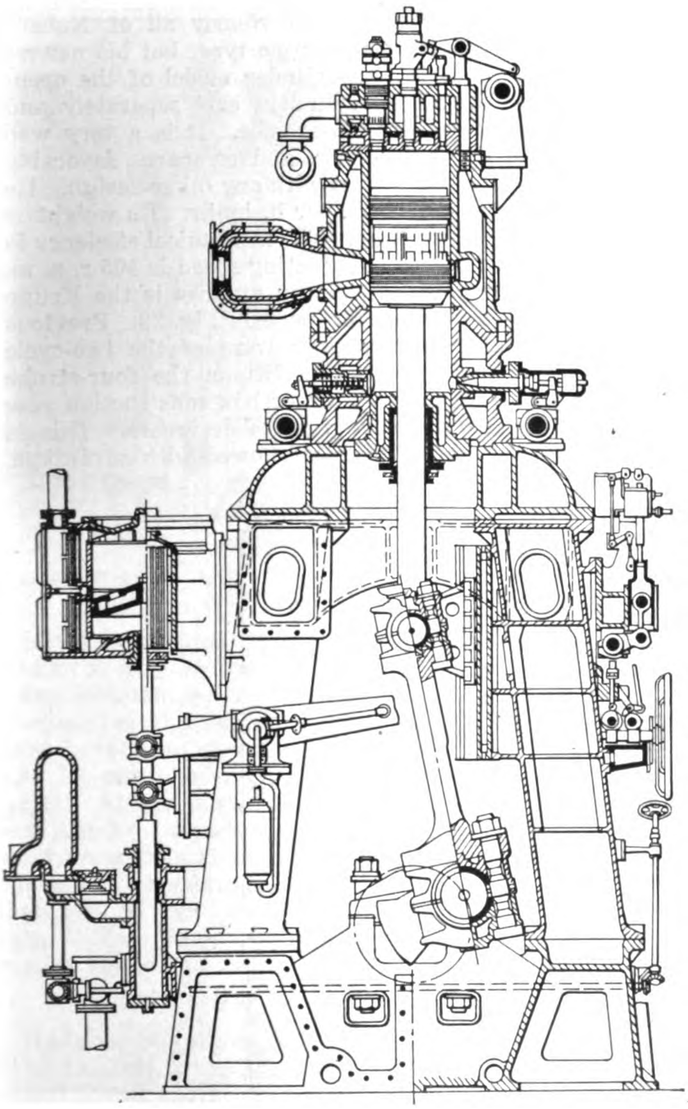


Fig. 25—A Three-Cylinder Double-Acting Two-Cycle 850-B. Hp. Engine

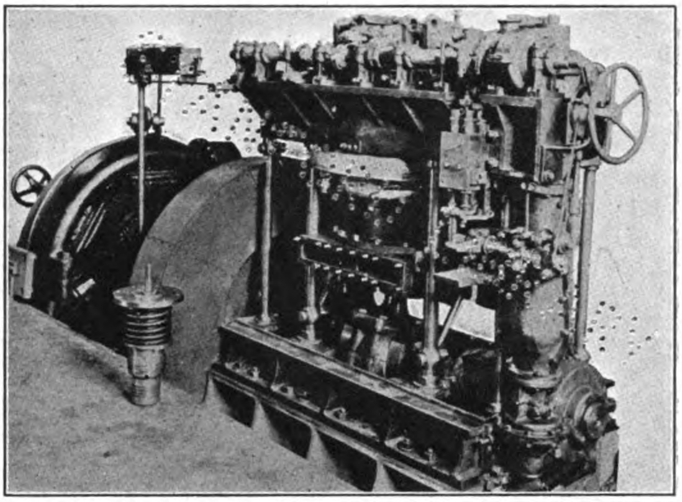


Fig. 26—An American Heavy-Duty Compound Engine Designed To Operate on Bunker Oil

Mechanical Injection and Double-Acting Diesels

To return to a marine engine of more conventional design, the Vickers engine shown in Fig. 24 is noteworthy for being the first high-pressure engine to have the airless injection of fuel that is known as solid or mechanical injection, in which the fuel is sprayed by a pump at a pressure of about 4000 lb. per sq. in. The engine shown is of the four-cycle type and develops 1250 b. hp. at 115 r. p. m. The cylinders are cast separately; they have deep bases bolted together that form an entablature which is mounted on A-type cast-iron frames that straddle the main bearings. This company also is developing a double-acting Diesel-engine, as well as a new type of opposed-piston engine, but its details are not available.

Fig. 25 is an illustration of a section of the three-cylinder Blohm & Voss 850-b. hp. double-acting two-cycle Diesel engine of the motorship Assyrian, formerly named the Fritz. The cylinder bore is 17 23/32 in., the stroke is 27 31/32 in. and the engine speed is 120 r. p. m. The stuffing-box is very deep and contains 11 sections of metallic packing; a grooved ring divides them to facilitate the distribution of lubricating oil. Each section consists of two rings split in halves that fit closely around the rod without springs, but they are kept close to the rod by a spring clip ring that is carried in a retaining ring clear of the rod. These 11 sections are entered into the stuffing-box and held in position by the gland ring. This feature was very successful and was designed in its final

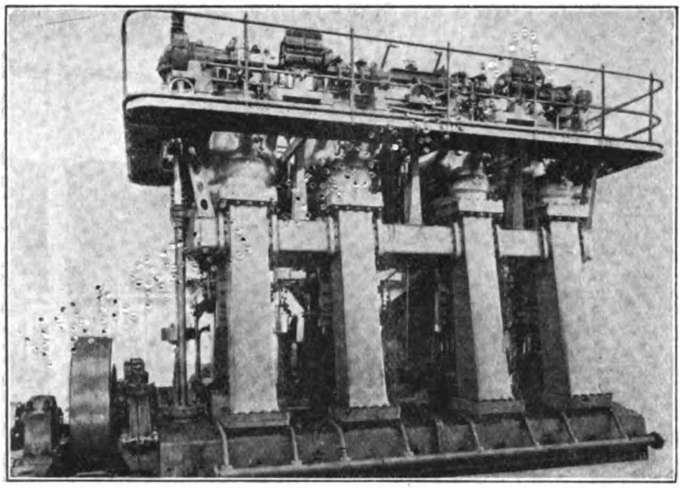


Fig. 27—A Swedish 1600-Hp. Two-Cycle Engine

form only after months of experimental work. A prominent German Diesel engineer informed me recently that the engine itself never would be built again in its present form. The engine works without apparent mechanical difficulties, but it leaves much to be desired as to its supremacy over other systems. Its construction is of exceptionally high quality.

Compound Engines

The Sperry compound oil engine shown in Fig. 26, differs materially from all other designs I have illustrated. The sturdy construction necessary for an engine of its speed is indicated by the size of the crankshaft, which is considerably greater in diameter than that of any other combustion engine of which I have any knowledge and approaches the bore of the combustion cylinders themselves. The large clearance dome, which forms the combustion-chamber of the compound, stands out in marked contrast to standard Diesel practice. This dome is large and forms an upward extension of the combustion cylinder, extending also to the right in a large sweep and surrounding the transfer valve that seals the transfer port. The sleeve-like induction valve is seated on top of the transfer valve and is controlled by the cam-operated fork. The first-stage annular compression-pump, surrounding the trunk-piston below the low-pressure piston proper, delivers its air to a small receiver which, in turn, discharges to the cored port surrounding the induction sleeve. The small balancing cylinder maintains a permanent connection with the low-pressure cylinder. The solid-injection fuel-valve and nozzle are placed approximately over the center of gravity of the large masses of air in the clearance dome.

Latest European Practice

Fig. 27 represents a 1600 shaft-hp. two-cycle engine that was built recently by Louis Nobel in his new factory near Stockholm, Sweden. Nobel has built hundreds of Diesel engines in 72 different designs for large tugs, tankers, gunboats and submarines on the rivers and inland seas in Russia. His first engines were installed in the tanker Sarmat in 1904 and are still in operation. The Polar engine company constructed a pair of Diesel engines from Nobel's designs for the tanker Wandal in 1903. The Sarmat's engines each developed 180 b. hp. at 240 r. p. m. and were installed in conjunction with an electric drive. With the exception of a French barge, these were the first heavy-oil-engined vessels to be built; hence Nobel has had a longer Diesel-engine experience than any other designer. Nearly all of Nobel's engines have been of the four-cycle type, but his newest creation is a two-cycle four-cylinder model of the open-crankpit type, having its cylinders cast separately and carried on heavy cast-iron A-frames. It is a very well designed piece of engineering, and compares favorably in regard to space and weight with any other design. Its fuel-consumption is 0.395 lb. per b. hp-hr. Its weight is 170 tons or 236 lb. per b. hp. The mechanical efficiency is 81 per cent and the normal operating speed is 105 r. p. m.

Another of the latest European engines is the Krupp 1400 shaft-hp. marine Diesel shown in Fig. 28. Previous Krupp merchant-marine engines were of the two-cycle type, but this new engine operates on the four-stroke cycle. It has the cylinder-beam or box construction now becoming so popular among marine designers. This is a practice that probably can be followed with advantage,

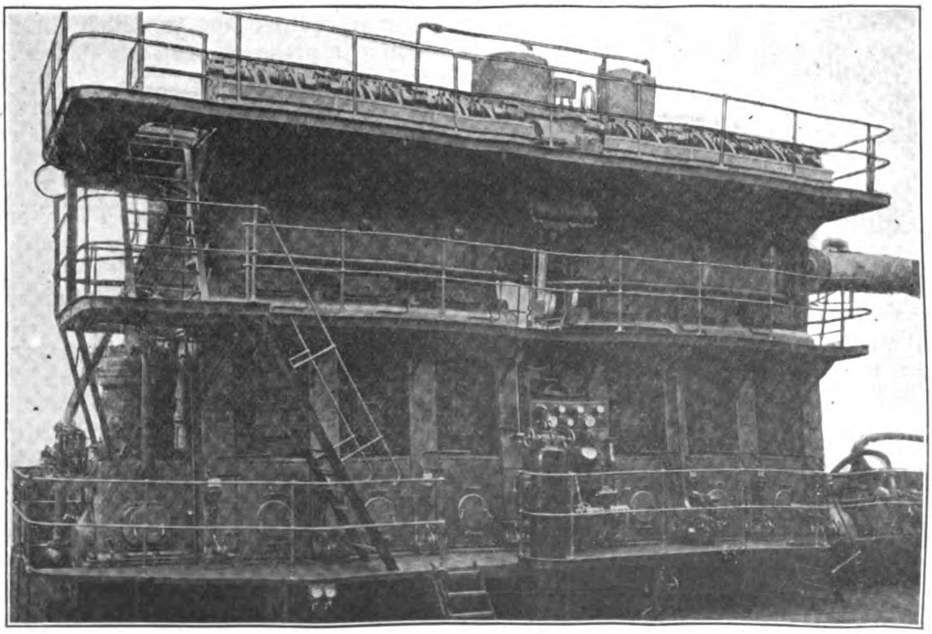


Fig. 28—A German 1400-Hp. Four-Cycle Engine

depending upon individual constructional facilities. However, it is not so suitable with two-cycle engines, because of the exhaust ports. The cylinder box in the Krupp design is carried on cast-iron columns of substantial section. The engine has many splendid engineering features.

I have endeavored to give an idea of the great variation of design that exists among the prominent Diesel engines now on the market. There are a number of others, among which are the Bethlehem-West, Craig, North British, Schneider, Ingersoll-Rand-Price, the new Worthington airless-injection engine, Sabathé, Normand, Goldberg, Steinbecker, Guldner, Atlas-Imperial, Western, Augsburg, Nurnberg, Holeby, Hocke-Fenchelle, Fletcher-Barnard, Allison, Frichs, Speedway, Washington, Knudson, Mianus-Leissner, Dodge-Hvid, Weyland, Renault, Brons-Hvid, Körting, Straus, Lübeck, Kind, Gardner, Wolverine, Kolomna and Wigelius Diesel-engines and the interesting Sumner surface-ignition type crosshead engine. All of these 37 types have their own individual outstanding features, but the 20 odd engines that I have illustrated afford a fair representation of the vagaries of modern marine-engineering progress.

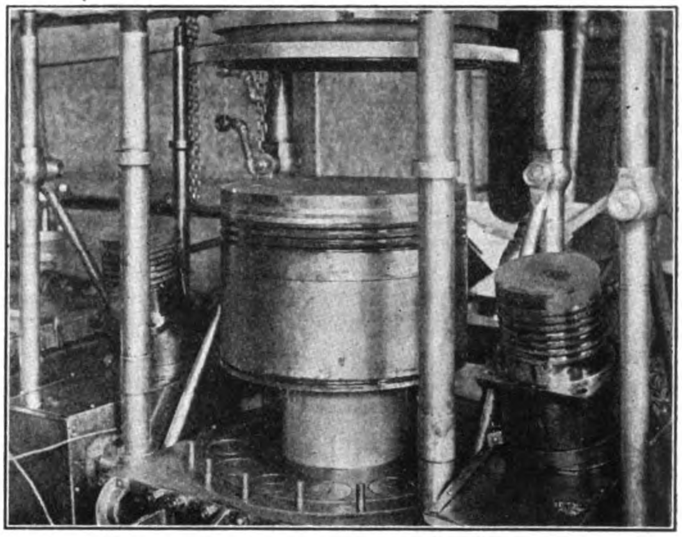


Fig. 30—View of the Engine Illustrated in Fig. 26 Dismantled, Showing the Two High-Pressure Pistons, the Low-Pressure Piston and the Trunk Extension

Diesel-Engine Fuel-Consumption

I wish to correct a misunderstanding that seems to exist regarding the fuel-consumption of modern Diesel-

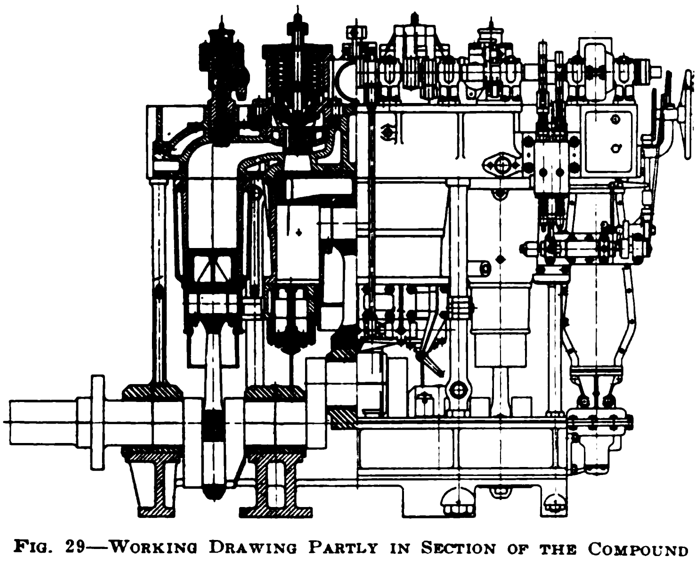


Fig. 29—Working Drawing Partly in Section of the Compound Engine Shown in Fig. 26

engined motorships. The consumption is *not* 0.50 lb. per hp-hr.; it is much less. I have been aboard many ocean-going motorships; each time I have made a special point of studying the engine-room log-book. In no instance have I ever seen recorded a fuel-consumption higher than 0.33 lb. per i.hp-hr., which was inclusive of all auxiliaries. At the most, this would be equivalent to 0.42 lb. per b.hp-hr. Generally, the fuel-consumption is somewhat lower than this, and it becomes lower after a year's operation. Sometimes, the fuel consumption is lower than 0.30 lb. per i.hp-hr. The Shipping Board's motorship William Penn averaged 0.29 lb. per i.hp-hr. on her voyage round the world.

THE DISCUSSION

Harman Scharnagel:—During this discussion it will be of interest to explain the operation of the Sperry compound Diesel engine. The compound principle as applied in this engine is an attempt to produce a light and compact internal-combustion engine using a wide range of fuels with ignition by the heat of the compression. The arrangement of the engine consists of two high-pressure four-cycle cylinders and a simple low-pressure cylinder. The high-pressure pistons are of plain trunk type. The low-pressure piston has an extension of smaller diameter

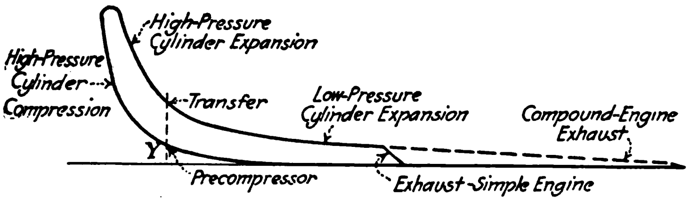


Fig. 31—Typical Indicator-Card

than the main piston. The annular space between this extension and the main piston serves as an air pump.

With reference to Fig. 28, the operation of the engine is as follows: The pump compresses air from atmospheric to a moderate pressure into a small receiver. On the down or inlet stroke of the high-pressure pistons, air under pressure from the receiver passes through the inlet-valve sleeve cooling the latter, until the pistons are at the end of the stroke. The air is then compressed on the up-stroke to about 500 lb. per sq. in. when fuel is injected. The resulting combustion and expansion of

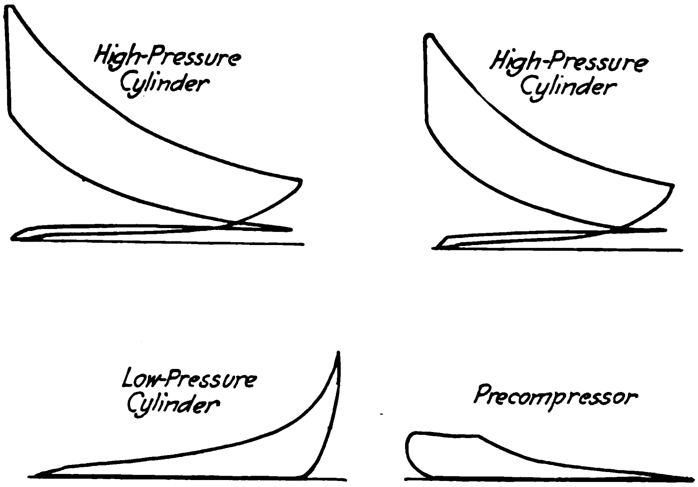


Fig. 32—Indicator-Cards Taken from Different Cylinders of the Engine

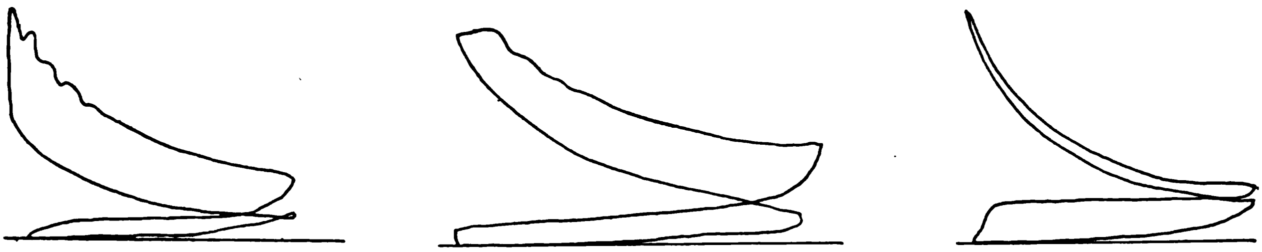


FIG. 33—CARDS OBTAINED UNDER VARIOUS OPERATING CONDITIONS
At the Left Is the Point-Top Card Obtained in the Regular Running of the Engine. By Adjusting the Fuel Injection System the Flat-Top Card in the Center Was Secured, and at the Right Is a Reproduction of a Card Taken from the High-Pressure Cylinder without Fuel

the gases drive down the high-pressure piston to the end of its stroke when the low-pressure piston, which is on the beginning of its working stroke, receives the gases from the high-pressure cylinder through the transfer port that has been opened by lifting the transfer valve from its seat into a water-jacketed cavity so that only its lower surface is washed by the passing gases. In the

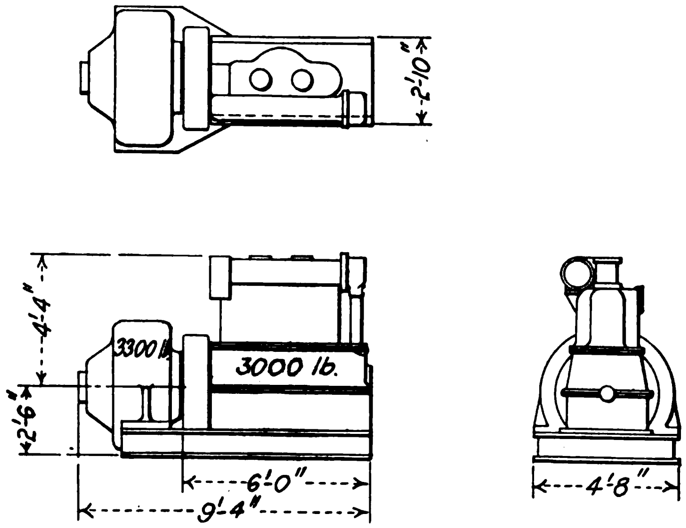


FIG. 34—THE ENGINE COUPLED TO A 50-KW. GENERATOR

low-pressure cylinder practically complete expansion of the working fluid takes place until the opening of the final exhaust near the bottom low-pressure center. Simultaneously with the working stroke of the low-pressure piston the pump space on its under side has compressed

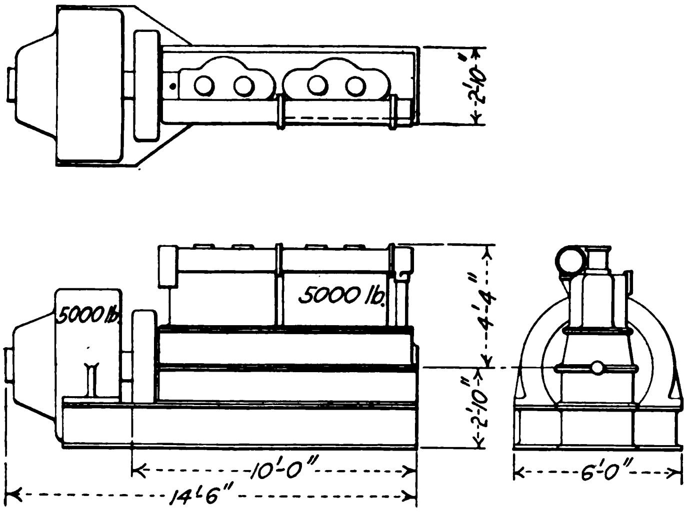


FIG. 35—MOUNTING TWO OF THE ENGINES ON A COMMON BEDPLATE GIVES A 100-KW. GENERATING SET

air into the receiver and the cycle is again carried out in the manner described. To prevent any serious drop in pressure between the high and low-pressure cylinders when the transfer takes place, the exhaust-valve is closed somewhat before the low-pressure top center and the gases are cushioned to a pressure equal to that being transferred from the high-pressure cylinder. The cranks of the two high-pressure cylinders are set together and 180 deg. from the low-pressure crank. The high-pressure cylinders fire and transfer alternately into the low-pressure cylinder so that every down stroke of that cylinder is a working stroke.

Fig. 30 is a view of the engine dismantled showing the high-pressure pistons, and the low-pressure piston with its trunk extension. The row of valves shown in the foreground are the pump inlet-valves. The delivery valves are on the opposite side in back of the piston.

The arrangement of the mechanical parts to carry out this compound cycle are such that many decided advantages are gained over the simple engine of the Diesel type. In the first place the high-pressure part of the cycle is carried out in small high-pressure cylinders which results in a great saving in weight and freedom from excessive metal heating and piston trouble. The low-pressure part of the cycle is likewise carried out in a cylinder designed for its work with advantages similar to those of the high-pressure cylinder. The same low-pressure cylinder and piston serving also as an air pump results in a twofold utilization of the same metal which means a decided saving in weight.

Fig. 31 is a diagram of the indicator-cards showing the functions of the engine as described above. The upper card shows how the regular Diesel card is divided up into the compound card. The portion of the card to the left of the vertical line is the part of the cycle carried out in the high-pressure cylinder, while that to the right is that of the pump and the low-pressure cylinder. The dotted line shows how complete expansion of the gases is obtained in the low-pressure cylinder. Analysis will show that the engine is both supercharging in the high-pressure cylinder and super-expanding in the low-pressure cylinder, thus making for economy and lightness. The lower cards show what the diagrams taken from the different cylinders look like.

The diagrams in Fig. 32 are actual cards taken from the engine. As stated before the high-pressure cylinders start on the compression stroke under a predetermined pressure from the receiver so that a much larger combustion space is needed for the volume of air after compression. This makes possible a simple means of direct fuel injection, without the use of a multi-stage air-compressor, with the resulting saving in weight and parts.

The view at the left of Fig. 33 shows the point-top card obtained in regular running of the engine due to the above combination. In the center is a flat-top card obtained by the same injection system with different adjust-

ment. At the right is a card taken from a high-pressure cylinder without fuel. The horizontal line above atmosphere shows the effect of supercharging on intake stroke with compressed air from the pump receiver.

The advantages of the compound engine are a saving in weight and floor space of between one-half and one-third in the former and more than one-half in the latter, as compared with the simple engine. The number of parts are fewer and simpler and therefore free from excessive manufacturing costs. Greater economy results on account of the complete use of the expansive force of the gases.

Fig. 34 gives an idea of what the compound engine looks like when worked up into our standard 50-kw. generating set. It is a very compact unit, its weight being about one-third that of a simple Diesel set of the same rating. By mounting two of the 50-kw. engine units on a common bedplate, we have the standard 100-kw. generating set as illustrated in Fig. 35.

Fig. 36 will interest the marine engine builders who prefer to use a reduction-gear. There are two double-unit engines connected to the reduction-gear by electromagnetic clutches giving a flexible smooth drive to the gears. This unit is laid out for 1800 shaft hp., the engine running at 250 r.p.m. with a tail shaft speed of about 90 r.p.m.

Fig. 37 shows the comparison between the compound engine and a simple Diesel engine for direct drive with the same piston speed and revolutions.

The engine shown averages about 0.5 lb. per b. hp-hr.

J. D. GILL:—What is a Diesel engine?

T. O. LISLE:—There are two essential elements of characterization in the true Diesel cycle that are not found in any other class of engine. There is also a third essential feature, but it is not so rigidly confirmed in its application. The three fundamental features are

(1) Compression of pure atmosphere to a degree such that the temperature produced is adequate for the inflammation and combustion of the fuel
(2) Injection of fuel at such a rate that the burning proceeds without a rise of pressure in the combustion space. This condition is not realized with absolute precision, there being always a slight rise of pressure when the fuel begins to burn
(3) Injection of fuel by air-blast that produces the turbulence needed for good combustion. This is essential, but it is not distinctive of or exclusive in the Diesel cycle

An engine either has these features or it does not have them; correspondingly, it is of the Diesel type or it is

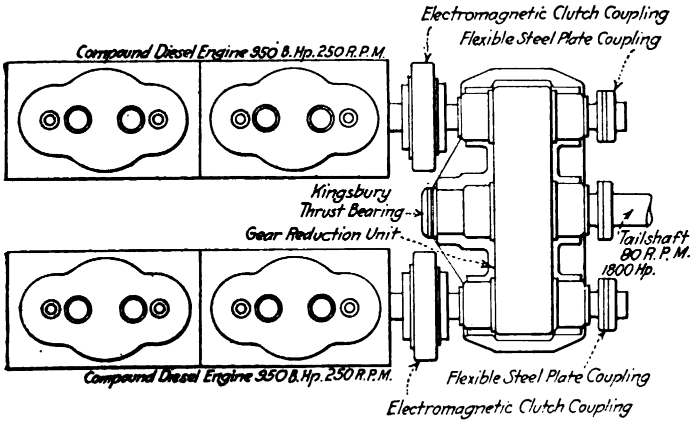


FIG. 36—TWO DOUBLE-UNIT ENGINES ARRANGED FOR MARINE USE BY CONNECTION TO A REDUCTION GEAR THROUGH ELECTROMAGNETIC CLUTCHES

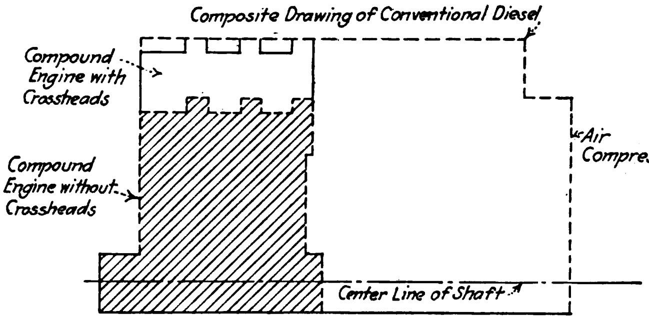


FIG. 37—COMPARISON BETWEEN THE COMPOUND ENGINE AND A SIMPLE DIESEL ENGINE FOR DIRECT DRIVE AT THE SAME PISTON SPEED AND NUMBER OF REVOLUTIONS

not. No air-blast was used for the first experimental engine that was built by Dr. Diesel. It was only after air-injection had been employed, when the first engine was rebuilt, that a sufficient number of revolutions per minute were obtained to secure indicator-cards of the whole cycle. The initial combustion obtained from the first engine blew the indicator to pieces. The true Diesel cycle, as now in worldwide use, is not in accordance with the master patents taken out in 1892, before Diesel made his conclusive experiments. It is in accordance with the design of the second engine, which was put on the test-bed at Augsburg in 1896. The final experiments that led to success were made upon this second engine, and 4 years were devoted to them. This engine used air-injection; hence, air-injection of fuel is an essential feature of the Diesel cycle. But this is not a distinctive or exclusive feature, because air-injection of fuel is used with oil engines that do not contain the other essential features of the Diesel cycle.

MR. GILL:—I asked for a definition of what a Diesel engine is because we have many engines today that are not Diesel engines according to the original conception and yet, if we include them in the list that has been given, they are Diesel engines. I think that we must draw the line somewhere. The Diesel cycle is what is called a constant-pressure cycle and, in the automobile engine, we have the constant-volume cycle. Dr. Diesel succeeded in obtaining pressures as high as 300 lb. per sq. in., but he did not have suitable material to withstand extremely high pressure. Nowadays, engines work at a pressure of 300 to 500 lb. per sq. in.; with solid injection, this means that there is no air condensation. Some of the Sperry class of engines hold to the same lines.

J. W. NORTON:—We can hardly say that the semi-Diesel engine holds to the same lines or class. The compression pressure is smaller, and some means of retaining the heat for the next charge is provided, such as the hot plate and hot bulb. These engines work around 300-lb. compression without spark-plugs, whereas the pure Diesel engine works around 400 to 500 lb. per sq. in. The diagram of a Diesel engine working on a constant-pressure cycle resembles that of a steam engine with a 10 to 12 per cent cut-off. The semi-Diesel and the automobile engine are mainly of the explosion or constant-volume cycle type and the diagram reaches a sharp peak, with the termination of the compression pressure half-way up, while in the true Diesel engine the compression line runs all the way up to the admission line.

CHAIRMAN HERMAN HOLLERITH, JR.:—I believe the fundamental difference to be that the Diesel engine burns its fuel at a constant pressure, and that the automobile engine burns its fuel at a constant volume. There are

variations between them. I believe that the engines Mr. Lisle has mentioned are all full Diesel engines.

MR. LISLE:—No, because the letters patent distinctly state that the fuel shall be injected by air and at a higher pressure than the compression pressure within the cylinder.

MR. FRENCH:—None of the engines mentioned is a true Diesel engine. Dr. Diesel's invention called for an engine that would burn powdered coal.

MR. NORTON:—It was Dr. Diesel's intention to use a very high pressure after the theory mathematically worked out that the higher the compression pressure, the higher the thermal efficiency. In the Diesel engine *pure air* is compressed and fuel is injected at the highest pressure. That is why the Diesel engine can work with high pressure-ranges. In engines using gasoline and kerosene, or in other words, volatile fuel, the pressure range is lower, due to a premixture of air and fuel, and possible backfire. Dr. Diesel used coal as fuel in his first engine and a compression-pressure around 3500 lb. per sq. in. It did not work out according to expectations, so it was changed to burn fuel oil and the compression-pressure lowered.

The two-cycle and four-cycle engines that Mr. Lisle has shown can be made both single and double-acting, but they are the only types we have. The difference in appearance of the engines is due solely to the mechanical features of the two types.

JOHN H. BARNARD:—I am still working with my engine to obtain regularity of performance. Like many experimenters, I began with too small an engine. It has only an 8-in. bore and a 10-in. stroke. That calls for only 0.01 cu. in. of fuel oil per stroke. Furthermore, it requires that the oil be sprayed progressively; in other words, the amount of oil delivered into the combustion-chamber must increase as the volume above the piston increases. Hence, instead of having the problem of trying to inject a small quantity of oil at what might be called an even rate of injection through a comparatively few degrees of the crank angle, which in the Diesel type averages 15 deg., I am trying to spray oil progressively through about 64 to 70 deg. of the piston stroke.

These spray requirements constituted the difficult part of the problem, but that has been solved. The present difficulty lies in the regularity of performance. It is difficult to obtain regularity with so small a fuel pump. Ordinarily a pump is used with Diesel engines that has a capacity considerably in excess of the engine's requirements with means for by-passing so much of its delivery as is in excess of the amount to be metered for each stroke. However, I cannot do that with progressive spraying, and it is very difficult to produce a small pump that will throw 0.01 cu. in. of fuel with absolute regularity. I have been forced into using an entirely novel form of pump. It is just now being made and I hope it will solve the difficulty, because this has now been located absolutely in the irregular performance of the fuel pump.

CHAIRMAN HOLLERITH:—Mr. Barnard has expressed one of the prime limitations of the small-sized Diesel engine; that is, the proper measuring of such very small quantities of fuel.

HARTE COOKE:—The fundamental difference between the Diesel cycle and other cycles is that, in the Diesel engine, only the air is compressed and, in the others, the explosive mixture is compressed to obtain the high efficiency required. With the ordinary engine the compression-ratio is limited if the explosive mixture will preignite. The ratio can be made whatever one pleases. Dr. Diesel originally intended to reduce the fuel at such a rate that it would burn at a constant temperature; in other words, he burned the fuel as the heat was extracted by the work on the piston. He found that this was not feasible; so, he used the constant-pressure cycle.

MUTUAL BENEFITS OF FOREIGN TRADE

(*Concluded from page 90*)

people is in their own powers of production. They must be able to use those powers and to pay in the products of their own labor or they cannot buy.

We are much inclined to magnify the danger of competition from other countries on equal terms. We hear much about the ability of this or that country to overrun and dominate the markets of the world. We used to hear that England could do it, but of late years Germany has been the bugaboo. It is possible, of course, that in given lines of industry certain countries may be superior to others; that situation is the very basis of international trade. There is not the least danger, however, that of its own choice any country will do more than its share of the work of the world. No people can work any more than all the time, and none of them want to give their goods away. They all want to get something for them. The great exporting nations always have been great importing nations. They have to be. They cannot sell their own products without taking the products of other countries. We have seen how trade balances in our favor have caused the exchange rates to rise against us, and put a check upon our exports.

The great lesson of the time is that of *the mutual interests which, rightly understood, tend to unite all countries in reciprocal and helpful relations*. Unfortunately, there is only a faint comprehension of the truth; and because this is so we have a world of rivalries and antagonisms which from time to time break out in war. The leading business interests of every country have a responsibility in this respect. The spirit of war is developed in mistaken ideas about national interest. If nations believe that their fundamental interests are in conflict, that there is irreconcilable rivalry and a struggle for existence, if people believe that the future of their country and their children is at stake, of course they will fight; nothing else can be expected.

But there is no such conflicting interest. There are trade rivalries, rivalries between traders of different countries, as there are between traders of the same country. Within proper limits they are stimulating and beneficial. But it is a mistake to emphasize them as though the success of one nation depends upon driving another out of the field. That idea is based upon the assumption that there is only a limited amount of business to be done and never enough for all and has been responsible for an infinite amount of mischief.

There is no limit to the amount of work to be done in the world, or of the amount of business to be had, because there is no limit to the amount of wealth that may be created from the resources of nature, or to the wants of the population.

We hear considerable about over-production, but there can be no such thing as general over-production of all the goods of common consumption. There is such a thing as *unbalanced* production, and we see it now; but there is not a family in a four-room tenement in this city that would not like to have six rooms, or one with six rooms that would not like eight, with everything else to correspond. The problem of society is to organize and integrate the productive powers of all countries to secure the greatest possible supply of the comforts of life for the population. That is the great appeal to the constructive forces of the world.

Tractor and Plow Reactions to Various Hitches

By O. B. Zimmerman[1] and T. G. Sewall[2]

Minneapolis Tractor Meeting Paper — *Illustrated with* Photographs and Charts

THE authors enumerate some of the questions that are involved and, after outlining a previous paper on the subject of plows, analyze these questions in part by the aid of diagrams and applied mathematics. Comparative draft data are presented in tabular form and commented upon, as well as comparative hitch-length data.

Tractor reactions are explained and discussed in some detail in a similar manner, special attention being given to the reactions on a slope and up-hill. The reactions on cross-furrow slopes are considered, comparisons being made between two tractors that were reported upon in the University of Nebraska tests. The factors involving tractor stability and resistance against overturn are analyzed. The authors state that the analysis presents a definite method of attack for the more correct solution of the proper hitching-point, as well as being a study relating to lug design.

THOSE familiar with the application of farm tractors to field use are well aware of the wide range of arguments concerning the questions of how to hitch the implements to a tractor and whether to run the tractor on the unplowed land or with one wheel in the furrow. Other questions are whether to hitch the

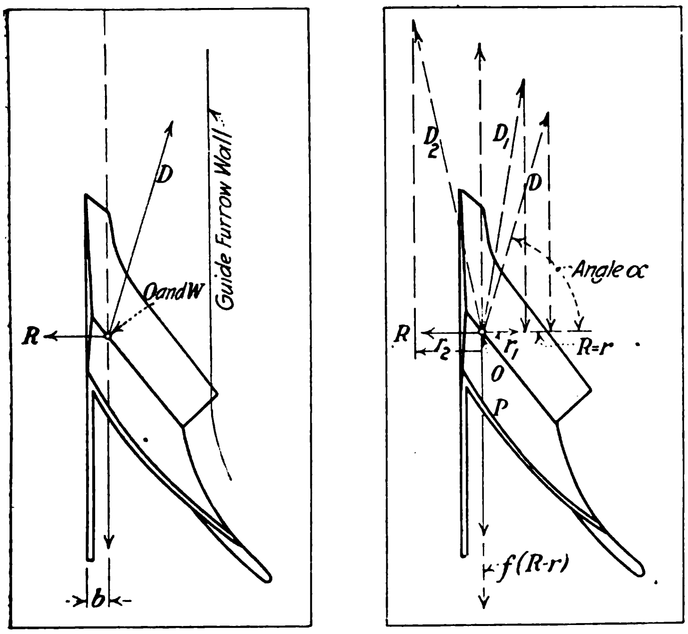


Fig. 1—Reactions with an Individual Plow

plow low or high on the tractor; whether the point of hitch shall be to the right or left of the plow center; where the center of plow reaction is located; what the nature of the reactions is; what the changes are in the

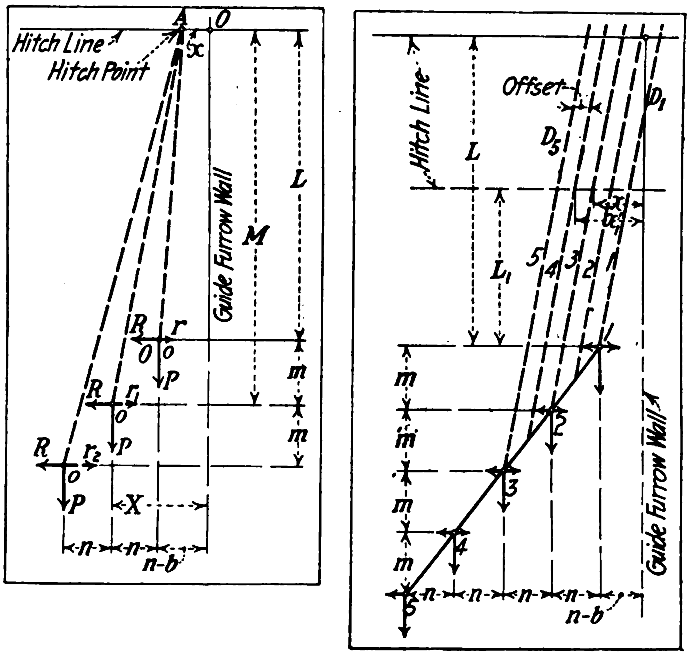


Fig. 2—Diagram Showing the Necessity for Exercising Care in Connection with the Hitch of a Multiple Plow

several reactions during motion as regards weight and stability; and how these changes affect tractor stability against overturn. The paper endeavors in part to analyze the answers to these questions.

In an article on Coordination of Theory and Practice in the Design and Operation of Plows, by A. C. Lindgren and O. B. Zimmerman,[3] which covers the subject of plow reactions, a careful analysis is given as regards the plow and this subject can be reviewed in its extended form in that article. But it is now desirable to carry this analysis still farther so as to cover the tractor actions and reactions also, and so make it more complete. Therefore, a review of the deductions of the article to which reference has been made becomes necessary.

In brief, we must start with the individual plow shown at the left of Fig. 1, in which the complex series of primary reactions can be considered best as having been resolved and integrated into one vertical force, W, one cross-furrow force, R, and one down-furrow force, P. These reactions will vary in magnitude as they are developed, due to the weight of the outfit, the soil operated upon, the condition of the plow, the shape of the moldboard, the speed of operation, the depth of plowing, the width of cut, the nature of the soil, the moisture content, the relative friction between steel and earth, and the like. It can be shown how these forces may be considered as acting upon one point, O, which will be called the center of reaction of the plow.

To overcome these reactions, we have the active force,

[1] M.S.A.E.—Engineering staff, International Harvester Co., Chicago.
[2] Experimental department, International Harvester Co., Chicago.
[3] See *Journal of the American Society of Agricultural Engineers*, January 1922, p. 3.

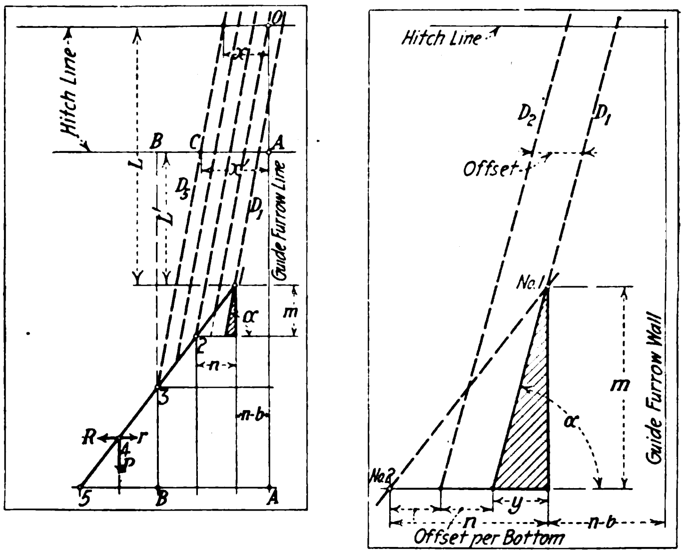


Fig. 3—Development of Formulas for the Correct Hitch with Multiple Rigid Units

D, or the draft, which is shown more clearly at the right of Fig. 1. This draft line, D, may vary in direction. A study of this direction of pull shows that we can add to or reduce all three primary forces, W, R and P, but very noticeably the cross-furrow force, R, and the down-furrow force, P, by overcoming the sliding friction between the plow landside and the furrow wall, which friction causes wide variation in D, according to the angle of pull. If D acts to the right of O as shown in both portions of Fig. 1, its cross-furrow component r opposes R. The force R is practically constant under any given set of conditions. When r equals R, the force D is at its minimum. Likewise, if we direct D to the left of O, we increase the total to $R + r$. However, a reasonable pressure in the direction of the furrow wall is necessary for stable action. The argument concerning this feature should be reviewed in the article that has been cited.

The multiple-plow unit quickly shows us the necessity of care in the matter of hitch because, as shown in the left half of Fig. 2, if each individual plow is permitted to receive its draft D from a common hitch-point, it is clear that the cross-furrow action of R and r must be different; hence, we must unite them rigidly or provide at least for possible interchanges of pressure between them so that they will work as a unit. Otherwise, each plow will tend to interfere with the others. Plow No. 1 next the furrow wall tends to crowd to the left, and plow No. 3 crowds to the right. The right-hand portion of Fig. 2 and both drawings in Fig. 3 illustrate the development of the formulas for the correct hitch in multiple rigid units.

Let

α = Angle of D with respect to the cross-furrow line
b = Distance of center of individual plow reaction to plow landside
D = Draft in the drawbar, to obtain the relations of draft
f = Coefficient of friction of earth on steel
L = Distance of hitch line from the center of reaction of plow No. 1
M = Spacing between plows, down furrow
N = Number of plow bottoms
n = Width of cut of a single plow
O = { Location of combined center of reaction from the furrow wall; Hitch distance from the hitch line to the center of reaction }
P = Down-furrow draft without furrow-wall friction
R = Cross-furrow reaction due to shape of moldboard, etc.
r = Cross-furrow component of D
x = Hitch-point distance from the furrow-wall measured on the hitch line

Then

$$D = \sqrt{([P + f(R - r)]^2 + r^2)} \qquad (1)$$

$$O = (w - b) + (N - 1)\, n/2 \qquad (2)$$

$$O = L + (N - 1)\, m/2 \qquad (3)$$

$$x = (n - b) + [N - 1] \times ([(n - m) \cot \alpha] \div 2) - L \cot \alpha \qquad (4)$$

$$\tan \alpha = \frac{P + f\,(R - r)^2}{r} \qquad (5)$$

The conclusion follows that the correct hitch direction should lead to the right of O in all cases by an amount that will reduce the friction to a practical minimum. Here is where the tractor plays its part and receives these reactions through the various styles of hitch, either rigid or flexible. The basic principle of flexible-hitch forms is illustrated in Figs. 4 and 5. From the foregoing text and from the illustrations, we can determine the desirability of long or short hitches, and high or low hitching.

In the field, we have a considerable range of hitch distance from the furrow wall, any point of which will answer for practical operation. Beyond these limits, we find that if we hitch the tractor drawbar to the right of this range the plows will run into the plowed land until the reactions balance. The plows then will not cut the full width for which they were designed. If we hitch to the left of this range, the furrow wall will crush down and the plows will run to the left until the reactions balance in that direction, leaving part of the area at the right unplowed. Standard tractor-drawbar adjustments in regard to plow-hitches were adopted March 11, 1922, and have been published.[4] The values given are based upon the best practical and theoretical arguments. The correct limits to the right are for light or fluffy soil; hence, as shown in Fig. 5, various hitch points are indicated from light, medium, heavy and very heavy soil as shown by draft values and locations L, M, H and $V\ H$. These values indicate that for an equal cross-furrow reaction, (R-r), the heavier soils require a hitch point closer to the furrow wall than lighter soils.

[4] See S. A. E. Handbook, p. K-40.

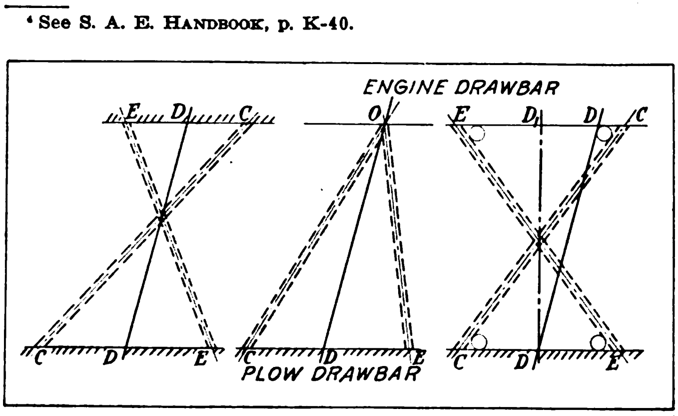


Fig. 4—Diagram Illustrating the Basic Principle of the Rigid Hitch for Plows

Comparative Draft Data

Figs. 6 and 7 are given with Tables 1 to 6 to emphasize the various relations that result from varying the hitch point under a given set of data, assuming that the plow units are designed without complete relief to the cross-

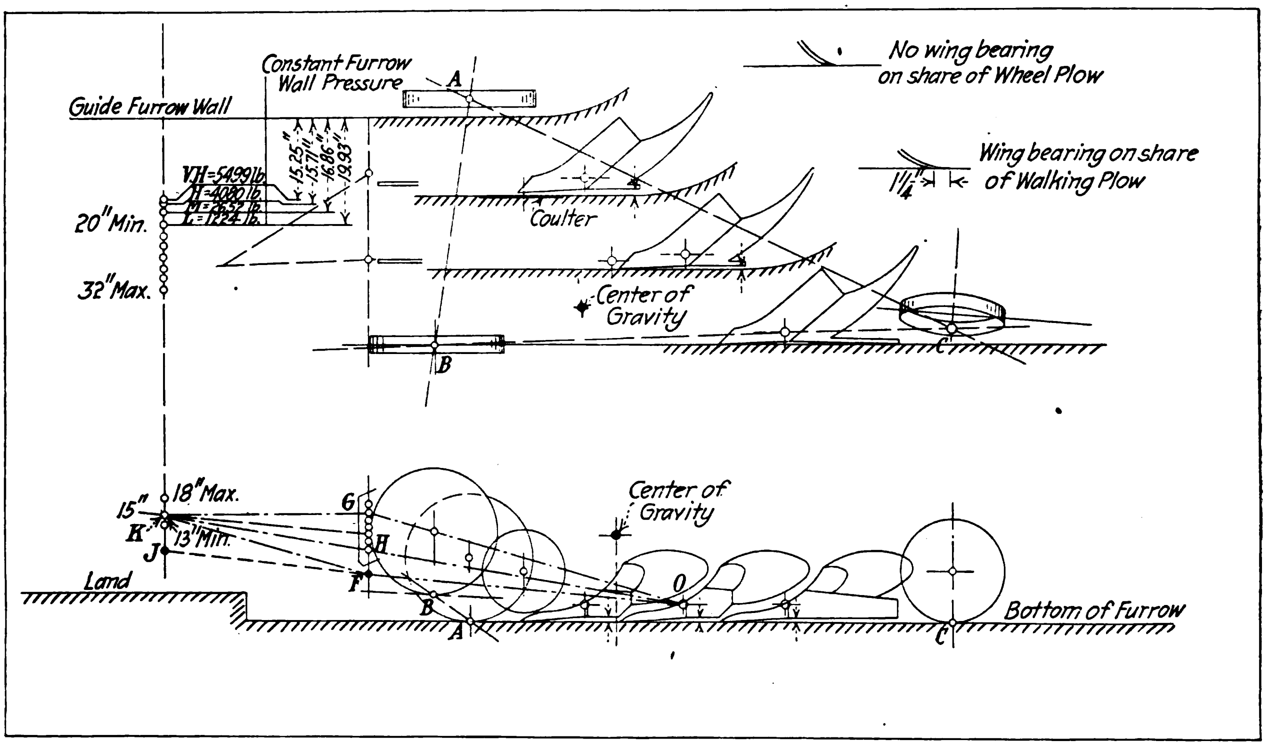


FIG. 5—HITCH POINTS FOR LIGHT, MEDIUM, HEAVY AND VERY HEAVY SOILS

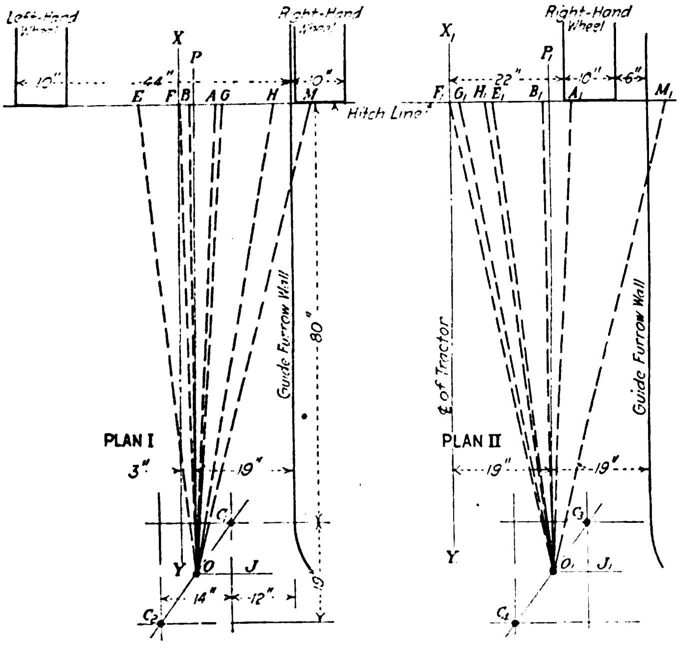


FIG. 6—RESULTS OF VARYING THE HITCH POINT FOR A GIVEN SET OF CONDITIONS WITH A TWO-BOTTOM PLOW

TABLE 1 — DISTANCES FROM THE CENTER LINE OF THE TRACTOR OF POINTS OF EQUAL DRAFT; TWO BOTTOMS

	Point Distance. In.			
Plan I	$M=+25.4$	$A=+7$	$B=+2$	$E=-8$
Plan II	$M_1=+41.4$	$A_1=+23$	$B_1=+18$	$E_1=+8$

furrow pressure by carrier wheels. While these carrier wheels ease off the plow pressure, we must still take note of their values since the wheel bearings or the tractor as well as the plow must take up these pressures; hence, a knowledge of their magnitude is valuable. Fig. 6 represents the conditions when a tractor pulls a plow having two 14-in. bottoms. The overall dimensions and tractor wheel widths are taken as being averages of the output of some 20 prominent manufacturers. Plan I represents the conditions when the tractor runs in the furrow; in Plan II it runs on the land. The lines O P and O_1 P_1 are down-furrow lines through the center of reaction of the plow unit, and lines X Y and X_1 Y_1 are the center lines

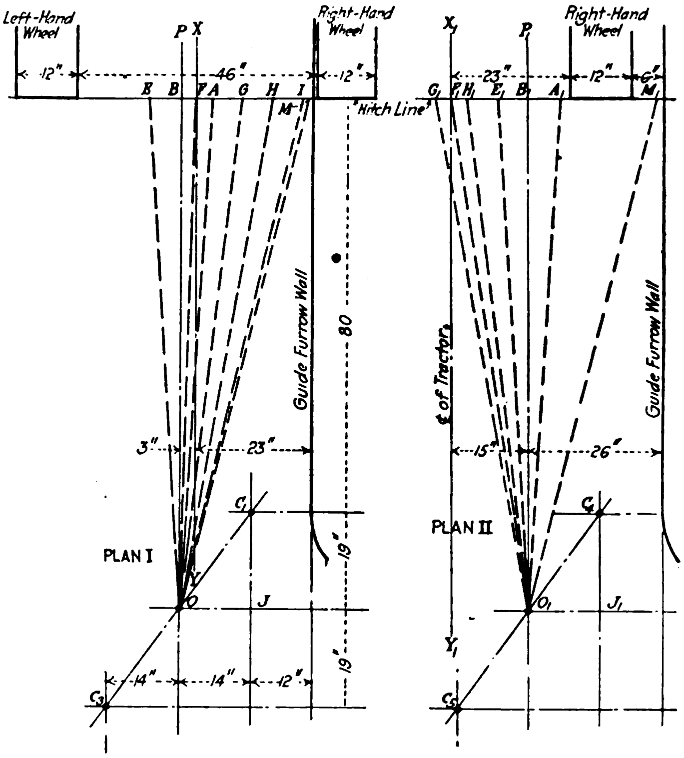


FIG. 7—RESULTS OF VARYING THE HITCH POINT FOR A GIVEN SET OF CONDITIONS WITH A THREE-BOTTOM PLOW

TABLE 2—DISTANCES FROM THE FURROW-WALL OF POINTS EQUI-DISTANT FROM THE CENTER LINE OF THE TRACTOR; TWO BOTTOMS

	Point Distance, In.				
Plan I	$F = +22$	$B = +20$	$A = +15$	$G = +14$	$H = +4$
Plan II	$F_1 = +38$	$G_1 = +36$	$H_1 = +31$	$E_1 = +30$	$B_1 = +20$

of the tractor. The points A, B and E and A_1, B_1 and E_1 represent the hitching points as recommended by the Society; namely, 15 in. as a minimum, 20 in. as a proper or best average position and 30 in. as a maximum, respectively, from the guide-furrow wall. A comparison of the hitching points as to (a) distance from the center line of the tractor and distance from the guide-furrow wall and (b), drawbar pull, is given in Table 1, using formulas (1) and (5).

TABLE 3—TOTAL DRAWBAR PULL AT POINTS THE SAME DISTANCE FROM THE CENTER LINE OF THE TRACTOR; TWO BOTTOMS

	Drawbar Pull at Point				
Plan I, lb.	$F = 1{,}033$	$B = 1{,}010$	$A = 958$	$G = 949$	$H = 869$
Plan II, lb.	$F_1 = 1{,}188$	$G_1 = 1{,}170$	$H_1 = 1{,}124$	$E_1 = 1{,}116$	$B_1 = 1{,}010$
Increase, Plan II over Plan I, lb.	155	160	166	167	141
Increase, Plan II over Plan I, per cent	15.0	15.8	17.3	17.6	16.2
Average per cent of Increase	....	....	16.4	....	....

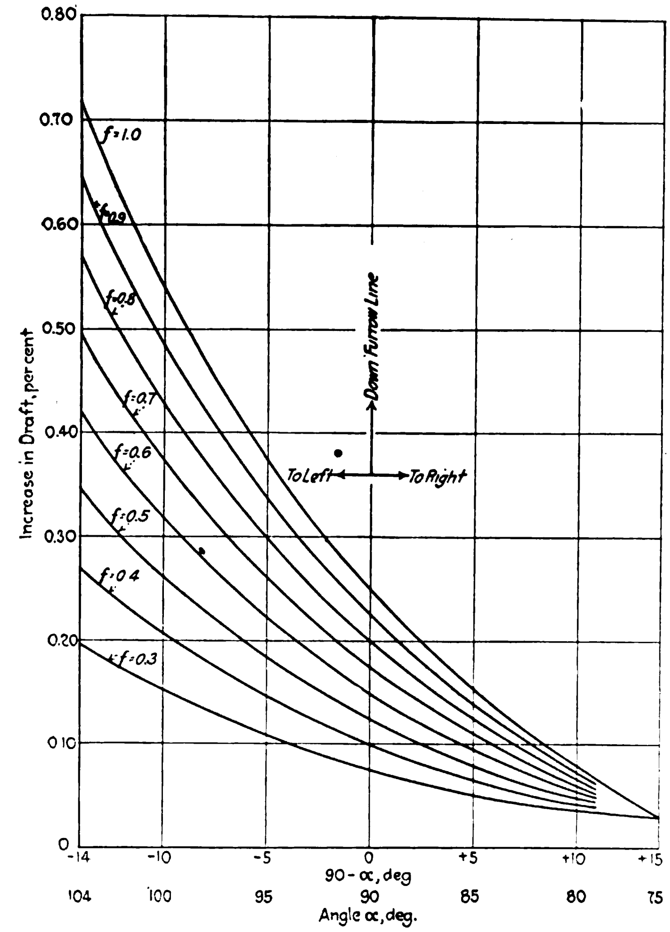


FIG. 9—RELATION BETWEEN THE INCREASE IN THE DRAFT AND THE ANGLE THAT THE LINE OF DRAFT MAKES WITH THE CROSS-FURROW LINE

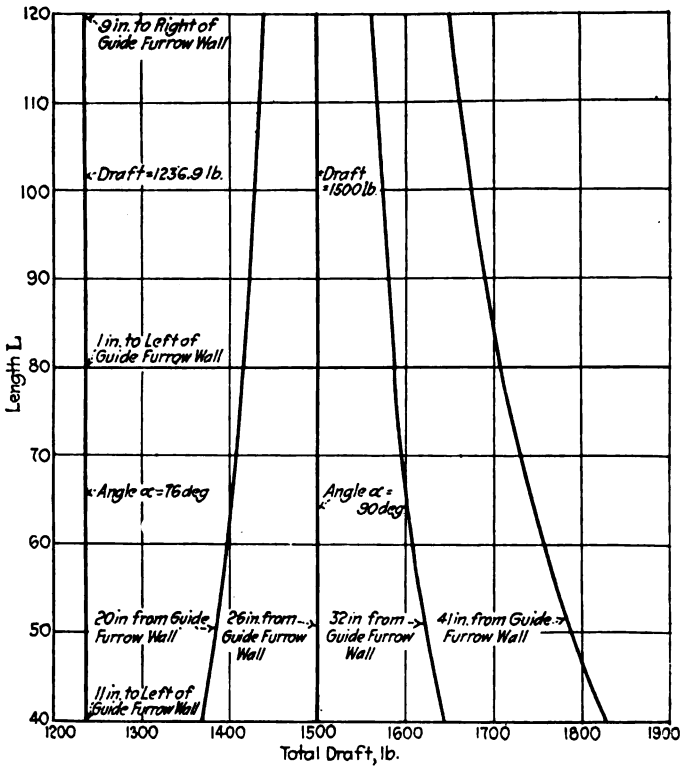


FIG. 8—RESULTS OF CHANGING THE LENGTH OF HITCH BETWEEN THE DRAWBAR OF A THREE-PLOW TRACTOR AND THE FIRST PLOW

TABLE 4—DISTANCES FROM THE CENTER LINE OF THE TRACTOR OF POINTS OF EQUAL DRAFT; THREE BOTTOMS

	Point Distance, In.			
Plan I	$M = +21.75$	$A = +3$	$B = -3$	$E = -9$
Plan II	$M_1 = +39.75$	$A_1 = +21$	$B_1 = +15$	$E_1 = +9$

The lines $M\ O$ and $M_1\ O_1$, have the angle α equal to 76 deg., where the draft is least, on account of the removal of all side pressure against the furrow wall in the case taken. The drawbar pull at the points A, B, E and A_1, B_1, E_1, for both plans is the same. A tractor running in the furrow, therefore, can use a hitch nearer to the center line of the tractor.

COMPARATIVE HITCH-LENGTH DATA

Fig. 8 is developed to emphasize still further the values resulting from changing the length of hitch between the first plow and the tractor drawbar.

It is clear that if we desire the tractor to operate on the land in preference to operating in the furrow, and

TABLE 5—DISTANCES FROM THE FURROW-WALL OF POINTS EQUI-DISTANT FROM THE CENTER LINE OF THE TRACTOR; THREE BOTTOMS

	Point Distance, In.					
Plan I	$B = +26$	$F = +23$	$A = +20$	$G = +14$	$H = +8$	$I = +2$
Plan II	$G_1 = +44$	$F_1 = +41$	$H_1 = +38$	$E_1 = +32$	$B_1 = +26$	$A_1 = +20$

TABLE 6—TOTAL DRAWBAR PULL AT POINTS EQUI-DISTANT FROM THE CENTER LINE OF THE TRACTOR; THREE BOTTOMS

	Drawbar Pull at Point					
Plan I, lb.	B=1,500	F=1,457	A=1,417	G=1,348	H=1,290	I=1,242
Plan II, lb.	G_1=1,746	F_1=1,709	H_1=1,670	E_1=1,588	B_1=1,500	A_1=1,417
Increase, Plan II over Plan I, lb.	246	252	253	240	210	175
Increase, Plan II over Plan I, per cent	16.40	17.30	17.85	17.80	16.28	14.10
Average, per cent of increase	....	....	16.62	....	....	....

with hitches near the tractor center, a long hitch is desirable with two and three-plow units. If we develop formula (1) through all the varying angles within the range of practical operation from right to left, we begin to realize the effect that results from choosing a direction of draft D correctly, especially for outfits which do not provide for relief of the R or cross-furrow pressure through carrier wheels, which not only support the unit but can relieve this thrust when set at a slant to the vertical so as to oppose this thrust. The several values of f are some of the coefficients of friction between various earths and steel. In Fig. 9, the 90-deg. angle represents a straight down-furrow draft through O, the center of plow reaction. The 85, 80 and 75-deg. angles indicate the angles of draft that D makes with respect to a cross-furrow line and, likewise, the 95, 100 and 105-deg. angles represent similar draft lines to the left of O.

The vertical scale indicates the percentage of increase above the original value of P, such as P = 400 lb. When D is pulling so that the angle α = 85 deg. in soil that has a coefficient of friction of 0.5, we find that the draft is 8 per cent higher than the base figure P. If we should set the hitch so as to develop a pull 10 deg. to the left with f = 0.5, we would have a draft 26 per cent higher than the base figure, which is that much higher than is necessary.

An attempt to visualize the series of variables is given in Fig. 10. Here we have a space relation in which the three coordinates are as follows:

(1) The ratio of P to R or the relation of the down-furrow force P to the cross-furrow force R that exists at the moment under consideration when the plow is in operation and when we are comparing the various kinds of moldboard, those having a long slope to those the slope of which is very abrupt. These ratios have been deduced only roughly from judgment and practical operation. The diagram necessarily does not cover all of them, but it serves a useful purpose

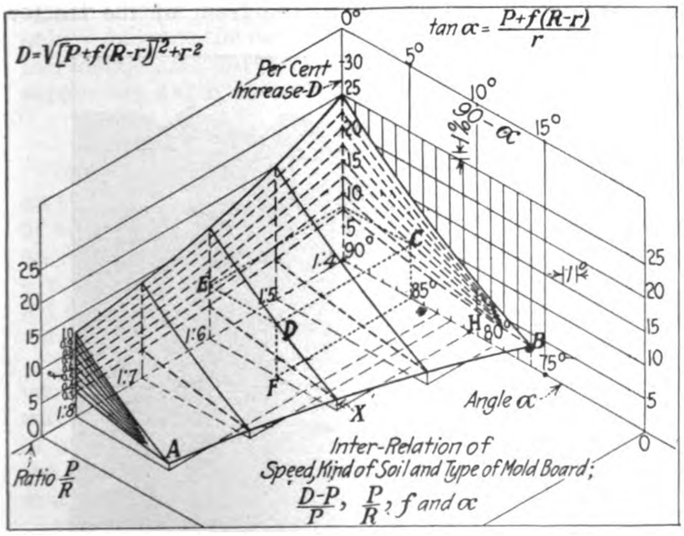


FIG. 10—THREE-DIMENSION DIAGRAM SHOWING THE INTER-RELATION OF THE SPEED, KIND OF SOIL AND TYPE OF MOLDBOARD

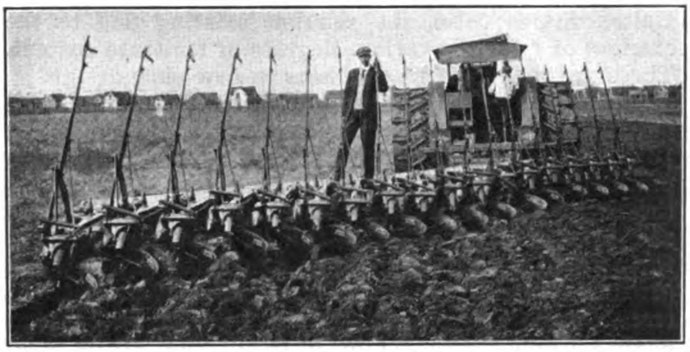


FIG. 11—AN 18-BOTTOM PLOW BEING DRAWN BY A SINGLE TRACTOR AT THE RATE OF 1 ACRE IN LESS THAN 12 MIN.

(2) The angles of pull of D between the center of plow reaction and the hitch point, with respect to either a cross-furrow line like R having values of α from 90 to 75 deg. or, as shown above, as the angle to the right of a down-furrow line like P, such as 90 deg. — α. The effect of angles of pull to the left of the down-furrow line, or those more than 90 deg., can be interpreted roughly by imagining an extension of the warped planes through the left coordinate planes and in accord with Fig. 9

(3) The percentage of increase of pull D over and above that needed is shown in percentages, from 0 to 25. As the draft D is affected markedly by the coefficient of friction f, this feature is incorporated also. These relations form the surfaces that appear like the leaves of a book

To interpret Fig. 10, let us consider a case where we are hitching so that the draft line D makes α = 85 deg. or 5 deg. from the down-furrow line and pulls to the right. Run down the line marked 85 deg. to F, which lies in the cross plane that represents P to R as being 6 to 1, then rise until the line pierces the plane as at D where f = 1. Then F D, read on the percentage scale, shows 8 per cent. This 8 per cent less X, which is 1 per cent, gives us a 7-per cent, net, greater draft for D with a hitch at 85 deg. than would be shown if the hitch had been set at H, or at 81.5 deg. run out from x. At 81.5 deg., the cross-furrow force would be zero; hence, the practical angle would be somewhere between 81.5 and 85 deg., according to the desired furrow-wall pressure and such other practical accommodation as is necessary.

A review of Fig. 10 will clear up the value of these relations, when thinking relatively of light and heavy-soil operation. We can say then that when a plow is operating in a field of varying resistance, the draft D will vary with a given angle of hitch, passing, say, from plane f = 1 to plane f = 0.6 back and forth. Also, as the speed might vary, causing the P to R relation to go from 1 to 6 to 1 to 5, and if for any reason the angle of hitch should vary a few degrees due to hill-slope plowing, say between 4 to 7 deg. from the down-furrow line, the draft would follow the corresponding relations according to the several coordinate values. This emphasizes the desirability of attempting to establish better the technical relations of P to R. These relations should be developed and also the values of f covering the typical soils tilled in the

United States; also, the relation existing due to the changes of f for the various degrees of moisture content. These are real research problems in new plow design.

The foregoing deductions are correct for large units pulling numbers of bottoms. In Fig. 11, a plow having 18 bottoms is being drawn by one power unit and this unit is plowing at the rate of 1 acre in less than 12 min., the center of the tractor being seen to be just ahead and between plows Nos. 6 and 7, counting from the furrow wall, thus leaving 11½ plows to the left. Fig. 12 gives an example of the mass cooperation of three 45-hp. tractor units that are pulling plows having 55 bottoms, again showing the application of minimum draft by relief of the cross-furrow reaction allowing perfectly true draft. The clean-cut furrow-wall is evident at the left, with no crushing down in this light soil. The resistance was about 330 lb. per 14-in. bottom. An area was plowed with the larger unit at the rate of 1 acre in less than 4 min.

Tractor Reactions

The development of the following formulas and discussion of them will apply equally well to any self-propelled wheeled type of motive power, carrying loads, hauling or doing both, over level ground, up and down hill or on hillsides where the slope affects side tipping. This includes tractors, trucks, automobiles, power-driven cultivators and the like.

Starting with the engine as the source of power, there is a limit to its average maximum effort. For successful continuous operation, the amount of power at the source must be greater than the sum of all resistance; also, means for exerting this power must be provided through wheel reaction to the soil underneath the tractor. Therefore, the engine power must be greater than the friction of transmission, plus the load, plus the rolling resistance, plus any lifting of the tractor due to the slope of the land traversed. In this paper we are concerned principally about the effects of power or torque being delivered to the rear wheel by gears, chains or a worm, the overcoming of the rolling resistance, the lifting up-hill of the unit and the drawbar resistance, as well as in regard to how those factors fluctuate.

Let us first consider the tractor statically. In Fig. 13, we have the weight of the tractor concentrated at its center of gravity, a height H from the ground. This total weight is distributed front and rear in values of w and W. When on the level, the perpendicular through the center of gravity divides the wheelbase line into the two parts x and $k-x$, from which we develop:

$$Wx = w\,(k - x) \text{ or } Wx + wx = wk; \text{ and } x = wk/(W + w) \quad (6)$$

$$(k - x) = k - wk/(W + w) \text{ and } (k - x) = Wk/(W + w) \quad (7)$$

This distribution of weight is the same for forward motion of the tractor when there is no rolling resistance and no load. As soon as resistance is offered underneath the wheels, as is always the case, the weights change in value from W and w as follows:

The transmission chains, gears or worm must exert added power to advance the tractor and, therefore, we have a lifting tendency on the front end of the tractor. If C is the chain pull and G the sprocket diameter, the tangential reaction at the ground below the rear wheel would be equal to the torque $C \times G/2 = E \times R/2$ and the lift action x at the front wheel would be $xk = CG/2$. Hence

$$x = CG/2k = \text{Torque} \div 2k = w_1 \quad (8)$$

This value can be understood better when we think of great resistance to translation; that is, to a point where forward motion is impossible. After a period of wheel slippage that buries the tractor, the chain then tends to wind on the sprocket and unwind on the gear, revolving the tractor about the rear axle as a center with a tendency toward overturn. The amount of this lift is as stated in formula (8), if it is within the capacity of the engine. The inertia of moving parts will add to or subtract from the value in accordance with the fluctuations in the rate of travel.

Let us suppose the case of a tractor running over medium soil and capable of a drawbar pull just sufficient to haul four similar outfits behind it. Then we have a rough measure of this resistance, in lieu of some better means of dynamometer measurement. We can specify the rolling resistance, R R, in terms of D in either case, or we would have a coefficient in percentage of D. Take the four outfits mentioned above; the coefficient would be

$$D/4 = 0.25\,D. \quad D/4 \times R/2 \doteq CG/2 = xk.$$
$$x = DR/8k \text{ or } 0.25\,DR/2k = 0.125\,DR/k = w_1 \quad (9)$$

The relation of tractor-wheel diameter with respect to rolling resistance was developed in some degree by A. F. Moyer,[5] but much work still remains to be done on this subject.

If, in addition, we have a developed load on the drawbar, the actual lift exerted on the front of the tractor

[5] See S. A. E. Transactions, vol. 13, part 1, p. 405.

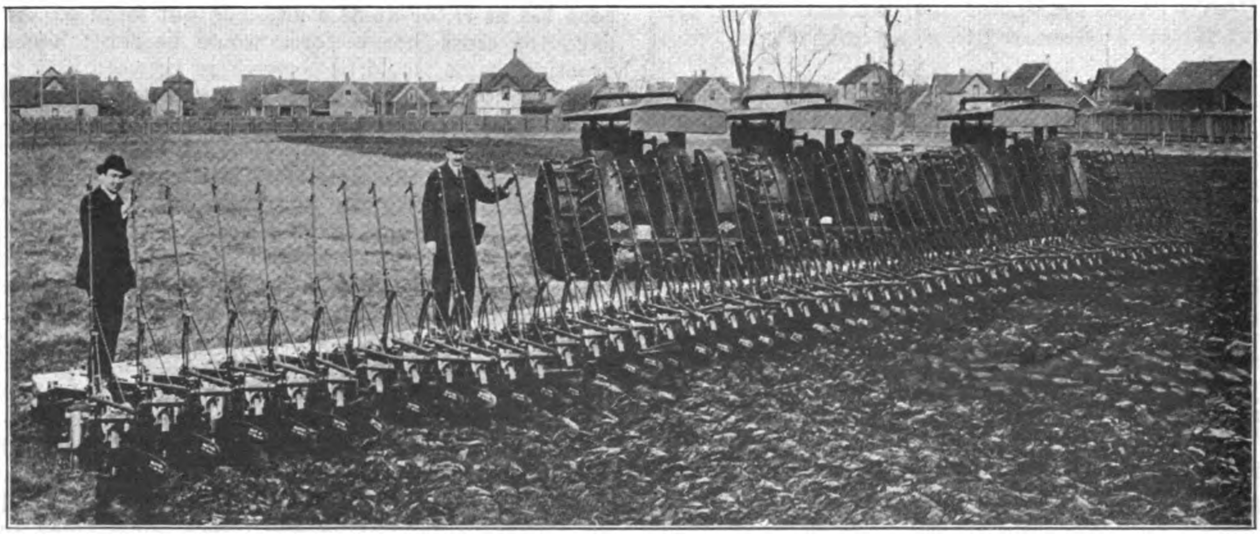


Fig. 12—Three 45-Hp. Tractors Pulling 55 Plow Bottoms

will again be resisted by the front weight at the end of its lever arm. The point about which the rotative action takes place can be considered as at the ground contact of the drive-wheel rim. If the resistance exerted by the drawbar amounts to a true anchor, one of two things occur. The engine will be killed by the reaction, in the endeavor to lift off the front weight w, or there will be a combination effect of a slight backing of the tractor and a lifting off the ground of the front. The danger of overturn depends on the sum of the reactions of steady power and inertia of the moving parts and the slope of the ground. In any event, the rolling resistance and drawbar reaction during translation tend to lighten the ground load under the front wheel and transfer these loads to the rear. In Fig. 13 we see the development of the formula by which we can calculate the drawbar-lift value. In the plow and the drawbar values we have

$$(h + d/2) : L :: x_1 : (L + a) \text{ or}$$
$$x_1 = [(h + d/2)\ (L + a)] \div L$$
$$x_2 = (x_1 - d/2) \cos \beta.$$

Combining we have

$$x_2 = [\{ (h + d/2)\ (L + a) \div L \} - d/2] \cos \beta \qquad (10)$$

Then the lift in front $= D\,x_2 = w_2$ or

$$w_2 = D/k\,[\{ (h + d/2)\ (L + a) \div L \} - d/2] \cos \beta \qquad (11)$$

Again, because we are working on compressible material, the actual supporting surface in translation is

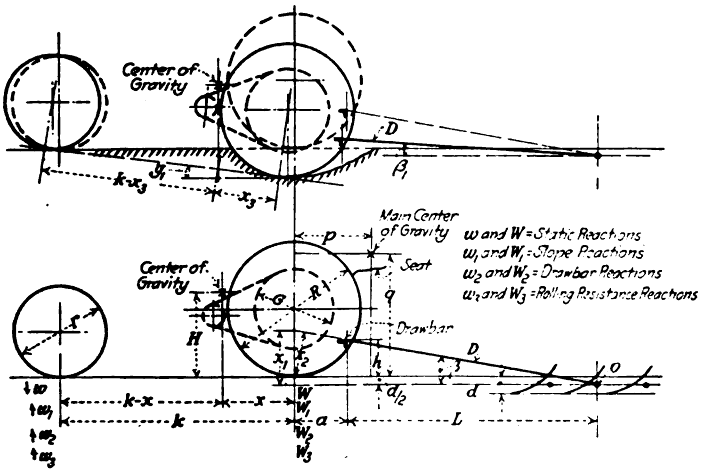


FIG. 13—DIAGRAM SHOWING THE VARIOUS REACTIONS IN THE TRACTOR

ahead of the vertical center-line through the rear axle by an amount which tends to set forward both support points, but not the center of gravity of the tractor, thus increasing still further the net weight carried by the supporting earth beneath the rear wheels.

REACTIONS ON A SLOPE AND UP-HILL

To study the effect of slope in the direction of travel on the distribution of the tractor weight under the points of wheel contact, let us refer to Fig. 14, using an engine angle γ with the horizontal; the wheelbase is as before, $a\ e$. On the level, the distribution of weight fore-and-aft due to the total weight at the center of gravity is inversely proportional to the distance $a\ c$ and $a\ e$.

To ascertain the relative value of these weights, it is necessary to develop a formula in accordance with the way the wheelbase line, $a\ e$, is divided by the projection of a perpendicular through the center of gravity, as at the point d, in Fig. 14. We then have w_1 the new weight at a_1 and W_1, at e such that

$$W_1 + w_1 = W + w \text{ or, } W_1 = w + w - w_1 \text{ and}$$
$$w_1 = W + w - W_1 \qquad (12)$$

also,

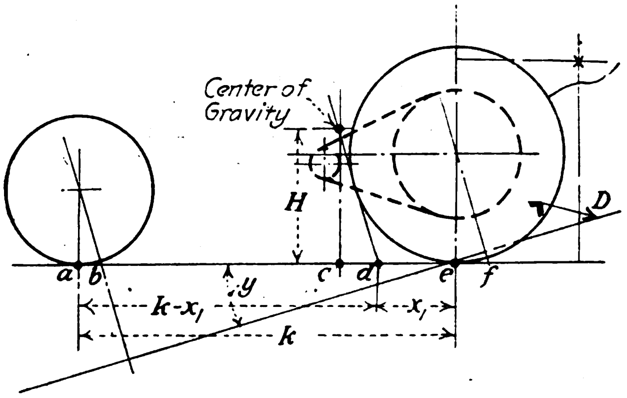


FIG. 14—EFFECT OF SLOPE ON THE TRACTOR

$$W_1ad = W_1de;\ ad = ac + cd \text{ and } de = ce - cd$$
$$w_1\ (ac - cd) = W_1\ (ce - cd)$$

Substituting the value of $a\ c$ and $c\ e$ in terms of W, w and k

$$w_1\ (Wk/W + w) + w_1H \tan \alpha = W_1\ (wk/W + w - H \tan y) \qquad (13)$$

Substituting values from formulas (12) and (13), first for w_1 and then for W_1 and reducing, we get

$$W_1 = W + (W + w/k)\ H \tan y \qquad (14)$$
$$W_1 = w - (W + w/k)\ H \tan y \qquad (15)$$

After interpreting these formulas, we find that the change in weight front and rear is directly dependent on the size of the angle of slope and inversely proportional to the wheelbase k. The original front weight decreases and the rear weight increases when going up-hill by the increments shown. Also, the higher the center of gravity is, the greater the change will be.

CROSS-THE-FURROW SLOPES

We still have to review the reactions covering the cross-furrow hillside-actions, the turning moment met by the front wheels in steering due to pulling off-center, the effect of the operator in the seat, and the like. With what has gone before, these forces and reactions can be determined readily. We can note the tendency to slip down-hill in Fig. 15 and how the weights front and rear which, on the level ground can be considered as divided equally between the two front and the two rear wheels, on hillsides are split into further differences in the same general manner as the longitudinal slope reactions.

Let W denote the weight of the tractor drive-wheel end and $W/2$ be the normal weight on each rear wheel. Let N equal the tractor width over the drive-wheel rims and H_1 the gravity-center height between these wheels. Then the load on the down-hill wheel will be to the load on the up-hill wheel as the perpendicular divides the ground line and we have:

$$W_u \times N/2 + H_1 \tan \delta = W_d \times N/2 - H_1 \tan \delta \qquad (16)$$

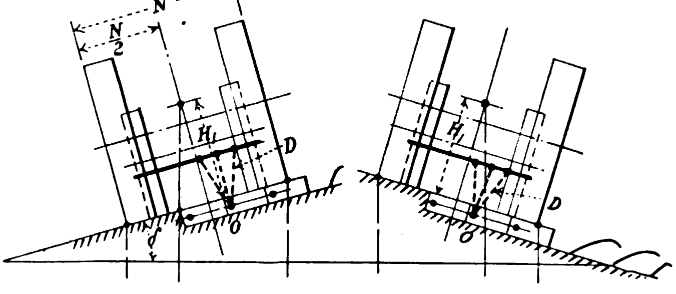


FIG. 15—DIAGRAM SHOWING THE TENDENCY OF THE TRACTOR TO SLIP DOWN HILL

$$W_u + W_d = W \quad \text{(For the rear of the tractor)} \qquad (17)$$
$$w_u \times n/2 + H_1 \tan \delta = w_d \times n/2 - H_1 \tan \delta \qquad (18)$$
$$w_u + w_d = w \quad \text{(For the front of the tractor)} \qquad (19)$$

The drawbar pull D likewise has its influence on slip sidewise of the tractor drive-wheels, and a change in weight. With small plow units operating on hillsides sloping down to the left, the pull in D tends to stabilize the overturn if to the right of O. The reverse is true on hillsides sloping to the right, and the operator can see readily that these side slopes add to the theoretical and practical range of hitching in a manner to overcome the tendency of the load to range. Since formulas at times do not bring out points so clearly as a concrete example, an endeavor is made to develop a comparison of two tractors by the use of formulas to indicate the relative value of these tractors as to stability. We have selected two popular tractors which were reported upon in the University of Nebraska tests.[6] Their weight data as given are supplemented by available data as to their centers of gravity and the like, as shown in Table 7.

TABLE 7—COMPARISON OF TWO TRACTORS

Data	Tractor A	Tractor B
Total Weight, lb.	2,710	5,708
Static Front Weight, lb.	1,075	1,890
Static Rear Weight, lb.	1,635	3,818
H, in.	28.0	33.5
p, in.	5	30
q, in. (Height of Center of Gravity of the Man on the Seat)	41	50
Engine Speed, r.p.m.	1,006	550
Drawbar Pull at the Speed Given, lb.	1,428	1,850
Drawbar Horsepower	8.26	15.65
Tractor Speed, m.p.h.	2.17	3.17
Wheelbase, in. ($= k$)	63	92
Rear Axle to Drawbar, in. ($= a$)	12.0	20.5
h, in.	15.0	15.0
L[7], in.	89.5	99.0
d, in.	6.0	6.0
r, in.	27.5	36.0
R, in.	42.0	54.0
G, in.	...	31.0
$k - x$, in.	38.0	61.5
x, in.	25.0	30.5

[7] Two-bottom for Tractor A and three-bottom for Tractor B.

To put the two on a more comparable basis we will interpolate the drawbar pull with respect to the speed in miles per hour. The ratio will be the same whether we

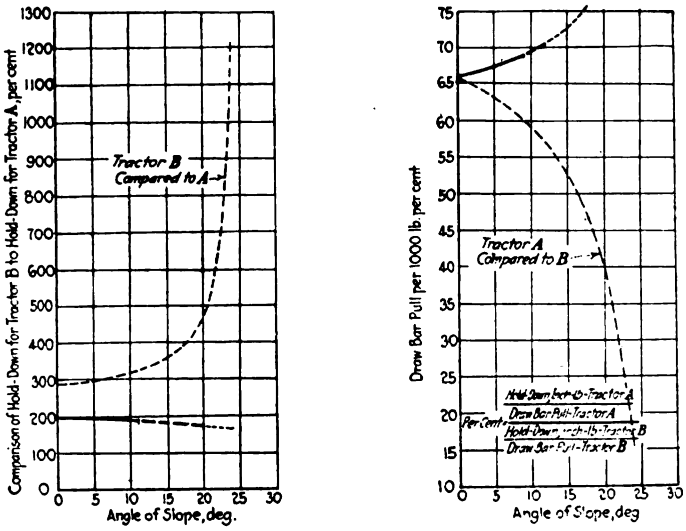


FIG. 16—RELATIVE STABILITY OF THE TRACTOR AGAINST OVERTURNING

[6] See THE JOURNAL, May, 1921, p. 391.

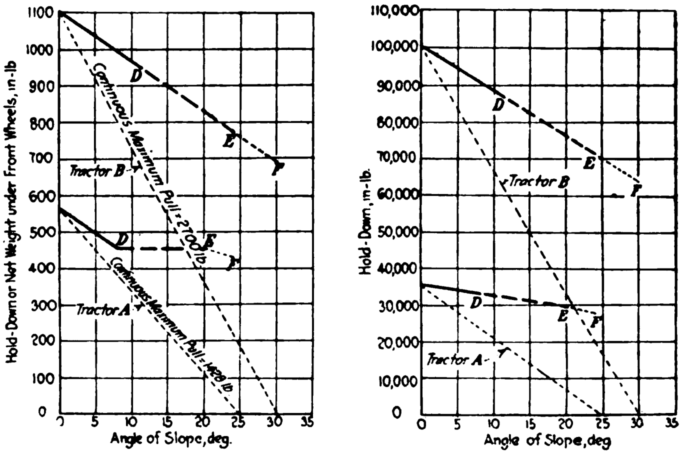


FIG. 17—COMPARISON OF TWO TRACTORS UNDER A CONSTANT OPERATING LOAD AT RATED ENGINE SPEED

increase the speed of one or decrease the speed of the other, or whether we assume an intermediate desired speed and bring both to that speed. Then (1850 lb. × 3.17) ÷ 2.17 = 2700 lb. This is a figure closely in line with the maximum pulled per the test; hence, this figure is practical to use. Applying formulas (6) and (7), we find the distances x and $k—x$ for each.

We can use formulas (8) or (9) for rolling resistance and we shall assume that this is such as to be equal to one-quarter of the drawbar pull, or 376 and 675 lb. respectively; hence, formula (9) applies and we have 119

TABLE 8—COMPARISON REGARDING SAFETY AGAINST OVERTURNING

	Tractor A	Tractor B
Weight, lb.		
Tractor	2,710	5,708
Operator	150	150
Drawbar Pull (Vertical Reaction= D sin B)	280	484
Total	3,140	6,342
D, lb	1,428	2,700
B	11 deg. 19 min.	10 deg. 18 min.
sin B	0.1962	0.1788

	Basis, Deg. (Tractor A)					Basis, Deg. (Tractor B)					
Lifting Effect, lb., due to	Level	6	12	18	24	Level	6	12	18	24	30
Slope	0	126	250	372	490	0	217	432	642	846	1,040
R R (Rolling Resistance)	119	119	119	119	119	198	198	198	198	198	198
Drawbar	384	384	384	384	384	540	540	540	540	540	540
Operator	12	22	33	44	56	49	58	67	78	90	104
Total	515	651	786	919	1,049	787	1,013	1,237	1,458	1,674	1,882
Net Weight, lb.											
Front	560	424	289	156	26	1,103	877	653	432	216	8
Rear	2,580	2,716	2,851	2,984	3,116	5,239	5,465	5,689	5,910	6,126	6,334

lb. and 198 lb. as the respective lifts on the front wheels of the two tractors due to the rolling resistance being overcome. Formulas (10) and (11) are used for the drawbar lift, and formulas (14) and (15) for the changes due to slope. The man-in-the-seat data are obtained from the formula.

$$w_4 = 150 \text{ lb. } (p + q \sin y/k) \qquad (20)$$

Then the net weight on the ground under the front wheel equals the original weight less the sum of the RR or rolling-resistance lift, drawbar-pull lift, slope lift and man lift. These weights are shown with respect to slopes in Table 8 and also the relative inch-pounds of advantage at the various slopes. However, to get the real relations we should solve the data with regard to the safety against overturn per 1000-lb. drawbar pull.

TABLE 9—DRAWBAR HORSEPOWER LOSSES DUE TO WEIGHT-LIFT UP-SLOPE

	Basis, deg.				
Loss Due to Weight-Lift, hp.	Level	6	12	18	24
Tractor *A*	0	2.28	4.54	6.74	8.78
Tractor *B*	0	4.18	8.32	12.37	16.13
Drawbar Pull, hp.					
Tractor *A*	8.27	5.99	3.73	1.53	—0.51
Tractor *B*	15.63	11.45	7.31	3.26	—0.50

Fig. 16 gives these values in definite figures on a comparable basis. As a matter of fact, neither of these two outfits could negotiate the angles calculated, as we have assumed the drawbar pull near to its maximum when operated on the level; but, by assuming the power developed to be sufficient to carry it through, we can see the relations.

To get the true relations, then, we must limit the discussion to those angles up to which the tractor can pull its rated load, or to two and three plows respectively. Assuming that this machine is working in average light soil of 450 lb. per 14-in. bottom, at 2.17 m.p.h., we have drawbar pulls of 900 and 1350 lb. respectively as minima usable loads for plowing. Checking off the horsepower required to negotiate the different angles of slope, with the weights of tractor, plow and man, at the constant rate of speed chosen, we obtain the values shown in Table 9.

From this we can interpolate and find that these units can negotiate a slope of about 8 deg. for Tractor *A* and 11.25 deg. for Tractor *B*, when pulling their rated plow loads. However, with continually reduced loads these units will be able to climb increased slopes until the tractors have no available drawbar pull. This maximum angle is 20 deg. for Tractor *A* and 24 deg. for Tractor *B*. The net load under the front wheel, however, persists in being positive until 24 deg. for Tractor *A* and until 30 deg. for Tractor *B*. Fig. 17 illustrates the actual ratios in percentages and per 1000-lb. drawbar pull.

Formulas (16) and (17) apply not only to hill slopes but likewise to tractors operated in the furrow. When thus operated, the average tractor tilts out of plumb 6.5 deg. for two-plow units; 6 deg. for three-plow and 5.5 deg. for four-plow units. When in action, this tilting throws a heavier load on the furrow wheels than normally is imagined. Applying formulas (16) and (17) to the two outfits operating on level ground, but having one wheel in the furrow, we have the values shown in Table 10.

While the above analysis might be refined more closely and correctly, the main features developed will clear up and segregate the main influences affecting the several actions and reactions the net sum of which at any instant can be ascertained approximately by the simple formulas developed. We believe that the solutions to the problems propounded can now be determined readily for every case involving static relations, rolling resistance, distribution of weights, balance, choice of hitch and position of operation, whether on land or in the furrow; also, that the analysis presented gives us a definite method of attack for the more correct solution of the proper hitching-point, as well as being a study relating to lug design.

TABLE 10—COMPARISON ON LEVEL GROUND WITH ONE WHEEL IN THE FURROW

	Tractor *A*	Tractor *B*
Width over Tires, in.	62	56
Values of *CG*, in.	28.0	33.5
Rear Weight, lb.	2,580	5,239
Weight on Furrow Wheel, lb.	1,390	2,949
Weight on Land Wheel, lb.	1,190	2,290
Difference, lb.	200	659

THE DISCUSSION

E. R. WIGGINS:—It was mentioned that various factors remain the same when going up a hill. Would not the weight on the rear wheels be more?

O. B. ZIMMERMAN:—The weights change back according as the center line between the points of wheel contact is divided by the perpendicular dropped from the center of gravity as described in the paper. It is remarkable what a wide range of change there is. For instance, take a small tractor weighing 2710 lb., place a man in the seat who weighs say 150 lb., add to this the other component, which is the down pull of the drawbar, and the total combined weight is then 3140 lb. When on the level and pulling the maximum, according to the Nebraska tests, 2580 lb. of the 3140 lb. is on the rear wheel and only 560 lb. on the front wheel. When the tractor is standing idle, we have 1070 lb. to begin with, which now has dropped to 560 lb. when the load is being pulled. Supposing there were engine power enough to carry the tractor up the 24-deg. slope, the net weight on the front wheel at 24 deg. is only 26 lb. and the weight on the rear wheels 3116 lb. One can see readily that when such an angle is reached it takes very little to turn the tractor over. Consider the heavier tractor, which weighed 5708 lb. according to the Nebraska tests. The man weighed 150 lb. Adding the down pull of the drawbar gives a total of 6342 lb. When we start on the level we have 1103 lb. on the front and 5239 lb. on the rear wheels; that is, in action. When we get up to the 30-deg. slope, we have an 8-lb. hold down on the front and 6334 lb. on the rear wheels. When we divide that rear-wheel weight according to the right and the left wheels and according to the slope that there may be on the hillside, we are aware that we have some tremendous loads under each wheel. The interesting point there in connection with lugs is that the drawbar pull backward is small as compared with the push downward underneath the wheels; so the lugs do not need to be so long as many manufacturers are making them.

G. D. JONES:—In regard to the plow test, what form of hitch did Mr. Zimmerman use in making the tests?

MR. ZIMMERMAN:—They were all made with the same hitch; but, at that time, we did not realize that the differences in hitching were so great between one position and another.

MR. JONES:—Did you have a hitch on the order of the old A-type?

MR. ZIMMERMAN:—We used the A-type hitch or a regular solid drawbar.

MR. JONES:—Were no tests made of various forms of hitch?

MR. ZIMMERMAN:—No. That is one of the problems that either agricultural engineers or the Society should study.

MR. JONES:—We experimented some 2 years on hitches, and found decided changes with different types of hitch. I am wondering what reaction one gets at various changes.

MR. ZIMMERMAN:—One would need to work out a theoretical hitch and work on that as a basis. There is much to be developed on this subject. This analysis has been based on practical experience and on definite deductions which for the particular moment were correct, but we still do not know the cross-furrow reaction, styled *R*, as represented for various peaks.

PROF. J. B. DAVIDSON:—I take it that the horse-drawn plow is better designed, because the cross-furrow reaction, styled *R*, is carried on the plow carriage and results in a lighter draft; in other words, the tractor plow is not scientifically designed.

MR. ZIMMERMAN:—That one wheel in the corner of the furrow takes the thrust and we thus change the thrust from a sliding friction, which we have to a large extent in the tractor plow, to a rolling friction. When we get into large units, such as 3, 4, 5, 8 or 12 bottoms, the furrow wall would not take the thrust component; so, we must set the tractor over to take care of the draft. But if we do pull on the slant in that way, according to the plow friction, we take it up in thrust on the engine bearings unless we provide for rolling friction there by using roller bearings or something of that kind.

PROFESSOR DAVIDSON:—I am glad Mr. Zimmerman mentioned that because it does not seem right to carry the side-thrust on a land-side with an increase in the resistance due to friction. We can demonstrate very easily with a horse-drawn plow and a dynamometer that this may mean about a 25-per cent increase in the draft.

I think it is a matter of record that some of the accidents due to tractors overturning have happened when the tractors were detached from their loads. This is explained by the fact that the tractor revolves about the rear axle when the drive-wheels become fixed. If the load is attached and the hitch is below the axle, the tractor could not very well turn over because, so long as the center of moments coincides with the center of the axle, the drawbar pull tends to keep the front end of the tractor down.

CAMBER AND GATHER RELATIONSHIPS

(*Concluded from page* 92)

TABLE 4—GATHER FOR DIFFERENT WHEEL DIAMETERS AND AREAS OF CONTACT

D, Wheel Diameter, In.	L, In.				
	6	7	8	9	10
30	0.180	0.210	0.240	0.270	0.300
32	0.192	0.214	0.256	0.288	0.320
34	0.204	0.238	0.272	0.306	0.340
36	0.216	0.252	0.288	0.324	0.360
38	0.228	0.266	0.304	0.342	0.380
40	0.240	0.280	0.320	0.360	0.400
42	0.252	0.294	0.336	0.378	0.420

from which we can prepare Table 4, which gives values of gather in inches.

CORRECT GATHER FOR FRONT WHEELS

Table 4 can be summarized by the following rule: The correct gather for front wheels, *G*, is obtained by multiplying the diameter of the tire in inches by the length of the area of contact in inches and dividing by 1000; that is, $G = DL/1000$.

The procedure in making use of Table 4 is to see first that the air pressure and load are normal, and that the front wheels rest upon a smooth flat surface; then to measure the length of the area of contact. The length of the area of contact can be determined accurately by first jacking up the wheel and then lowering it carefully upon a piece of white paper covered with a piece of carbon paper or multigraph ribbon. When the wheel is lifted again, a print of the area of contact will be found on the white paper, from which the length *L*, in inches, can be obtained. If, for lack of equipment, the above instructions cannot be followed, an approximation of the lengths of the areas of contact can be taken from Table 5, which gives values of *L* for various sizes of pneumatic tires under S.A.E. Standard loads and inflation-pressures. Second, jack up both front wheels and turn them, scribing a fine line near the center of the tread, as indicated in Fig. 4. Third, refer to Table 4. For example, suppose that the tire is 32 in. in diameter and the length of the area of contact is 8 in. A glance at Table 4 shows that the gather should be 0.256 in. or slightly more than ¼ in.; that is, the distance between the tread lines should be about ¼ in. greater behind than in front. Scribing the lines on the treads eliminates any error due to wobbling of the wheel, but the bearings and the bushings should be adjusted closely.

It should be understood clearly that the above method applies only to the average case in which *S* equals approximately 500 in. This is sufficiently accurate in nearly all cases. There are, however, a few cases in which the camber differs noticeably from the average and, therefore, *S* is noticeably greater or less than 500 in. For these special cases the formulas, rather than the tables, should be used.

The entire operation of checking the front-wheel alignment of a vehicle requires very little time and practically no expense. It is worth any owner's time to construct a suitable "tram" which can be made easily of a few pieces of wood, as indicated in Fig. 4. Public garages should be equipped with a tram for this purpose. The life of a front tire may be reduced many thousands of miles through a misalignment of ⅜ in.

TABLE 5—APPROXIMATE LENGTHS, *L*, OF AREAS OF CONTACT

Fabric Tire Size, In.	Cord Tire Size, In.	S.A.E. Standard: Air Pressure, Lb. Per Sq. In.	S.A.E. Standard: Load, Lb.	L, Length of Area of Contact, In.
30x3	....	45	375	6.3
30x3½	....	55	570	6.6
....	30x3½	50	600	6.5
....	32x3½	50	600	6.4
....	31x4	60	850	6.9
32x4	....	65	815	7.0
....	32x4	60	850	7.0
....	33x4	60	850	6.8
....	32x4½	70	1,200	7.6
....	34x4½	70	1,200	7.6
....	35x5	80	1,700	8.6
....	36x6	90	2,200	9.0

Current Standardization Work

ALTHOUGH the work of the various Divisions during the last 2 months was concentrated on those recommendations that were presented for approval at the Standards Committee Meeting on June 20, many subjects have received consideration which indicates that the reports to be presented at the Standards Committee Meeting in January 1923 will require much time and effort during the last half of 1922.

Eleven Division and Subdivision meetings were held during May and June as shown in the accompanying table. No Division meetings have been scheduled for the summer months, pending a review of the work now in progress and the formulation of sufficient Subdivision reports to warrant calling the Division members together.

The discussion which was presented at the Standards Committee meeting on June 20 and an account of the official action taken on the various reports will be printed in full in the August issue of THE JOURNAL. A brief summary of the action taken, however, is printed in this issue in the account of the Summer Meeting beginning on p. 1.

Axle and Wheels Division	May 2
Electric Vehicle Division	May 10
Lighting Division	May 3, June 2
Motorboat Lighting Subdivision	May 2
Non-Ferrous Metals Division	May 1
Parts and Fittings Division	May 9
Passenger Car Division	May 2
Passenger-Car Hubs Subdivision	April 17
Screw-Threads Division	May 1
Springs Division	May 5
Storage-Battery Division	May 5

BALL BEARINGS

A series of meetings was held on April 27 and 28 at which the subject of international standardization of ball bearings was carefully considered and tentative proposals adopted which have been submitted to England, Sweden, Germany and other European countries.

The Committee on Information, appointed by the American Sectional Committee on Ball Bearings, met on the morning of April 27 to consider the report of O. R. Wikander, who had just returned from a trip abroad, covering results of his discussions with European ball-bearing representatives as to international standardization. The Subdivision reported to the Sectional Committee on the afternoon of April 27, its report being approved as submitted and informally referred to the ball-bearing committees of the American Society of Mechanical Engineers and the Society of Automotive Engineers, the sponsor bodies for the Sectional Committee, for consideration on the morning of April 28. The Division of the Society's Standards Committee and the committee of the American Society of Mechanical Engineers approved the corner radii proposed by the German Ball Bearing Committee and favored the other proposals as submitted by the Sectional Committee which are given hereinafter. The proposals were referred back to the Sectional Committee in the afternoon of April 28 and definite proposals were prepared which were subsequently submitted, as stated, to other countries interested in establishing international ball-bearing standards. The proposals of the American Sectional Committee on Ball Bearings are as follows:

(1) That the outside diameters proposed by the German Ball Bearing Committee be accepted for the Light, Medium and Heavy Series

(2) That the widths of the bearings covered by the above Series be retained for those sizes for which International uniformity exists at the present time, that is, up to and including 110-mm. bores for the Light Series, 95-mm. bores for the Medium Series and 85-mm. bores for the Heavy Series; but that above these sizes, the width of each bearing shall be approximately 80 per cent of the radial difference between the bore and the outside diameter, which conforms closely to Swedish practice

(3) That if an Extra-Light Series, which is not favored, should be adopted, the widths of such bearings should be made approximately 80 per cent of the radial difference between the outside diameters and the bores

(4) That the corner radii proposed by the German Ball Bearing Committee be adopted

(5) That the width tolerances adopted by the Society of Automotive Engineers be adopted instead of the very close German width tolerances

The German tolerances for bores and outside diameters correspond so closely to the tolerances adopted by the Society of Automotive Engineers that they are identical for all practical purposes.

No further action will be taken by the American Sectional Committee on Ball Bearings until information is received from abroad as to whether these proposals are satisfactory and, if not, what revised proposals would meet with approval.

BASES, SOCKETS AND CONNECTORS

At the meeting of the Lighting Division on May 3 a Subdivision was appointed to investigate thoroughly the present S. A. E. Standard for Bases, Sockets and Connectors, p. B4 of the S. A. E. HANDBOOK, and if advisable to recommend revisions to meet the objections that have been submitted against this standard by motor-truck engineers. The Subdivision will also consider the standardization of this type of equipment for motorboats. The personnel of the Subdivision appointed is:

C. E. Godley, *Chairman*	Edmunds & Jones Corporation
A. K. Brumbaugh	Autocar Co.
J. T. Caldwell	National Lamp Works
B. H. Kenyon	Providence Base Works
J. C. Stearns	Culver-Stearns Mfg. Co.
G. A. Walters	Chicago Electric Mfg. Co.
Ernest Wooler	Cleveland Automobile Co.

BESSEMER STEELS

As a result of a request submitted by the International Harvester Co., the addition of bessemer steels to the S. A. E. Steel Specifications was considered at the last meeting of the Iron and Steel Division. It was pointed out that many users feel that including bessemer steel specifications in the S. A. E. Standard would not be advancing the art, but it was thought that separate specifications might be desirable to meet requirements for agricultural implements. It was considered advisable, however, to obtain definite evidence supporting the need of bessemer steel specifications before taking final action.

After general discussion, it was decided to reserve the 1400 series for bessemer steels with the understanding that this would not commit the Society to formal approval of such compositions or their uses, and to recommend that the implement manufacturers cooperate with each other toward deciding upon a limited series of bessemer steels, in which work the Iron and Steel Division will assist as an advisory body.

COTTER-PINS

At the meeting of the Parts and Fittings Division on May 9 it was stated that trouble had been experienced in obtaining S. A. E. Standard cotter-pins due to the drill numbers for the holes as given in the present S. A. E. Standard, p. C7 of the S. A. E. HANDBOOK, being too large for the manufacturers' standard sizes of cotter-pin. Cotter-pin

wire sizes are determined from the drill size for the holes in which they are to be used rather than from the actual diameter of the wire. Thus, according to the present S. A. E. Standard, a purchaser would have to buy oversize cotter-pins to fit S. A. E. Standard holes. It was suggested that the present standard should be revised by eliminating the 7/64 and 9/64-in. cotter-pin sizes and by changing the 11/64-in. size to 5/32 in. and the 13/64-in. size to 3/16 in. and by inserting the actual diameter of the wires and decreasing the drill number for the holes. This led to discussion as to how tight a cotter-pin fit is desired, but no definite recommendation was agreed upon.

The subject is to receive further consideration, particularly in regard to what effect changes such as suggested would have on other S. A. E. Standards.

Bolt Heads and Rivets

As the Sectional Committee on Bolt, Nut and Rivet Proportions, sponsored by the American Society of Mechanical Engineers and the Society, is working on the standardization of bolt heads and rivets because they are of general interest to other than the automotive industry, it has been decided to hold the standardization of these subjects by the Screw-Threads Division in abeyance.

Engine Mufflers

The reasons in support of the negative votes cast by Society members against the adoption of the S. A. E. Standard for Mufflers, p. A13 of the S. A. E. Handbook, were reviewed at the last meeting of the Engine Division. It was thought that no revisions should be made in the present standard until it shall have been applied in actual practice and further information obtained as to its practicability.

Engine Support Arms

At the meeting of the Engine Division on April 17 the further standardization of engine support arms was discussed with special reference to sub-frame construction. It was considered inadvisable to standardize on a sub-frame construction unless there shall be a clear demand for such action. The matter is to be referred to engine and motor-truck manufacturers for comment.

Felt Specifications

Criticisms of the proposed felt specifications, resulting from circularizing the tentative report of the Subdivision on Felt, published on p. 435 of the May issue of The Journal, were discussed at the Parts and Fittings Division meeting held on May 9. It was the thought that these criticisms should be sent to the members of the Subdivision for further study and that laboratory tests should be specified in the recommendation. Reference was made to the more or less satisfactory results of tests that have been developed and enforced by a number of automobile companies and it was felt that the Subdivision should obtain as much information as possible regarding these tests.

Head-Lamp Brackets

At a meeting of the Lighting Division on May 3 the suggestion that the present S. A. E. Standard for Fork-Type Head-Lamp Brackets should be cancelled was discussed. It was brought out that this type of head-lamp mounting is still used to a considerable extent on motor trucks and motorcycles and that the present standard should be retained as a guide to passenger-car builders using it.

Fuel and Lubrication Pipe Fittings

At the meeting of the Parts and Fittings Division on May 9 it was stated that the present S. A. E. Recommended Practice for Flared-Tube Type of Fuel and Lubrication Pipe Fittings, p. C46 of the S. A. E. Handbook, was being largely replaced by the compression type of coupling, particularly on truck and tractor installations, owing to the severe vibration. It was decided, however, to retain this standard temporarily at least.

As it was thought that the Division should standardize a series of compression-type couplings, a Subdivision was appointed to prepare a report that will include a list of sizes of compression-type coupling which will be submitted to the industries for comment.

The present S. A. E. Recommended Practice for Soldered-Type Couplings, p. C47 of the S. A. E. Handbook, was reviewed. As this specification is used principally by motorcycle manufacturers and has evidently proved satisfactory, it was decided to retain it without revision.

Poppet Valves

A Subdivision has been appointed to extend the present S. A. E. Standard for Poppet Valves, p. A3 of the S. A. E. Handbook, so as to include the dimensions for the valve head. The present standard specifies only the port diameter and the corresponding valve and stem diameters.

Passenger-Car Front-Axle Hubs

At the Division meeting held on May 2 five sizes of passenger-car front-axle hub were studied in connection with ball and roller-bearing layouts that had been submitted. The ball-bearing layout included a new series of inch dimension ball bearings that have the same bores as the proposed roller-bearing series, as it is thought that the ball-bearing manufacturers will be willing to develop such a series. It was decided that the recommendation should include the spindle lock-nuts and washers and hub-cap threads.

To facilitate the work of the Subdivisions, a ball-bearing as well as a roller-bearing subcommittee was appointed with axle, wheel and passenger-car representation on each. The subcommittees are to prepare jointly a single proposal for ball and roller-bearing applications, this to be reported to the Subdivision so that flange diameters, spoke widths and flange-bolt dimensions may be added. The proposal will then be circularized among the passenger-car and parts manufacturers for comment.

BALL-BEARING SUBCOMMITTEE PERSONNEL

F. W. Gurney, *Chairman*	Gurney Ball Bearing Co.
R. S. Begg	Jordan Motor Car Co.
H. E. Brunner	S. K. F. Industries, Inc.
E. R. Carter, Jr.	Fafnir Bearing Co.
L. A. Cummings	Standard Steel & Bearings, Inc.
C. S. Dahlquist	Eaton Axle Co.
A. M. Dean	Templar Motors Co.
F. G. Hughes	New Departure Mfg. Co.
A. M. Laycock	Sheldon Axle & Spring Co.
A. L. Putnam	Detroit Pressed Steel Co.
O. J. Rohde	Wire Wheel Corporation of America
L. M. Stellman	H. H. Franklin Mfg. Co.
H. Vanderbeek	Timken Detroit Axle Co.
Andrew S. VanHalteren	Motor Wheel Corporation

ROLLER-BEARING SUBCOMMITTEE PERSONNEL

T. V. Buckwalter, *Chairman*	Timken Roller Bearing Co.
R. S. Begg	Jordan Motor Car Co.
L. W. Close	Bock Bearing Co.
C. S. Dahlquist	Eaton Axle Co.
A. M. Dean	Templar Motors Co.
G. W. Dunham	Savage Arms Corporation
C. T. Hagenlocker	Wright Roller Bearing Co.
A. M. Laycock	Sheldon Axle & Spring Co.
A. L. Putnam	Detroit Pressed Steel Co.
O. J. Rohde	Wire Wheel Corporation of America
R. G. Schaffner	Bower Roller Bearing Co.
L. M. Stellman	H. H. Franklin Mfg. Co.
H. Vanderbeek	Timken-Detroit Axle Co.
Andrew S. VanHalteren	Motor Wheel Corporation

Storage-Battery Monoblock Containers

Tentative specifications for storage-battery monoblock containers were approved at the meeting of the Storage-Battery Division on May 5. These specifications are intended as an extension of the present S. A. E. Standard for Storage

(*Concluded on page* 124)

THE SUMMER MEETING

(*Concluded from page* 8)

J. G. Vincent supplemented Mr. Crane's remarks with a discussion of the relative merits of the three types of engine construction: L-head, I-head and overhead-camshaft. Experiments had been made by the Packard company on all three types. In the overhead-camshaft type the lack of accessibility had been the principal disadvantage. From the practical point of view he considered that the choice lay between the L-head and the pushrod-operated overhead-valve engine. Each of these types has its advantages, and both are equally quiet. He questioned whether it is worth while to put valves in the head to get better volumetric efficiency. He recommended that particular attention be paid to carburetion. In conclusion, he remarked that the overhead-camshaft engine cannot be produced in quantities because of its high cost and of the difficulty of securing quiet operation.

F. S. Duesenberg called attention to the design of an overhead-camshaft engine constructed so that the cylinder-head can be removed without interfering with the timing of the valves. R. E. Fielder spoke of the merits of the sleeve-valve engine. In concluding the discussion of the relative merits of the different types of engine construction, A. L. Nelson stated his preference for the valve-in-head engine, while Mr. Heldt summed up by predicting that the overhead-camshaft type will be confined to the higher-priced cars.

H. M. Crane then read his paper on a New System of Spring-Suspension for Automotive Vehicles. He indicated what the history of spring-suspension has been, but discussed only the conventional type of four-wheeled design, in which the front wheels are used for steering and the rear wheels for driving and braking. The problem of front-axle suspension is mentioned in the paper but that of proper rear-axle spring-suspension, especially for passenger cars, is discussed in detail because it is a much more difficult one.

Mr. Crane mentioned the advantages of the Hotchkiss drive for shaft-driven cars and some of its disadvantages.

W. C. Keys inquired whether Mr. Crane had made an investigation of the characteristics of cars using full-elliptic springs. Mr. Cravens made an allusion to the types of spring-suspension used in the Lanchester, the Lafayette and a number of other cars. Mr. Crane was of the opinion that full-elliptic springs do not afford sufficient resistance to torque reaction. The design he had adopted permitted the amount of longitudinal rigidity to be regulated. He emphasized the importance of rigidity against thrust and brake action.

In reply to Mr. Keys' inquiry as to the type of universal-joint used, Mr. Crane said he used a rigid universal-joint at the front end, while the rear-end joint was of the type in which rollers travel in a slotted member. This type of universal-joint is characterized by unlimited angularity together with great freedom of longitudinal movement. Mr. Crane referred to the popularity of the torque-arm drive and said that the chief objection to it is its great rigidity in absorbing torque reactions. He expressed the opinion that the use of cantilever springs necessitates the torque-arm construction. He was in favor of this type of construction for smooth roads. He said that another objection to the cantilever type of spring is the difficulty of supporting bodies owing to the overhang at the rear of the car. He emphasized the difficulty of making the rear frame strong enough to carry the rear weight, particularly in closed-body designs. He laid particular stress on the importance of frame stiffness rather than frame strength.

F. A. Bonham read his paper, The Automotive Engineer and Our Service Problem. He touched upon the work of the National Automobile Chamber of Commerce Service Committee and attributed the present condition to a misunderstanding of the service problem by the automotive industry generally. Mechanics were inadequately trained and were not equipped with the proper tools. To alleviate the present unsatisfactory state of affairs, he suggested that parts be made interchangeable as far as possible, that flat rates be charged for repairs, and that the necessity for special tools for certain makes of car be eliminated by standardization. He mentioned as the chief requirements, accessibility and simplicity and uniformity in the design of units and wearable parts. He particularly urged that the necessity for special and elaborate tools be eliminated, because the average service-station has not the time or the money or the space for them.

Mr. Bonham stated that the Service Committee is now running a series of tests on the road to determine which parts wear out soonest, and how long it takes to replace various parts; which parts are most prone to give trouble when there is divergence in manufacturing specifications or when in the hands of careless mechanics. The service problem is stated to be not one of design or of assembly and construction but rather of incompetence and ignorance on the part of the average repair-shop mechanic.

In discussing the points raised, T. J. Litle, Jr., stressed the necessity for frequent conferences between the engineering and the service departments of the manufacturing companies. C. M. Manly thought that the problem is largely one of management and that complaints should be carefully investigated and cleared up. He also emphasized the necessity of cooperation between the different departments.

Mr. Bachman alluded to the matter of deterioration. He stated that the service end needs competent men who can give service and sales men a rational idea of what problems are of the most pressing importance.

Prof. W. K. Hatt presented an abstract of his paper on Highways. He described the organization of the Highway Research program that is now in progress under the auspices of the National Research Council, and mentioned the various bodies that are participating in its work throughout the Country. Slides were shown in setting forth the organization of the project, and illustrations were given of the equipment used and the methods of attacking the problem.

T. V. Buckwalter opened the discussion by stating that the kernel of the highway problem lies in banking the curves and cambering the tangents. H. W. Alden brought out a very interesting point when he said that a bad road is as bad for the road as for the vehicle; in other words, that the stresses of a vehicle on a bad road are as destructive to the road as to the vehicle. Professor Hatt agreed with this view and stated that in considering the highway problem we must evaluate the road in terms of transportation. He remarked that all roads im-

prove with age, because they settle. A new road is almost invariably a bad road.

Professor Hatt made a plea for the cooperation of automotive engineers in dealing with the problems that best the highway engineers. Primarily, the building and maintenance of good roads is not the concern of the automotive engineer, but it is very much to his interest to see that the Country is provided with a network of good roads, scientifically constructed so as to inflict the minimum wear or stress on automobiles and trucks. The abuse of roads by any one class of vehicles inevitably causes hardship to another class. Nothing can be achieved without the cooperation of automobile engineers and the National Research Council is doing its part by placing in their hands all the necessary information on the tractive resistance of roads to various types of vehicle, and on the relative efficiency and wearing qualities of different types of road.

THE SPORTS PROGRAM

With unusually comprehensive and well-maintained sports facilities at White Sulphur Springs, it was anticipated that the sports program would be one of the outstanding features of the meeting. There were tournaments in baseball, golf, tennis and trapshooting, in addition to the aquatic meet and the field day. Inter-Section rivalry was always in evidence and added greatly to the interest in the program. A very fine collection of prizes rewarded the victors in the many contests, as well as a large percentage of the also-rans. These handsome prizes were made available through the generosity of over 200 firms in the industry who contributed to the prize fund.

GOLF TOURNAMENTS

Golf continues to be the most popular sport among the automotive engineering fraternity, judging from the number of participants in the annual Society tournament. Well over 100 followers of the Scottish pastime answered the call of Chairman Frank Lawrence on Tuesday and played their qualifying rounds. The contestants were grouped in three flights after the qualifying round, the subdivision being based on the qualifying scores for 18 holes. Match play continued in each of these flights until Friday afternoon, when the finals were played. In the Society Championship flight, Claude Foster was opposed by John Warren Watson, the latter winning the match and receiving the medal emblematic of the 1922 Championship. Mr. Watson having won the same honor at West Baden in 1921, it seems proper to suggest that those planning to attend next summer's meeting get plenty of coaching and come prepared to terminate this monopoly. The second-flight prize was taken by Sanford Brown, who battled C. S. Pelton in the finals. D. L. Gallup won the third-flight final match from W. Ray. The driving contest was very close, Jack Gray winning with a drive that was only 2 yd. ahead of Sanford Brown's. C. H. Foster was third. The golf putting contest was won by E. O. Jones, R. A. Watson finishing second and George Case third.

The ladies golf events attracted a much larger field than in years past. Mrs. George Case, who won, found her laurels none too easy to attain. Mrs. J. B. Funk was second in the ladies' golf tournament, with Mrs. C. H. Foster placed third. Interest was added to the ladies putting contest by the opening of a typical pari-mutuel betting establishment where wagers could be placed on one's favorite. As usual, the favorites fell by the boards and Mrs. C. H. Foster won with a comfortable margin from Mrs. George Case and Mrs. J. B. Funk, who were second and third respectively. The clock-golf contest resulted in a triumph for Mrs. Jack Gray.

TENNIS TOURNAMENTS CLOSELY CONTESTED

The lovers of tennis who attended the Summer Meeting were rewarded with the opportunity to play on exceptionally fine courts and amidst keen competition. The champion of last year, C. F. Clarkson, met his Waterloo in the person of C. A. Thompson. The two sets of this match were hotly contested and drew a large and appreciative gallery. Mr. Thompson was returned victor, 11-9, 6-4. The doubles match was equally close and interesting, requiring five sets and presenting many exciting rallies. Herbert Chase and Walter Buettner won out over H. M. Crane and C. F. Clarkson, 10-8, 6-3, 1-6, 4-6, 6-3. The ladies tennis championship was won by Mrs. Snead and the mixed doubles were captured by Miss Jessie McKenzie paired with C. A. Thompson.

Lon R. Smith and his enthusiastic clay-bird marksmen enjoyed 4 days of sharpshooting. It was understood that the winner on Friday was to be declared Society Champion and that he would receive the championship medal. W. S. Harley carried off this honor, with W. H. Miller close on the trail. The prize shooters the other three days were W. H. Miller, George Duck and R. M. Owen.

SWIMMING EVENTS AROUSE INTEREST

One of the most entertaining features of the sports program, and the one attracting the largest audience, was the swimming meet in the large pool on Wednesday evening. An exceptionally large field of entrants participated in the events and in most instances the races were closely contested. Gordon Brown proved to be very accomplished subaquatically and garnered unto himself the lion's share of the spoils. Many of the stunt races presented amusing competitions, George Briggs particularly surprising the gallery with his brilliant water-golf victory. The ladies came into the spotlight in several races and naturally added color to the program. The complete summary of the swimming races follows:

Swimming Events

Inter-Section Relay—First, Metropolitan; second, Detroit

33-Yd. Swim (Men)—First, W. H. Miller; second, W. S. Davidson; third, Neil MacCoull

33-Yd. Swim (Men over 40)—First, Rollin Abell; second, W. S. Harley

33-Yd. Swim (Ladies)—First, Mrs. Ernest Dickey; second, Miss Catherine Cramer

66-Yd. Swim (Men)—First, Gordon Brown; second, W. H. Miller; third, W. S. Davidson

66-Yd. Swim (Ladies)—First, Mrs. J. F. Winchester; second, Miss Anne Koch

Section Balloon Relay—First, Detroit; second, Cleveland

Plunge for Distance—First, T. A. Peck; second, L. C. Hill; third, A. K. Brumbaugh

Egg Race (Men)—First, Neil MacCoull; second, Gordon Brown

Candle Race (Ladies)—First, Mrs. E. Dickey

Blindfold Race (Men)—First, Gordon Brown; second, Mason Rumney; third, F. A. Thompson

Diving Contest (Men)—First, Harold Butcher; second, Neil MacCoull; third, Gordon Brown

Plate Diving Contest (Men)—First, A. C. Bigelow; second, Gordon Brown

Nightshirt Race (Men)—First, Gordon Brown; second, T. A. Peck; third, A. C. Bigelow

Water Golf Race (Men)—First, George Briggs; second, Neil MacCoull; third, F. A. Thompson

ANNUAL FIELD DAY A FEATURE

Led by a typical Darktown Alexander's Ragtime Band, the whole assemblage marched to the big field meet on Thursday afternoon and enjoyed the antics of our versa-

tile automotivian athletes. There were jumps, runs, stunts and what-nots to enable the ambitious ones to unkink muscles long since labeled dormant. No casualties were recorded but limps and charley-horses were quite the fashion on Friday. Of course H. E. Kirby pushed the shot many feet beyond his nearest rival, as is his custom; Norma Porter tossed the pellet of national pastime yards beyond any other miss; and Burt Brodt proved that he is still master of the sprints and jumps. The great surprise of the field day was the unheralded speed of Indiana's youthful relay team, which was assembled and groomed by George Briggs. Here are the summaries

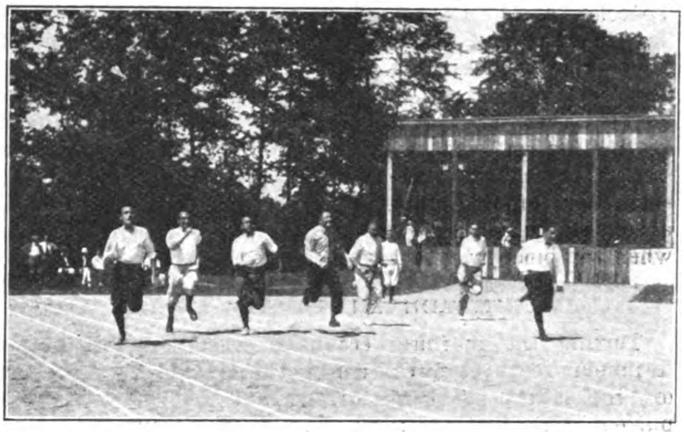
Track and Field Meet

50-Yd. Dash (Men under 30)—First, B. W. Brodt; second, M. P. Whitney; third, E. O. Jones
50-Yd. Dash (Men 30 to 40)—First, Neil McMillan, Jr.; second, B. S. Pfeiffer; third, W. F. Rockwell
50-Yd. Dash (Men over 40)—First, Mark Smith; second, W. S. Harley; third, L. W. Rosenthal
50-Yd. Dash (Boys under 12)—First, Bobby Germaine; second, Danny Duesenberg; third, Warren Elliot
50-Yd. Dash (Boys under 16)—First, Robert Jardine; second, Gould Klinedinst; third, Danny Duesenberg
50-Yd. Dash (Ladies)—First, Mrs. Ernest Dickey; second, Miss Anne Koch; third, Mrs. Hauser
Fat Man's Race—First, H. L. Williams; second, T. V. Buckwalter; third, G. E. Strohm
Three-Legged Race (Men)—First, Neil McMillan, Jr., and E. O. Jones, second, W. F. Rockwell and B. S. Pfeiffer
Potato Race (Men)—First, E. O. Jones; second, Neil McMillan, Jr.; third, M. L. Hull
Potato Race (Ladies)—First, Mrs. E. Dickey; second, Mrs. Harry Tarantous; third, Miss J. McCormick
One-Legged Race—First, E. P. Warner; second, W. F. Rockwell; third, B. S. Pfeiffer
Three-Legged Race (Mixed)—First Miss Anne Koch and Mr. W. F. Rockwell; second, Miss J. McCormick and Mr. B. S. Pfeiffer
Shot Put—First, H. E. Kirby; second, T. V. Buckwalter; third, I. S. Snead
Standing Broad Jump (Men under 40)—First, B. W. Brodt; second, F. G. Whittington; third, F. F. Kishline
Standing Broad Jump (Men over 40)—First, W. S. Harley; second, J. G. Vincent; third, T. V. Buckwalter
Hop, Skip and Jump—First, B. W. Brodt; second, W. L. Batt; third, W. S. Davidson
High Jump—First, B. W. Brodt; second, M. P. Whitney; third, F. G. Whittington
Throwing Baseball (Ladies)—First, Miss Norma Porter; second, Mrs. Beegle; third, Miss Ruth Porter
Egg Race (Ladies)—First, Miss J. McCormick; second, Mrs. George Case; third, Mrs. Metz.
Inter-Section Relay Race—First, Indiana, Mark Smith, George Briggs, F. A. Clawsen, M. L. Hull; second, Detroit; third, Metropolitan

The Inter-Section baseball series had many followers and was directly responsible for numerous frozen voices among the ardent rooters. Cleveland defeated Metropolitan on Wednesday 19 to 3, this game being mistaken several times for a fly-chasing contest. Wild heaves, muffs and boots were the order of the day and the least-worst team triumphed. Air-tight ball featured the Detroit-Indiana game on Thursday, the former Section winning 6-3, largely due to the dependable manner in which E. O. Jones garnered flies in left field. Harry Figgie's Cleveland warriors defeated Detroit in the final on Friday, 10-8, the game being very close until the last man was retired. Figgie will hold the Inter-Section Cup in Cleveland until next year, when the riot will be renewed.

INTER-SECTION CHAMPIONSHIP TO METROPOLITAN

The handsome cup emblematic of the Section Athletic Championship goes to Metropolitan Section for 1922, with a score of 121½ points; Cleveland was second with 118 points; Detroit third with 105 points; and Mid-West



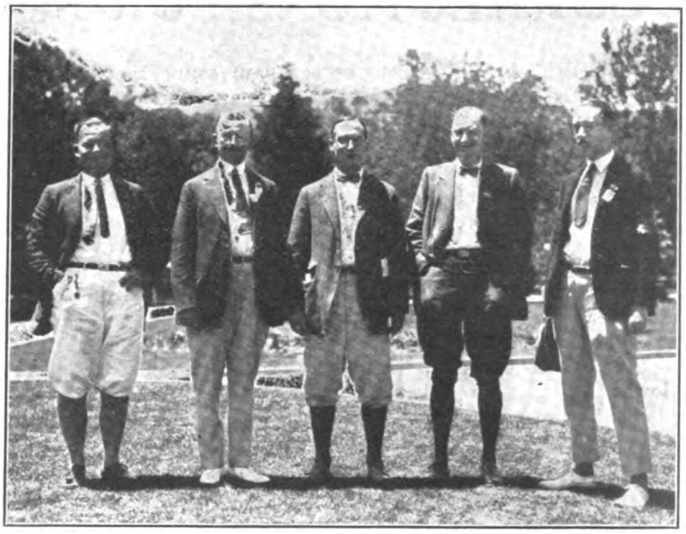
"They're Off"

fourth with 55 points. The Metropolitan points were collected mostly in the swimming meet and the tennis tournament. Cleveland was proficient in baseball, tennis and golf, while Detroit showed to best advantage in the track events. The plan of encouraging friendly rivalry between the Sections was commended by all of the athletes. It unquestionably adds interest to the entire sports program, results in a larger entry list and builds Section morale. The Inter-Section Championship cup will be contested for again next year and without any doubt the various Section leaders will come well fortified to wrest it from the tribe of Manhattan.

Another feature of the scoring plan was the awarding of a handsome prize to the individual all-around athletic champion. Gordon Brown, whose swimming was a pleasure to watch, captured this honor largely because of his prowess in the water. His total score was 24½, Burt Brodt being close behind him with 22½ points.

The Annual Dinner Burlesqued

The Metropolitan Section was responsible for the one big surprise of the meeting. On Thursday evening all of the members dined together and were astonished to find themselves in the midst of a huge burlesque of the 1922 Annual Dinner of the Society in New York City. The printed menus were badly garbled, much liberty being taken with the names of the honored speakers of the evening. Catchy parodies were sung to old familiar tunes and music was provoked by a burlesque song-leader. The



The Meetings Committee

speeches of the evening were patterned after those of last January, but little respect was shown the feelings of those whose names were involved. It was all in good fun, very well worked out and provoked many a laugh. This Section stunt was followed by a wireless concert arranged by the Detroit Section. The demonstration was such a far cry from the real thing that the audience became suspicious of its genuineness in the early stages and absolutely ruined the perpetrator's plans of a grand finale when the hoax was to have been announced.

Entertainment for the Ladies

During the morning technical sessions many entertainment features were arranged to engage the attention of the ladies. Several bridge and 500 parties were played, Mrs. C. C. Carlton, Mrs. F. S. Slocum and Mrs. J. W. Ruzicka being the winners at bridge on different occasions. The 500 winners were Mrs. Ernest Dickey, Mrs. Jack Gray and Mrs. F. G. Whittington. On Wednesday morning an automobile ride was taken by a large party of the ladies through the surrounding country and the Greenbrier Valley. Of course, there were golf, tennis and other sporting events for the gentler sex; these are described in the section of this report covering the sports. Wireless concerts were provided through the courtesy of the Westinghouse Electric & Mfg. Co. Baritone concerts were given each day by Robert Crawford, who was accompanied by Rockwell Ferris at the piano. This highly complimented feature was arranged by the Cleveland Section. Dancing and motion pictures ran simultaneously in the evenings, the dancing program reaching a climax in the Grand Ball on Friday evening with its interesting formal and jazz dancing contests. The judges had great difficulty in picking the most graceful steppers but made popular decisions in choosing Miss Catherine Cramer as the winner of the formal contest and Mrs. J. B. Funk as the best portrayer of the syncopated gyrations of the present era.

Gratitude to the Committees

In closing this report of the 1922 Summer Meeting, credit must be given to those who worked arduously, at no little sacrifice of their own pleasure, to make every feature of the program a success. The Meetings Committee, directed by Carl F. Scott, spent many months preparing the details of the technical, sports and entertainment phases of the meeting. Mason P. Rumney selected the members of his Sports Committee many weeks ahead, held several meetings to discuss the athletic events and perfected an organization of committeemen that guaranteed the smooth running of each contest. The commendable result at White Sulphur Springs reflected the value of this forethought. The following members of the Sports Committee deserve full credit for the success of the athletic program:

Sports Committee

Mason P. Rumney, *Chairman*
Frank Lawrence—Golf
Lon R. Smith—Trapshooting
C. A. Thompson—Tennis
Neil McMillan, Jr.—Baseball
B. W. Brodt—Track
C. H. Brennan—Swimming
Walter C. Keys—Inter-Section Contest

The entertainment program included several innovations along with the enjoyable dances and card parties of past summers. Wide approval of the entertainment features was evidenced throughout the meeting. Credit for this branch of the work is due to M. C. Horine and Orrel A. Parker. Howard Spohn deserves special mention for his labors as official announcer and for the assistance he rendered the Entertainment Committee.

The *Daily S.A.E.*, that frivolous sheet which in jocose vein kept all informed of the meeting news and scandal, was published this year by the Timken Roller Bearing Co. under the direction of R. E. McKenzie. Those responsible for the 1922 editions of the *Daily S.A.E.* deserve particular mention for their news-gathering ability, the good-natured humor of the personal items and the very attractive appearance of the paper.

The 1922 Summer Meeting was distinctly creditable to all those whose efforts accomplished its success. Few summer gatherings of the Society have excited such general voluntary approval on the part of those in attendance. White Sulphur Springs lacks nothing a first-class meeting place should have. There seemed to be a universal sentiment among the members in favor of returning to this resort for the 1923 meeting.

CORRECTIONS FOR SUMMER MEETING PAPERS

Unfortunate errors occurred in connection with two of the Semi-Annual Meeting papers that were printed in the June issue of The Journal. In the paper by Thomas Midgley, Jr., and T. A. Boyd entitled Detonation Characteristics of Some Blended Motor-Fuels, the cuts for Fig. 3 on p. 454 and Fig. 5 on p. 456 were transposed, as would be apparent from the captions. In this same paper on p. 456, an entire line was omitted from the manuscript as submitted by the authors. The following should be inserted between the words "of temperature" in the 11th line of the second column, "the percentage distilled and that of the horizontal axis was in terms of"

The 8th to the 15th lines as corrected should read as follows:

axis. From this curve the percentages of the fuel distilling in each 10-deg. interval were obtained, and these values were plotted on a chart in which the scale of the vertical axis was in terms of the percentage distilled and that of the horizontal axis was in terms of temperature. The average boiling-point of the fuel was taken as the point at which a perpendicular passed through the center of gravity of the area enclosed under this differential distillation-curve cut the horizontal or temperature axis.

The caption for Fig. 2 of the paper by H. M. Crane entitled New System of Spring-Suspension for Automotive Vehicles on p. 464 should have indicated that the spring-suspension was applied to a passenger-car chassis.

The Sections Lunch at the Summer Meeting

THE Sections Luncheon at White Sulphur Springs attracted a very representative group of members interested in the administration of the 11 Sections of the Society. Every Section was represented by at least one of its members and all of the Sections Committee were present. Each of the Section representatives was asked to present a brief summary of his Section's activities during the past year, to outline its plans for the coming fall and winter, and to discuss any problems whose solution seemed essential to the future success of his Section. Section dues, affiliation with local engineering bodies, the relative attractions of technical and non-technical papers and the important matter of securing increased Section membership and larger attendance at meetings, were the major topics of discussion.

A. K. Brumbaugh, chairman of the Sections Committee, presided at the luncheon. He asked the Section officers to place their problems before the meeting in order that there might be a free interchange of experience and that the Sections Committee might be in a position to make recommendations to the Council on any questions that deserved such consideration. Hugh R. Corse spoke briefly for the Buffalo Section. Orrel A. Parker, chairman Cleveland Section, discussed the matter of Section dues and felt that the plan of making all Society members Section members without additional dues, had many good points to recommend it. J. H. Hunt, chairman Dayton Section, said that his Section was fortunate in having the facilities of the Dayton Engineers Club at its disposal. Dayton meetings were well attended, good engineering papers always available and no difficulty experienced in securing Section members. Mr. Hunt believed in the preparation of programs and papers well in advance and stated that all the new committees of the Dayton Section were organized and functioning at the present time for the coming year.

The Detroit Section has just completed a very successful year according to George E. Goddard, its new chairman. Mr. Goddard outlined the program that had attracted such satisfactory attendance and mentioned particularly the great interest shown in production engineering subjects. He recommended strongly the scheduling of meetings by the other Sections on machine tools, manufacturing methods and means of attaining decreased production costs. K. K. Hoagg, vice-chairman, Detroit Section, represented his Section recently in a series of conferences that led to an allied organization of engineering society sections in Detroit. Mr. Hoagg said the Detroit Section of the Society did not affiliate with the other organizations because the cooperative nature of the joint meetings called for the presentation of papers on general rather than specialized subjects. The Detroit Section felt it could not conduct such meetings so that they would be of real constructive value to Society members and that there was danger also of losing the Section's identity and scattering the interest of the Section members.

There appears to be some sentiment in the Indiana Section for the waiving of Section dues according to its chairman, O. C. Berry. Mr. Berry was strongly opposed to the use of Section funds for the provision of professional entertainment at Section meetings. He felt that money thus spent could be used to much better advantage in securing able speakers and providing attractive engineering programs. The Indiana Section intends to organize its meetings well in advance and use them as an inducement in the solicitation of new Section members.

Taliaferro Milton, chairman, Midwest Section, discussed the plans of his Section to overcome the handicap of diversified employment and widely scattered geographical location of the members in Chicago and its environs. The Section has organized a large and active meetings committee. They are hoping to secure men of national reputation in the automotive field as speakers at their meetings. The topics are to be of general rather than specific technical interest. Highly technical meetings have not attracted satisfactory numbers and are believed unsuitable in the Chicago district because of the aforementioned diversification of automotive interest in this territory. Mr. Milton expressed a favorable opinion of section membership without added dues.

The Minneapolis Section finds a predominating interest in tractors and power-farming in its district, according to L. A. Emerson, who represented the Section at the luncheon. Naturally, interest in the Minneapolis meetings had suffered somewhat from the recent depression in the agricultural implement business, but the new officers are finding that the return of prosperity has awakened new and greater enthusiasm in their district and are confident of having an active year for the Section. During the luncheon, the following telegram was received from Phil Overman, secretary, Minneapolis Section, as evidence that things in the northwest are not quiescent.

GREETINGS FROM MINNEAPOLIS SECTION

> Our Summer Meeting and picnic today successful demonstration advantage social contact. Repetition as annual event assured.

R. E. Plimpton, secretary, Metropolitan Section, summarized the program of meetings held in New York City during the past year and mentioned the great interest shown in the subject of motor rail-cars. He said the plans for the series of meetings for the coming season were completed and speakers selected. The Metropolitan Section has organized a joint membership and reception committee the function of which is to introduce non-Section members at the meetings soliciting their Section membership at the same time. This suggestion is worthy of adoption by all of the Sections. It should be more productive of results than appeals by letter since the average prospect appreciates the personal interest shown in him.

It has been the experience of the New England Section officers that highly technical papers do not attract as large an attendance as those on general and non-technical subjects. R. E. Northway, who represented the Section, recommended that the Sections turn their attention to subjects not treated comprehensively in the national meetings, the manufacture of tires being an example. Sentiment in New England favored the waiving of Section dues. Mr. Northway did not believe Section membership could be built up satisfactorily except by personal solicitation.

The Washington Section was represented by Conrad H. Young, its treasurer. He said that attendance at their meetings was necessarily limited by the comparatively small Society membership in the district. The papers at Washington meetings are always of excellent calibre, many of them being presented by scientists and engineers in the Government service. However, it is difficult to secure the release of these papers for publication. The problem of collecting Section dues confronted the Washington officers but they were not convinced that waiving them would be an advisable step.

T. F. Cullen, chairman, Pennsylvania Section, discussing the matter of Section dues felt that the members showed more interest in the Section when they had to pay for the privilege of membership. The possibility of noon meetings had been proposed in the Pennsylvania Section but was dismissed in favor of the customary evening meetings after close study. Mr. Cullen favored interesting local automotive associations in the Section meetings and inviting their members as guests to increase the attendance, later soliciting for

membership in the Society any of those who are qualified.

Past-President David Beecroft recommended that greater thought be given by the officers to the staging of meetings. He did not believe our members appreciated or favored professional entertainment. The work of administering the Section's affairs should be well distributed, not shouldered by two or three officers. Mr. Beecroft favored having prominent local business men as guests at Section dinners. The Section should be a factor in all things automotive in its district and the business men should recognize its position in this respect. Service should be discussed in a local way, also automotive legislation. The Sections should each maintain close contact with the universities and educational institutions in their locality. Large meetings should be held in conjunction with the local automotive shows. It was Mr. Beecroft's belief that Section dues were not an obstacle if the meetings were made sufficiently attractive to the members.

President Bachman, though of an open mind on the matter of Section dues, did not believe the waiving of them would increase interest in the meetings. He cited cases he had personally investigated to substantiate his belief. He favored affiliation with local engineering bodies but only when the Society's identity was not submerged. Mr. Bachman hoped that the new Sections officers would recognize as one of their most important duties, the conduct of the election of delegates to the Nominating Committee of 1923. This committee is charged with what is virtually the selection of general Society officers and the election of its members should be conducted in the most serious and earnest manner.

A number of valuable suggestions were made in the round-table discussion following the short talks. Hugh Corse believed it advisable that the membership committee of each Section include a man of sales instinct able to formulate convincing letters for membership and dues solicitation. R. J. Nightingale likened the Sections to the roots of the national tree and emphasized the importance of keeping them healthy. He favored production meetings, and cooperation with other local engineering groups. Thos. J. Lítle, Jr., suggested a closer contact with the automotive companies in each Section territory, posting notices of the meetings in their factories and asking that they request their executives to attend the Section meetings. Conrad H. Young believed the dealer in automotive products could be interested more strongly in the Section meetings. George Goddard made the suggestion that the Sections especially invite to their dinners those pioneer members of the Society who by reason of attaining positions of great responsibility found it difficult to attend Section meetings.

A. K. Brumbaugh thanked the several Section officers for their suggestions and constructive thought, on behalf of the Sections Committee. He impressed them with the necessity of recognizing the responsibility that was placed upon them by the members. Without such recognition and an accompanying sacrifice of personal time, there could be anticipated only a mediocre success of the Sections. The Sections Committee is anxious to have every problem faced by the Sections placed in its hands for careful consideration and is certain that with the cooperation of the Council and Section officers there is no obstacle that cannot be surmounted.

Those who attended the luncheon were:

B. B. Bachman	J. H. Hunt
David Beecroft	Thos. J. Lítle, Jr.
O. C. Berry	H. O. K. Meister
A. K. Brumbaugh	Taliaferro Milton
Coker F. Clarkson	R. J. Nightingale
Hugh R. Corse	R. E. Northway
G. W. Cravens	Orrel A. Parker
T. F. Cullen	B. S. Pfeiffer
L. A. Emerson	R. E. Plimpton
G. W. Gilmer, Jr.	H. L. Pope
George E. Goddard	H. W. Slauson
L. Clayton Hill	E. W. Weaver
K. K. Hoagg	Conrad H. Young

CURRENT STANDARDIZATION WORK

(*Concluded from page* 118)

Batteries, p. B23 of the S. A. E. HANDBOOK, and are dimensions for the type of unit container that represents a new development in jar construction. The specifications follow.

(1) *Height.* Containers shall be made in two heights only; the B height for plates approximately 4¾ in. high and the C height for plates approximately 5¼ in. high.

(2) *Plate Supporting Ribs.* There shall be four ribs in each compartment on 1⅜-in. centers. The rib height shall be ¾ in. for B-height containers and ⅞ in. for C-height containers.

(3) *Top of Ribs to Top of Container.* These heights shall be

B-height containers..............6½ in.

C-height containers..............7 in.

(4) *Inside Width of Compartments.* The inside width of the compartment shall be 5 31/32 in. with tolerances of plus or minus 1/32 in.

(5) *Inside Lengths of Compartments.*

(a) Six-Compartment Containers

B Height		C Height	
S-3-B	1 5/16 in.	S-4-C	1½ in.
S-4-B	1½ in.	S-5-C	1 11/16 in.
S-5-B	1 11/16 in.		

(b) Three-Compartment Containers

B Height		C Height	
S-7-B	2 1/16 in.	S-8-C	2⅜ in.
S-8-B	2⅜ in.	S-10-C	2 13/16 in.
S-9-B	2 7/16 in.	S-13-C	3¼ in.
S-10-B	2 13/16 in.	S-14-C	3 5/16 in.
S-16-B	3 11/16 in.	S-16-C	3 11/16 in.
S-18-B	3 15/16 in.		
S-19-B	4⅛ in.		

(6) *Partitions Between Compartments.* The thickness of partitions between compartments shall be 3/16 in. minimum and ¼ in. maximum.

There was considerable discussion as to what the unit-battery container should be called. The following suggestions were submitted: cellbox, jarbox, cellblock, unit-container, integral container, container-in-block and monoblock container. The last name was finally decided upon as being the most descriptive and in accord with the use of the same term in cylinder construction.

It is not considered advisable to specify the dimensions for the outside of the container, since this would have a tendency to limit future developments. It was thought, however, that the dimensions agreed upon would serve as a very good basis for determining the ultimate outside dimensions.

The understanding was that the dimensions determined upon would be submitted to battery and hard-rubber manufacturers for further comment before final Division action is taken.

WIRE MESH

At the May 9 meeting of the Parts and Fittings Division it was stated that the wire-mesh manufacturers would like to have a standard established because at the present time each manufacturer makes practically any mesh that is ordered, one company alone making many hundreds of meshes of different wire sizes and mesh.

Applicants *for* Membership

The applications for membership received between May 18 and June 15, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ADAMS, JOHN NEWELL, student, University of Michigan, *Ann Arbor, Mich.*

ANTHONY, JOHN EDWARD, designing engineer, International Harvester Co., *Chicago.*

AUSTIN, E. W., district manager, Timken Roller Bearing Co., *Cleveland.*

BARTLETT, KING S., automobile mechanic, 770 South Commercial Street, *Salem, Ore.*

BARTHOLOMEW, J. R., assistant engineer, Westinghouse Pacific Coast Brake Co., *Emeryville, Cal.*

BASCH, JACOB JUSTIN, sales engineer, G. & O. Mfg. Co., *New Haven, Conn.*

BAUCH, CHARLES H., Navy Department, *City of Washington.*

BITTERMAN, SIMON, president and manager, American Auto Products Co., *Denver, Col.*

BREWER, HENRY, district sales manager, Leeds & Northrup Co., *Philadelphia.*

BUCKBEE, GEORGE A., salesman, Gurney Ball Bearing Co., *Jamestown, N. Y.*

BURT, LEO O., designer, Chevrolet Motor Co., *Detroit.*

BUTTERICK, W. B., chief mechanic and superintendent, Miller North Broad Storage Co., *Philadelphia.*

CHAMPION, E. M., works manager, American Motor Body Co., *Philadelphia.*

CLARK, WILLARD T., resident partner, Henry W. Peabody & Co., *Buenos Aires, Argentine Republic.*

CRAWFORD, KENNETH G., draftsman, Sanford Motor Truck Co., *Syracuse, N. Y.*

CROSS, CHARLES H., student, Ohio State University, *Columbus, Ohio.*

DANLY, PHILO H., engineer, tractor works, International Harvester Co., *Chicago.*

DOYLE, WILLIAM EDWARD, JR., student, Stevens Institute of Technology, *Hoboken, N. J.*

DUFFEE, FLOYD W., assistant professor, University of Wisconsin, *Madison, Wis.*

ERSKINE, ALBERT R., president, Studebaker Corporation of America, *South Bend, Ind.*

FOSTER, WILLIAM J., aeronautical mechanical engineer, Air Service, McCook Field, *Dayton, Ohio.*

FRENCH, C. A., engineer, International Harvester Co., *Chicago.*

GABBER, JACOB E., student, Ohio State University, *Columbus, Ohio.*

GARY, C. E., draftsman, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*

GINGHER, D. B., automobile mechanic, H. B. Tait & Co., *Columbus, Ohio.*

GUNN, GEORGE, JR., president and manager, Kelly-Springfield Truck Sales Co., *Seattle, Wash.*

HAMPTON, E. H., sales engineer, Hampton-Watson & Cia, *Buenos Aires, Argentine Republic.*

HERMANN, JOHN F., assistant engineer, Phelps Light & Power Co., *Rock Island, Ill.*

HINKLEY, RAY A., engine designer, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*

HOLMES, JOHN Q., metallurgical engineer, Nordyke & Marmon Co., *Indianapolis.*

HORTHY, WILLIAM A., mechanical engineer, tractor works, International Harvester Co., *Chicago.*

HOUSE, BRYAN E., designer, Maxwell Motor Co., *Detroit.*

JACKSON, E. F., manager manufacturers' sales, Goodyear Tire & Rubber Co., Inc., *Detroit.*

KELLEY, GEORGE L., metallurgist, Edward G. Budd Mfg. Co., *Philadelphia.*

KENNEDY, JAMES T., sales department, Goodyear Tire & Rubber Co., *Detroit.*

KNOWLES, FRANK LESTER, student, Ohio State University, *Columbus, Ohio.*

KNOWLTON, H. B., instructor, Central Continuation School, *Milwaukee.*

KOLB, GEORGE F., manager, motorcycle division, Bullard Machine Tool Co., *Bridgeport, Conn.*

LADAIR, THOMAS F., automobile distributor, 504 Cass Street, *Milwaukee.*

LAWRENCE, F. W., salesman, A. O. Smith Corporation, *Detroit.*

LEIGHTON, LIEUT. BRUCE G., bureau of aeronautics, Navy Department, *City of Washington.*

LEWIS, MAJOR BURTON O., Ordnance Department, *City of Washington.*

LONG, RAY A., chief engineer, Columbia Motors Co., *Detroit.*

LUTZENBERGER, LOUIS DICKSON, student, Ohio State University, *Columbus, Ohio.*

MCCORMACK, MERLE H., student, University of Michigan, *Ann Arbor, Mich.*

MCFAWN, FRED, district sales engineer, Stanley Works, *New Britain, Conn.*

MCWHIR, DAVID, chief inspector, Wright Aeronautical Corporation, *Paterson, N. J.*

MORTIMER, BRUCE G., assistant superintendent, garage service station, Wilson Motor Sales Co., *Toronto, Ont., Canada.*

MRAZ, EMIL, superintendent, Temme Spring Corporation, *Chicago.*

NATIONAL MALLEABLE CASTINGS Co., 10600 Quincy Avenue, *Cleveland.* (Affiliate member.)

PALMER, H. A., chief engineer, Reynolds Spring Co., *Jackson, Mich.*

PLEISS, PAUL, secretary and director, Budd Wheel Co., *Philadelphia.*

PLUMRIDGE, TOM G., consulting engineer, American Technical Society and American School of Correspondence, *Chicago.*

PRODOEHL, H. G., charge of tool design department, International Harvester Co., *Chicago.*

RAHUSEN, E. N., manager, Handelsbureau voor Automobiel & Vliegtuig-Industrie, *Ryswyk, Holland.*

RAGSDALE, R. J. W., engineer, Budd Wheel Co., *Philadelphia.*

REESE, EDWIN KENNETH, sales manager, vice-president, King Tool Co.; salesman and production manager, Packard Engineering Co., *Cleveland.*

RICARDO, HARRY R., consulting engineer, Ricardo & Co., Ltd., *Old Shoreham, Sussex, England.*

RICHARDSON, ARCHIBALD P., sales manager, Gurney Ball Bearing Co., *Jamestown, N. Y.*

ROUSE, GEORGE ALAN, assistant superintendent motor vehicles, Standard Oil Co. (N. J.), *Baltimore.*

SAVANT, A. K., student, University of Michigan, *Ann Arbor, Mich.*

SCULLY, JAMES N., president, Houdaille Co., *Buffalo.*

SEILER, PAUL W., president and general manager, Ternstedt Mfg. Co., *Detroit.*

SKINNER Co., LTD., *Gananoque, Ont., Canada.* (Affiliate Member)

SKURRAY, ERNEST CLEMENT, motor engineer, Skurray's, *Swindon, Wiltshire, England.*

SMITH, LEOPOLD J., draftsman, American Railway Express Co., *New York City.*

SMITH, STANFORD ALLEN, chief inspector, Lexington Motor Co., *Connersville, Ind.*

SMITH, T. D., sales manager, Harvey Electric Co., *Chicago.*

STINSON, KARL W., instructor in mechanical engineering, Ohio State University, *Columbus, Ohio.*

STOVER SIGNAL ENGINEERING Co., *Racine, Wis.* (Affiliate Member)

SUN Co., Finance Building, *Philadelphia.* (Affiliate Member)

SUTHERLAND, J. D., general sales manager, Wyman-Gordon Co., *Worcester, Mass.*, and *Harvey, Ill.*

TALL, G. W., JR., sales manager electric furnace division, Leeds & Northrup Co., *Philadelphia.*

TICHY, V. L., assistant metallurgist, White Motor Co., *Cleveland.*

TIMKEN, H. H., president, Timken Roller Bearing Co., *Canton, Ohio.*

VOORHEIS, GLENN IRVING, student, Michigan Agricultural College, *East Lansing, Mich.*

WAGNER, LEONARD J., draftsman, Ohio Body & Blower Co., *Cleveland.*

WELCH, STANLEY P., New York branch manager, Kelsey Motor Co., *Newark, N. J.*

WHARTON, THOMAS P., president, Wharton Motors Co., *Johnstown, Pa.*

WILSON, CHRISTIAN, president and chief engineer, C. Wilson Co., *Cliftondale, Mass.*

WILLS, C. HAROLD, president, C. H. Wills & Co., *Marysville, Mich.*

ZIMMERMAN, M. A., student, Ohio State University, *Columbus, Ohio.*

ZIMMERMAN, PAUL G., engineer, Aeromarine Plane & Motor Co., *Keyport, N. J.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between May 10 and June 9, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ADAMS, CONRAD A. (M) assistant professor of mechanical engineering, Tufts College, Mass., (mail) 108 College Avenue, *Medford, Mass.*

ALLEN, STANLEY C. (A) experimental engine laboratory, Reo Motor Car Co., *Lansing, Mich.*, (mail) 231 North Clements Avenue.

ANDERSON, NELS G. (M) chief engineer, International Harvester Co., Chicago, (mail) International Harvester Co., Lagonda Avenue, *Springfield, Ohio.*

BACHLE, ANDREW (M) vice-president, engineering division, Paige Detroit Motor Car Co., *Detroit*, (mail) 1409 Boston Boulevard West.

BAYERLINE, J. GEORGE (M) president and general manager, Columbia Motors Co., *Detroit*, (mail) 1003 East Grand Boulevard.

BROOKS, DONALD B. (E S) student, department of chemistry, Ohio State University, *Columbus, Ohio.*

BUCHER, CLARENCE DEAN (E S) student, Ohio State University, *Columbus, Ohio*, (mail) 410 Wyandotte Avenue.

BURLEY, HARRY B. (A) president, Boston Insulated Wire & Cable Co., *Dorchester, Mass.*, (mail) 65 Bay Street.

CLARK, HAROLD EDMUND (J) motor truck salesman, Packard Motor Car Co. of New York, *New York City*, (mail) 943 Teller Avenue.

COFFEY, HARRY L. (J) draftsman, (mail) 716 West Front Street, *Plainfield, N. J.*

CORMIER, ANDREW L. (A) U. S. S. Litchfield No. 336, *Portsmouth, Va.*

CRAIG, NORMAN (A) president and general manager, Light Alloys Co., 7016 Euclid Avenue, *Cleveland.*

CRISSMAN, IRA STANLEY (M) wheel engineer, Disteel Wheel Corporation, *Detroit*, (mail) 1633 Richton Avenue.

CROWELL, WILLIAM S. (A) instructor, Y. M. C. A. Automobile School, *San Francisco*, (mail) 1426 Waller Street.

DAVIS, WILLIAM N. (M) body engineer, Cadillac Motor Car Co., *Detroit*, (mail) 3794 West Euclid Avenue.

DORAN, FELIX, JR. (A) special representative, Chevrolet Motor Co., *Kansas City, Mo.*, (mail) 3714 Broadway.

DROPINSKI, ADOLPH (E S) student, Armour Institute of Technology, *Chicago*, (mail) 543 North Ashland Avenue.

EBERHART, CLEBURNE, JR. (M) president, Eberhart Steel Products Co., Inc., 812 East Ferry Street, *Buffalo.*

ENGBRECHT, HERMAN J. (A) service manager, National Motor Sales Co., *Chicago*, (mail) 2142 West 21st Place.

FIELD, BURNHAM E. (A) metallurgical engineer, Union Carbide & Carbon Research Laboratory, Inc., Thompson Avenue and Manley Street, *Long Island City, N. Y.*

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FOSTER, J. P. (M) superintendent, Maui Agricultural Co., *Paia, Maui, T. H.*

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GIRDWOOD, R. F. (A) automobile sales manager and engineer, Ledoux, Jennings, Ltd., *Montreal, Que., Canada*, (mail) 615 University Street.

GRAHAM, WELLINGTON R. (A) chief engineer, Kelland Motor Car Co., Newark, N. J., (mail) R. F. D. 2, Box 348, *Elizabeth, N. J.*

GROSS, EDWARD L. (M) vice-president, Perolin Co. of America, Peoples Gas Building, *Chicago.*

HARRIS, ELMER P. (A) proprietor, Harris Mfg. Co., 30 Church Street, *New York City.*

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KAU, PHILIP FUI (E S) student, Pratt Institute, *Brooklyn, N. Y.*, (mail) 231 Ryerson Street.

KAUFFMANN, ALFRED (A) vice-president and general manager, Link-Belt Co., P. O. Box 85, *Indianapolis.*

KELLER, ALEX W. (A) assistant to general superintendent, Atlas Powder Co.; Zapon Leather Cloth Co.; Celluloid Zapon Co.; *Wilmington, Del.*, (mail) Atlas Powder Co.

LEWIS, ROBERT H. (J) assistant superintendent, Maccar Truck Co., *Scranton, Pa.*, (mail) 434 Harrison Avenue.

LINDECKER, JOSEPH BRIGHAM (E S) student, Ohio State University, Columbus, Ohio, (mail) 1531 West Bancroft Street, *Toledo.*

LYNAH, JAMES (M) vice-president, Barnard-Lynah, Inc., 321 Broadway, *New York City.*

MADELUNG, GEORG H. (M) Glenn L. Martin Co., 16,800 St. Clair Avenue, *Cleveland.*

MAGGS, ALBERT H. (J) designing engineer, Ahrens Fox Fire Engine Co., *Cincinnati*, (mail) 40 East McMillan Street.

MANN, ERNEST W. (M) assistant engineer, Ward Motor Vehicle Co., Mount Vernon, N. Y., (mail) 123 Third Avenue, *North Pelham, N. Y.*

MAPEL, LEWIS A. (A) vice-president and general manager, Automatic Appliance Co., 416 North Third Street, *St. Louis.*

METZROTH, WILLIAM (A) factory manager, Dyneto Electric Corporation, *Syracuse, N. Y.*, (mail) P. O. Box 458.

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MORRIS, VICTOR ROSS (E S) student, Ohio State University, *Columbus, Ohio*, (mail) 58 West Woodruff Avenue.

PARKHILL, JAMES (A) plant manager, Armstrong Spring Co., *Flint, Mich.*, (mail) 1952 Miller Road.

PRAY, MAYNARD (J) assistant to manufacturing executive, Holt Mfg. Co., *Peoria, Ill.*, (mail) 616 North Jefferson Street.

PROVOOST, WILLIAM REES (J) Aeromarine Plane & Motor Co., *Keyport, N. J.*

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SMELLIE, EDWIN FROST (E S) student, University of Michigan, *Ann Arbor, Mich.*, (mail) 547 Elm Street.

SOMERBY, CHARLES T. (A) secretary and service manager, United Motor Car Co., Inc., 6 North Willow Street, *Trenton, N. J.*

STEPHENS, GEORGE H. (J) chief engineer, Fansteel Products Co., Inc., North Chicago, Ill., (mail) 954 Vernon Avenue, *Glencoe, Ill.*

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TABER, MELBERT W. (M) special representative, Motor Truck Corporation, Lansing, Mich., (mail) 601 Capitol Theatre Building, *Detroit.*

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TOUR, SAM (A) metallurgist, Doehler Die-Casting Co., *Brooklyn, N. Y.*, (mail) 2225 Ditmas Avenue.

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WERDEHOFF, ALBERT B. (M) chassis engineer, Zeder-Skelton-Breer Engineering Co., Newark, N. J., (mail) 55 Fairbank Street, *Hillside, N. J.*

WHITTLESEY, F. E. (M) secretary, treasurer and general manager, Raymond Mfg. Co., Ltd., *Corry, Pa.*, (mail) P. O. Box 401.

WICKEL, RALPH O. (E S) student, Armour Institute of Technology, *Chicago*, (mail) 3942 North Lawndale Avenue.

WINNER, CARL S. (S M) mechanical draftsman, Rock Island Arsenal, Rock Island, Ill., (mail) 13,304 Milan Avenue, *East Cleveland, Ohio.*

WOLCOTT, R. G. (A) vice-president, Victor Bearings Co., *Indianapolis.*

WOODS, LEONARD R. (J) salesman, Dorris Motor Car Co., *St. Louis*, (mail) 6248 Washington Avenue.

ZEDER, FRED M. (M) Zeder-Skelton-Breer Engineering Co., 24-26 Mechanic Street, *Newark, N. J.*

ZIRKEL, P. J. (M) chief tool designer, International Harvester Co., *Akron, Ohio.*

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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Chronicle and Comment

Operations Engineering

WE should maintain the broad view that the operating phase of the automotive business is just as important as the building of motor vehicles, in connection with bus lines or other motor-vehicle transport.—Past-President Manly.

Motor Car Engineering Vice-Presidency

BY way of addition to its report printed in the July issue of THE JOURNAL, the Nominating Committee of the Society has advised that it has nominated A. F. Masury to serve as second vice-president representing motor-car engineering during 1923.

Membership Increase

WE want to increase the membership of the Society, but we must be particular as to the kind of men we try to interest. There is not anything to be gained by getting a man in this year who will drop out next year.—Lon R. Smith, Chairman of Membership Committee.

The Sections

THE Sections Committee will render every assistance possible to the Sections officers during the summer when the latter will be planning meetings and selecting authors of papers of greatest general interest in the respective Section territories. The Committee is anxious to receive suggestions and constructive criticisms from members with a view to making the Sections meetings as interesting and valuable as may be.—A. K. Brumbaugh, Chairman of Sections Committee.

Standardization Surely Pays

ACCORDING to the statistics furnished by manfacturers and other authorities in the automotive industry, the annual savings resulting from the standardization effected by the Society of Automotive Engineers amount to 15 per cent of the annual turnover in the industry. This turnover is about $5,000,000,000, and the estimated saving therefrom amounts to $750,000,000 a year. No other society or association that we know of has made such great progress in the practical application of standardization. Hundreds of items have been made the subject of investigation and study, and many of the component parts used in automobile construction are now made by all manufacturers according to definite specifications. The marked improvements made in automobiles, without a proportionate increase in price, have been due in large measure to this work.—*Machinery*.

A Meeting on Production Problems

A NATIONAL meeting of the Society will be held in Detroit, Oct. 26 and 27, to discuss problems of automotive production. The meeting is to be known as the Automotive Production Meeting. Papers treating current production problems in a simple and practical way will be read and fully discussed in morning meetings on each of the two days. The afternoons will be devoted to factory inspection-trips especially arranged for the purpose of viewing new and advanced production methods that will particularly interest the tool, inspection and production men. The principal object of this meeting is the promotion of an interchange of experiences between practical factory men on automotive production problems that are troubling them in their daily work. Formality will be ruled out of the sessions, which are to be as near the round-table type of impersonal discussion as possible; they are to be forums of practical experience and not theoretical controversies. A Production Dinner will be held Thursday evening, Oct. 26, where social friendships between production men will be promoted in an atmosphere of informality and good fellowship. Announcement of further details of the Production Meeting will appear in the September issue of THE JOURNAL and a special issue of the *Meetings Bulletin*.

Grading of Members

IT has been the policy of the Society for many years, in the interpretation of Constitutional provision, that, aside from minimum age and distinguished service or noteworthy accomplishment, evidence of ability in some important phase of automotive engineering is requisite to qualify for Member grade. The committees and Councils of the Society have followed this interpretative policy faithfully and consistently in intent and in action; although, as stated by President Bachman at White Sulphur Springs, neither a committee nor a Council can be infallible.

In connection with the proverb, "Give a dog a bad name and hang him," it has been said that rules can be

made acceptable and ideals realized only as they appeal to something in human nature and awaken in it an active response. Rules have to be interpreted and responsibility for the interpretation taken by those making it. If the members of the Council knew personally those applying for membership or grade transfer, it would still be difficult to reach decisions in many cases. Dealing almost exclusively with written evidence, the problem is indeed difficult.

There was a healthful discussion of important phases of this whole matter at the business session of the Summer Meeting, statements being advanced to the effect that it is unduly difficult to secure Member grade and also in support of the general idea that it has been and is granted too freely.

The Society is an engineering organization. Only those of Member grade have voting or office-holding power in it as a national organization. It is recognized that many who cannot be classified as engineers can be very helpful in and assisted by the avowed and generally accepted activities of the Society. This has been fundamental in our procedure for over a decade, and those of the types indicated have been and are welcomed into the ranks.

In maintaining the integrity of the Society as an engineering organization, the members as a whole can aid greatly by familiarizing themselves adequately with the methods followed and advising individually in close cases in grading. They should, of course, return promptly, after filling them out very conscientiously in view of the rules, the reference blanks furnished them from time to time; and, in addition, they should write special letters as occasion requires. Each application for membership or transfer is considered carefully by several persons before it is acted upon by the Council. The more specific information available in each case, the better.

Applicants should give in suitable written form a comprehensive specific record of their education and professional or business experience. Frequently men who might well be considered for Member grade if sufficient information with regard to their work were at hand, fail to furnish the data needed. Naturally, in case of doubt the Member grade will not be recommended or granted by the Council. An applicant should have a clear idea of just what status he seeks in the Society and furnish full information so that an intelligent decision can be made in the first instance on this basis. Otherwise, in the event of dissatisfaction on his part, a needless amount of his time, as well as that of others, will be taken in further steps.

Of course, a technical-school education or an engineering degree is not requisite to Member-grade qualification. They have weight and may be decisive in a case that is otherwise close. Neither is the immediate occupation conclusive. A person who is qualified by technical training or experience to design or construct complete automotive apparatus or their important component parts may not be engaged in such work, but that would not bar him from election to Member grade. An applicant may be competent to exercise responsible technical supervision of the production of materials germane to the construction of automotive apparatus; to take responsible charge of automotive engineering work; or to impart technical instruction in the design and construction of automotive apparatus. Having any of these qualifications in the degree contemplated by the constitutional provisions, he may be elected a Member regardless of the work he is doing at the time. However, if there were any doubt as to his engineering qualification, he might be graded as an Associate or a Junior, or not elected to any grade of membership.

Frequently applications are received from men who are not qualified for any grade of membership in the Society. They are generally advised to subscribe for THE JOURNAL, placing them on a basis somewhat similar to that of the Enrolled Student. These men are for the most part mechanics or repairmen who are endeavoring to follow automotive engineering matters in a somewhat serious way.

It is the custom to publish the names of applicants in THE JOURNAL prior to considering electing them; and precaution is taken not to elect persons who are not of reputable character.

One of the difficult phases of the interpretation of the constitutional provisions as to Member qualification, C8 of the Constitution, arises in connection with the clause "qualified to act as designers or constructors of complete automotive apparatus *or their important component parts*." The italicized words obviously raise a real question as to the line of division. What is an *important* component part of automotive apparatus? Incidentally, how many or who are qualified to design or construct a *complete* automotive apparatus, involving the application of knowledge of probably more sciences and arts than any product of manufacture other than shipbuilding? Rational methods of procedure must be followed in various respects in interpreting any constitutional provision.

It was argued forcefully at one time that a magneto designer was not entitled to Member grade. This has not been the prevalent view for some time. Latterly there has been discussion as to the grade of membership to which specialists in the design of such articles as helical springs and pipe fittings are entitled, some insisting that the most experienced and successful men in these lines should not be elected Members. This is not the view of the Council.

In general, those who are engaged in the automotive and related industries in a responsible commercial, financial or manufacturing capacity, or so occupied as to be able to *cooperate* with automotive engineers, are qualified to be Associates of the Society. The Associateship is not an inferior grade as such, but merely a ranking in accord with the basic fabric of the Society. There is nothing inconsistent in the fact that an Associate of the Society may in his own business employ a Member of the Society. The Society is an *engineering* organization, in which cognizance is taken of the many ramifications of the work of the automotive engineer, and a commonsense method of procedure provided for and followed. If a man is competent to and so situated as to be able to cooperate helpfully in the broad work of the Members, he is qualified to be an Associate of the Society.

Somebody said, "No generalization is true, including this one." Many cases have to be considered specially. What has been said or done before does not always fit. The best basis for judging an individual or an organization is *performance*, giving due weight to what is important.

Members who do not understand or are dissatisfied with membership gradings should send to the Council specific statements of their view. The Council is without question liberal in the interpretation of what an engineer is, based on what he has done, with reference to automotive engineering. The fact that a man is very well qualified in some unrelated field of engineering does not, of course, entitle him to membership in the Society.

Temperatures of Pneumatic Truck-Tires

By F. O. Ellenwood[1]

Buffalo Section Paper *Illustrated with* Charts and Photographs

After pointing out that the operating temperature is a vital factor in the life of a pneumatic truck-tire, the author outlines an investigation that was conducted at the plant of the Goodyear Tire & Rubber Co. This sought to determine (*a*) the best means of measing tire temperatures; (*b*) the temperature effect of inflation-pressure, load, long runs, frequency of stops, and the sizes of the rim and the tire; (*c*) the temperature of various designs of tire; and (*d*) some suitable means of reducing large-tire temperatures. The main reason for the rise in the temperature of a tire is stated to be the generation of heat resulting from rapid flexing; and the various factors having to do with this generation of heat and its dissipation to the atmosphere are listed.

The laboratory testing-machine and the methods and apparatus employed to measure the temperatures are described. One method was to measure the initial temperature and determine the rise by the increased pressure as shown by a gage fastened to the hub. The necessary corrections for air leakage and changes in the volume of the tire during operation were determined; the results are presented graphically. A type of tire thermometer was subsequently designed and used, but this was not satisfactory on account of the great differences between the thermometer readings and the temperatures as calculated by the pressure-volume method.

Thermocouples placed inside the tubes were also tried and results that checked fairly well with those calculated by the pressure-volume method were obtained. Also a detachable type of thermocouple was placed in a hole drilled through the tread to the cushion or carcass as desired.

The necessity of knowing that the air in the tubes was not saturated required the use of an air separator for inflating the tires. The effects of convection currents inside the tube on the thermocouples and thermometers are discussed. The theory as to the manner in which the temperature of a tire is reduced by using a small quantity of water inside the tube is given and convincing experimental evidence is presented to substantiate this theory.

After summarizing his conclusions, the author expresses the belief that the study of pneumatic-tire temperatures is of much greater commercial importance and possibly of greater scientific interest than is generally realized. The rise in the temperature of a tire is a direct and sensitive indication of the waste of energy due to its operation. While this loss cannot be eliminated, it can be reduced by proper selection and combination of materials.

The development of pneumatic tires has been very great and rapid, so that now we secure wonderful service if they are given half a chance in the way of proper care. It is only natural that owners of trucks should desire pneumatic tires on their vehicles used in service where speed is of considerable importance. These large tires carrying heavy loads have brought many new and perplexing difficulties to their manufacturers. One such problem is that of making a large tire that will carry heavy loads for long distances at continuously high speed in hot climates and yet have a long life, for the casing and the tube. Under such conditions there has been some trouble from premature failure and yet the same casings and tubes would be very satisfactory in cool climates, thus showing clearly that the operating temperature is a vital factor in the life of a tire. The investigation described herein was therefore undertaken for the purpose of trying to do the following things:

(1) Find out the best means of measuring tire temperatures
(2) Determine the temperature effect of inflation-pressure, load, long runs, frequency of stops, undersized rims and tire size
(3) Determine the temperature of various designs of tire
(4) Find some suitable means of reducing large-tire temperatures

The experiments have all been made during the past 2 years at the Akron plant of the Goodyear Tire & Rubber Co. and I desire to express my appreciation of the commendable policy that permits a large part of the information thus far obtained to be given to the public.

What makes a tire become hot? This question has been asked many times by laymen and often by engineers. Various answers have been given. Many laymen think it is due to the slippage between the tread and the road. This is a factor of only the very slightest importance in nearly all cases. The main reason for the rise in temperature of a tire is the generation of heat by the rapid flexing of it or, in other words, its hysteresis.

A tire is made up of many layers of cotton cord and rubber compound, each of which has been most carefully prepared and fitted together in a very definite and painstaking manner; then the whole mass is cured under heavy pressure at a definite temperature for a certain time. This gives a tire that is remarkable on account of its strength, wearing qualities and flexibility. Although it is probably more flexible than any other material of equal strength, it is not perfectly elastic; hence, each time a section of such a tire is bent, energy appears at that section in the form of heat. This heat is conducted to the adjacent parts, so that the tire temperature rises rapidly immediately after starting the tire, and then less rapidly to some maximum value where it remains so long as the operating conditions do not change. This maximum temperature is reached when the generation of heat is just balanced by its dissipation to the atmosphere. The first of these two factors is influenced by

(1) Amount of Flexing
 (*a*) Load

[1] Professor of heat-power engineering, Cornell University, Ithaca, N. Y.

(b) Inflation-pressure
(c) Size of tire (chiefly sectional diameter)
(d) Supporting surface (flat, round, rough, smooth)
(e) Torque
(f) Side-thrust

(2) Rate of Flexing
(a) Speed of vehicle
(b) Circumference of tire

(3) Kind of Tire
(a) Size
(b) Design (thickness in particular)
(c) Fabric
(d) Compound
(e) Building
(f) Curing
(g) Previous use

(4) Sunshine

(5) Tread Slippage

The rate at which the heat produced is dissipated is regulated by

(1) Atmospheric Conditions
(a) Temperature
(b) Wind
(c) Moisture

(2) Speed

(3) Stops
(a) Frequency
(b) Duration

(4) Supporting Surface
(a) Temperature
(b) Material

(5) Mounting on Vehicle
(a) Mud-guards
(b) Nearness to exhaust
(c) Kind of wheel

(6) Kind of Tire
(a) Size
(b) Thickness
(c) Material
(d) Previous use
(e) Nature of external surface

(7) Artificial Cooling

Some of these factors are of only minor importance, while others are vital. Those of the utmost importance are

(1) Speed
(2) Load
(3) Inflation-pressure
(4) Length of run
(5) Kind of tire
(6) Atmospheric temperature
(7) Sunshine
(8) Supporting surface

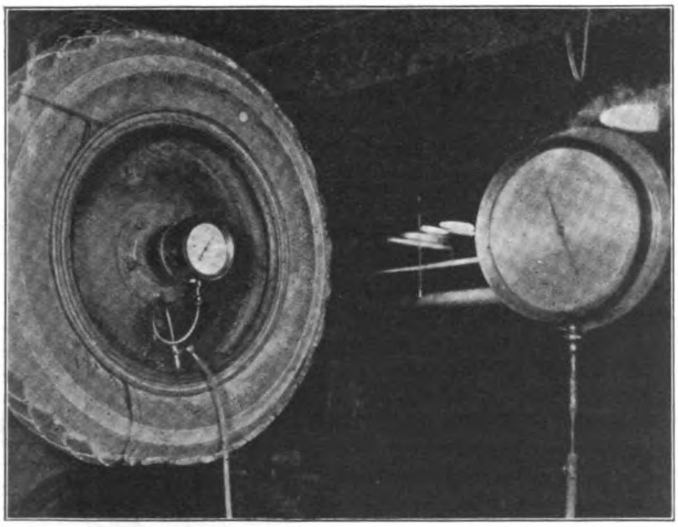

Fig. 1—General View of the 40 x 8-In. Tire Mounted on Testing Machine Showing the Pressure Gage Fastened to the Hub of the Wheel and the Large Standard Test-Gage Connected to the Tire for Daily Calibration of the Pressure Gage

A glance at these factors will indicate that it is not always easy to answer the common question, How hot does a tire become? Furthermore, a little consideration given to the question, How is the best way to determine this temperature? will show that this is a matter that cannot be answered easily. A search of the scientific literature yielded almost no information concerning the temperatures attained by pneumatic tires or the methods of measuring these temperatures. It was therefore necessary to develop methods of making such measurements.

From the beginning of the experiments it was felt that two or more independent methods of determining the tire temperatures should be used, if possible, to have a check on the results. One method that appeared to be accurate and reliable was the pressure-volume method or, in other words, measuring the initial temperature and the increase of the air pressure and volume very accurately and then calculating the temperature of this air in the tire. The question of how to determine the pressure was next in order. A detachable-gage method could not well be used as the leakage of air each time the connection was made and broken would be too serious. It was therefore necessary to use a stationary gage with some form of special connection to the rotating wheel, or mount the gage upon the hub of the machine so that it would turn with the tire. This last method was used very successfully. The wear on the gages is appreciable but not nearly so much as had been feared, especially by the gage manufacturers. Some gages are in fairly good shape after having been turned over more than 6,000,000 times.

Fig. 1 shows a tire and its attached gage mounted on the testing machine. The gages were taken off and calibrated frequently by the usual dead-weight tester. They were also checked in place at the beginning of each run by the large portable standard-gage shown at the right of the illustration. This check was made also for the reason that it is essential that the initial pressure be obtained as accurately as possible, since all subsequent calculations are based upon it. To connect this large portable standard-gage with the tire under test it was necessary to design and build a special valve-opener so that the connection might first be made to the valve-stem and then the valve itself opened without losing any air. With proper attention paid to the gasket in this special valve-opener it proved very satisfactory.

Fig. 1 also shows how the tires were mounted for these tests. The cast-iron pulleys by which the tires are driven are about 45 in. in diameter and a suitable framework holds the tires in position on top of these pulleys. A known load is applied by weights to the axle on which the wheels are mounted.

Leakage Corrections

There is always bound to be more or less leakage from any inner tube. This leakage depends upon a number of factors, such as the composition, thickness, temperature, age, and previous use of the tube, and the pressure of the air inside it. It is therefore necessary that the leakage shall be determined carefully for each run, thus giving the leakage under the complex conditions just as they exist.

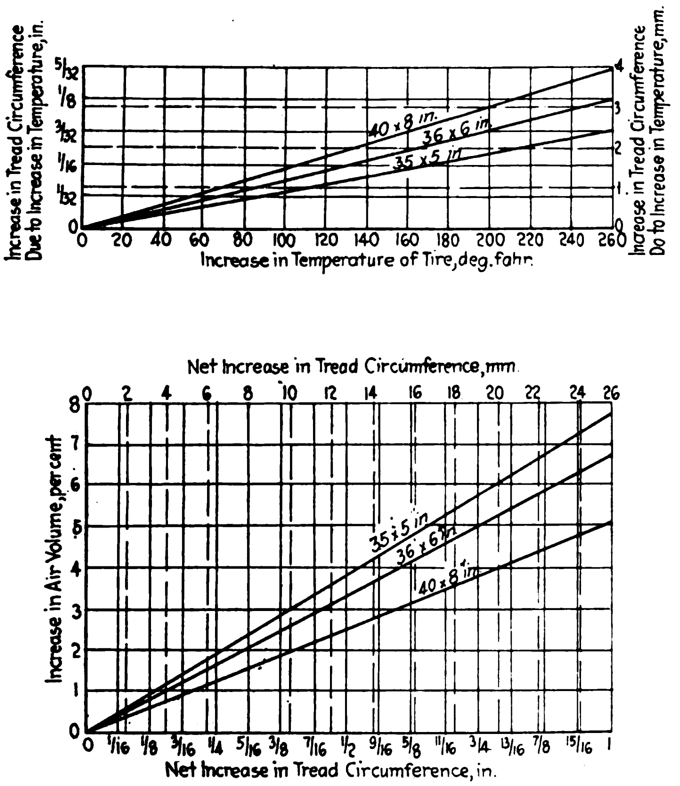

FIG. 2—CURVES SHOWING VOLUME CHANGES IN OPERATION
The Upper Curves Show the Increase in the Circumference of the Tread Due to the Thermal Expansion of the Tread and the Carcass, while the Lower Set Shows the Percentage of Increase in the Air Volume for Various Increases in the Tread Circumference Resulting from the Stretching of the Tire

The best way to make the leakage correction is to determine the average rate for the 24-hr. period following the start of the test. If this rate is small, as it usually is, the distribution of it may very properly be made on the basis of time alone. If it is appreciable the additional factors of temperature and pressure should be introduced in its distribution. Special effort was made to secure correct leakage corrections in all of these tests. A very close check was obtained on the observed leakage of one old tube by taking the pressure readings each hour during a 6-hr. run and the 18 hr. following it, then calculating the leakage on the assumption that it would be proportioned to the absolute temperature of the air in the tube and to the square root of the gage pressure. This method of distribution proved to be very satisfactory. Special curves were prepared to facilitate its application. In obtaining the 24-hr. leakage rate it will be found that usually the temperature and volume of the air in the tire are not the same at the end of this period as they were at the beginning. This necessitates that these values shall be determined and the proper corrections applied.

VOLUME CHANGES DURING OPERATION

Soon after beginning this work it was found that a pneumatic tire changes its volume very appreciably with variations in pressure, temperature and use. Various means of measuring this expansion were tried, the most convenient and accurate one being the measurement of the longitudinal tread circumference by a steel tape After the increase in tread circumference has been determined, calculations may then be made to show the corresponding change in the volume of any tire whose cross sectional dimensions are known. Fig. 2 shows such relations for three different sizes of tire. Fig. 3 shows directly the temperature corrections that are due to changes in tire volume, for various initial temperatures and pressures. The temperature corrections for the change in volume are surprisingly large for the first day's run, being as much as 15 deg. Fahr. for a 40 x 8-in. tire in some cases, while the corrections on subsequent days would be only from 3 to 6 deg. It is also very interesting to note that after the tire has been run for a considerable mileage and then deflated and left to stand for an appreciable time, it will also nearly recover its original volume. When a tire of this kind is again used its volume will increase very materially during its first run after its long rest, just as a new tire would do.

TIRE THERMOMETERS

Soon after attempting to ascertain the best ways to measure tire temperatures, we believed that a successful tire thermometer might be designed and built. There would be two advantages in such a thermometer. In the first place, if it could be built to stand up under operating conditions and at the same time be reliable and not too much trouble to install, it would be the means used in all future tests to determine tire temperatures. On the other hand, if it were only partially successful it certainly would be worth while to have one solely for the purpose of obtaining the actual temperature of the air in the tire at the beginning of each test. This initial

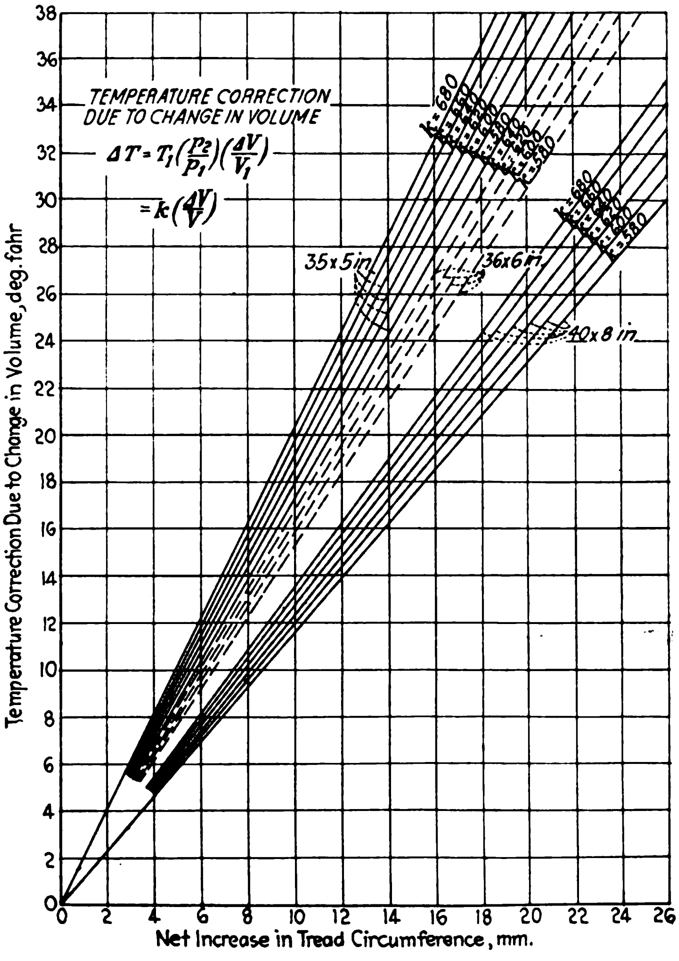

FIG. 3—CURVES SHOWING CORRECTION TO BE ADDED TO THE TEMPERATURES AS CALCULATED FROM THE INCREASE IN PRESSURE

V_1 = Initial Volume of the Air in the Tube
ΔV = Increase in Volume of the Air in the Tube
P_1 = Initial Absolute Pressure of the Air in the Tube
P_2 = Final Absolute Pressure of the Air in the Tube
T_1 = Initial Absolute Temperature of the Air in the Tube
ΔT = Temperature Correction

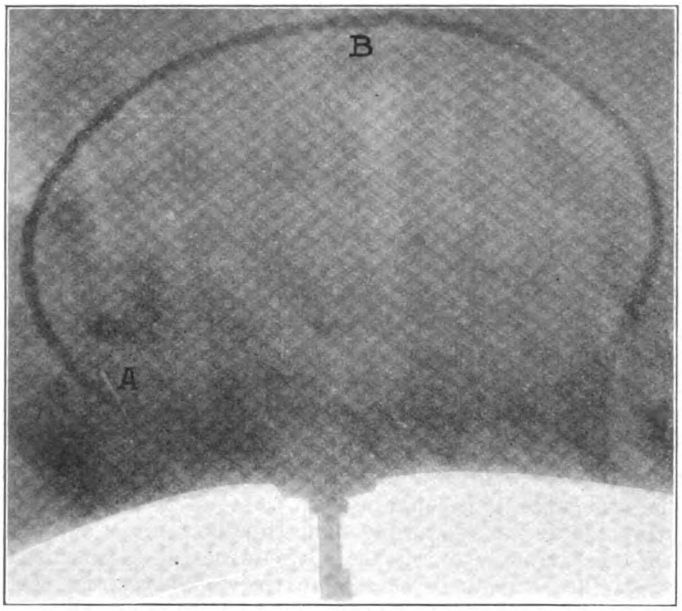

Fig. 4—X-Ray Photograph of a Side View of Part of a 40 x 8-In. Inner Tube with Thermocouples at *A* and *B*, Each Kept in Position by Wrappings of Piano Wire That Are Soldered to the Stem

temperature is very important because all of the subsequent temperatures depend upon it when calculated from the pressure and volume measurements. The only other way of determining this initial temperature of the air in the tire is by placing a thermocouple inside the tube; or by keeping a record of the room temperature for several hours preceding the test and determining the lag of the temperature of the air in the tire with reference to room temperature. At the time mentioned we had not been able to make a satisfactory thermocouple to place inside the tube.

Tube Thermocouples

We tried various ways before getting the thermocouple wires inside the tube so that there would be no leakage,

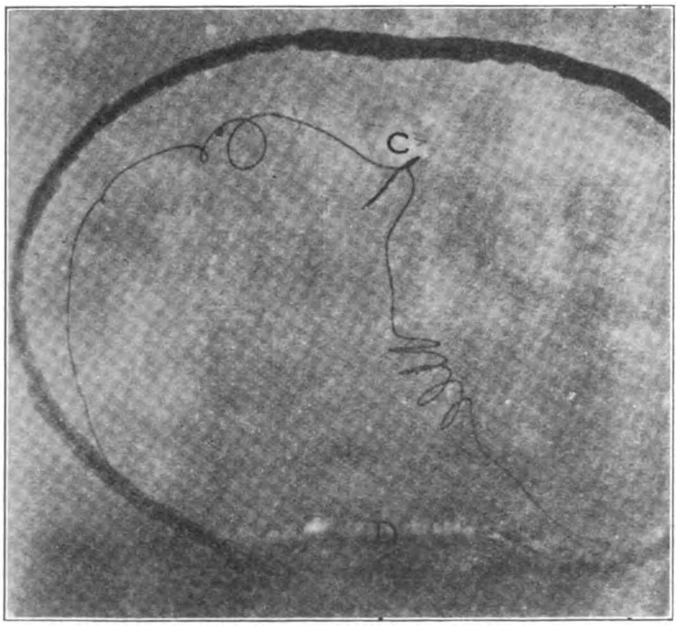

Fig. 5—X-Ray Photograph Showing a Side View of Part of a 36 x 6-In. Inner Tube with Thermocouples Held near the Center at C and the Bottom at D by Cords Attached to the Piano Wire

the installation correct electrically, the construction strong, the couple held near the center of the air-space inside the tube, and the couples attached to the regular valve-stem. This last item is of considerable importance since it means the elimination of the trouble and annoyance of having to prepare a special tube with its extra valve-patch and the extra hole through the rim, as well as the additional trouble of mounting the tire and tube having this extra valve-stem.

Fig. 4 is an X-ray photograph of a 40 x 8-in. tube containing two thermocouples, one near the center of the tube at *A* and one near the bottom or rim at *B*. This type is very rugged mechanically, as shown by the fact that each one survived the splicing operation without distortion, but the heat that may be conducted from the couple itself by having the couple wires wrapped to the insulated supporting piano wire is a possible objection to this type. Fig. 5 is another side-view of an X-ray photograph of a tube which was taken after the couples in this 36 x 6-in. tube had been run during many tests. The couple near the center *C* is supported by an invisible cord that runs horizontally from one side of the wrapped supporting wire to the other. The bottom couple *D* has been broken from its supporting cord. The coils of fine thermocouple wire were symmetrical before the tube was spliced and inserted in the casing. However, it should be noted that despite its rough usage during splicing the center couple is still in its proper position. This is a very good type of construction as it is rugged and the heat conducted away from the couple is a minimum. With this type we obtained some very satisfactory checks on the temperature as obtained by the pressure-volume method.

Convection Currents

The low readings obtained from the tire thermometers and the tube thermocouples as compared with the temperatures calculated from the pressure and volume puzzled us very much. To eliminate the possibility of moisture getting into the tubes from the compressed air, a special separator containing calcium chloride was made and used when inflating each tube. By this means and also by samples of air drawn from the inflated tires we satisfied ourselves that the amount of moisture inside the tubes was very much less than that required for saturation. We were now sure that our calculated temperatures were correct, but we could not believe that the small wires used in the thermocouples would conduct away enough heat to cause the low readings obtained from them.

We next began the investigation of the effect of the convection currents inside a hot tube when standing still and when running. This proved to be very illuminating. We had always been reading the temperature from the inside couple when in a position some 30 deg. from the bottom (see Fig. 6) as this was the position in which the tires were always stopped to have the gage exactly right to obtain its correct reading. We now compared the couple readings in this position with those of the same couple in positions near the top to which we would quickly turn the tire. This gave a difference that usually would be 5 deg. or more. It is not easy to obtain this difference correctly as the tire must stand in one position long enough to enable the couple to acquire the temperature of the air in that position; then the time must be noted when the reading is taken to correct for the cooling that occurs after taking the reading in the first position. However, we did establish to our own satisfaction the existence of the convection currents shown by Fig. 6 for a hot tire that has just been stopped.

Using a set of collecting rings made of the same material as the couple wires themselves, we were next able to determine while the tire was running the difference in temperature readings between the couple near the rim and the one near the center of the tire. This difference was found to be large as will be shown later. The rotation of the tire causes very rapid convection currents in the transverse section of the tire. This also will be discussed more fully later. We were convinced that the thermocouples are subject to appreciable correction due to convection currents, and that the magnitude of these corrections depends upon the position of the couple relative to the center of the tube, whether the tire is running or standing still. These same convection currents would account for a considerable part of the discrepancy still found to exist in the case of the tire thermometer in a well with many fins on it. This information, coupled with that concerning the heat conduction, which was such a big factor with the tire thermometer, now gave us peace of mind regarding our temperatures as obtained by the pressure-volume method, and made us feel certain that for comparing the temperatures of different designs of tire this is the best method of which we have any knowledge.

Detachable Thermocouples

After having experimented with tube thermocouples and realizing fully some of the serious difficulties that cannot be overcome on account of their being inside of the tube, it was felt that it would be worth while to experiment with detachable couples. By this term we mean couples that are inserted in small holes drilled through the tread to the cushion, breaker or carcass as desired. After making satisfactory holes through the tread, it is necessary to have a couple made of very fine wire mounted on the end of a small insulating plug that can be inserted in the hole. Wood and hard rubber are suitable but the latter is preferred because it is not broken so easily.

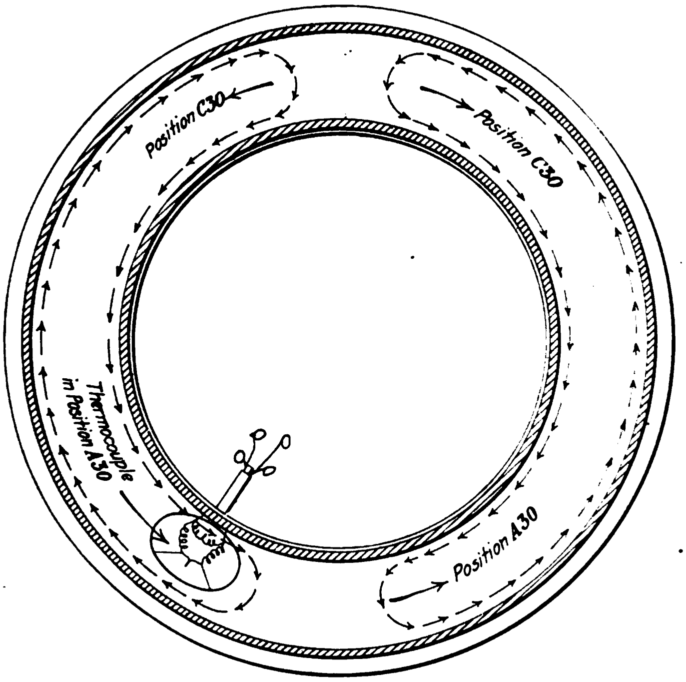

Fig. 6—Convection Currents in a Hot Tire Immediately after Stopping
The Dotted Lines Indicate the Path and Arrowheads the Direction of the Convection Currents when the Tire Is Standing Still. All Temperatures Were Obtained by the Pressure-Volume Method with a Load of 2500 Lb. at a Speed of 30 M.P.H.

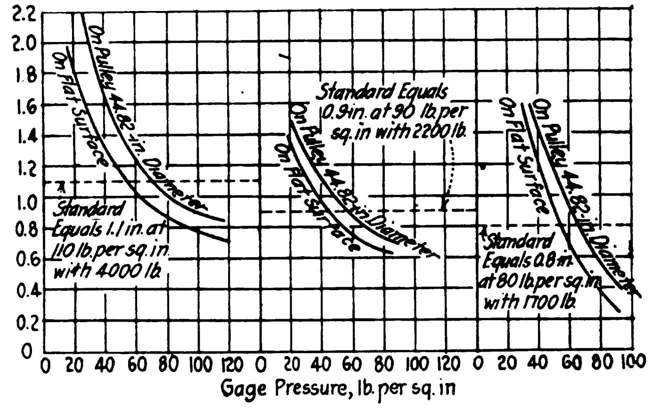

Fig. 7—Deflections of Three Sizes of Tire Having Various Inflation-Pressures when Supported by Flat and Curved Surfaces

This type of detachable couple can be inserted immediately after stopping the tire and the temperature indicated by the potentiometer will rise to its maximum value very quickly after insertion. It is not, however, an instantaneous process and the reading must not be made too quickly after the insertion of a cold plug, as it may require at least 30 sec. to reach the maximum temperature. With every care that we could exercise in all particulars it was still found that for some unknown reason this method would give occasional readings which were 10 or 15 deg. too high or too low to plot a smooth curve. Such cases, however, were very infrequent and our experience indicates that this is a method of great value, especially as a supplement to the pressure-volume method. Obviously one of its greatest advantages is that it enables the temperature of the stock itself to be measured and our investigations have shown that this temperature is considerably higher than the temperature of the air inside the tube. The other chief advantages of this system are the simplicity of the apparatus and the ease of renewal of the couple. It is also obvious that it has one disadvantage, namely, the injury to the casing resulting from drilling a hole through the tread. For laboratory-test tires this in itself is not a serious matter because there is no sand or dirt to work its way inside the carcass. It seems likely that there may be an appreciable correction to be added to the readings obtained from couples inserted in a hole in the rubber or carcass. This correction is due to heat conduction through the wires from the couple, and is greater the shallower the hole. This phase of the subject is now being carefully investigated by the research division of the Goodyear company.

Some Test Results

Before making any comparative temperature-tests it was necessary to determine the deflection of various sizes of tire when resting on the curved surface of the testing machine. By deflection of a tire we mean the vertical drop of any part of the wheel on which the tire is mounted as a result of applying a known load to the tire. This deflection causes the lower part of the tire to flatten until the contact area becomes sufficiently large to support the load with the air pressure existing in the tire.

Fig. 7 shows the results of deflection tests on three sizes of tire for various inflation-pressures, two kinds of surface and one load. These curves show what load is needed to give a normal deflection of the tire when mounted on the testing machine. For the runs that were made on the 40 x 8-in. tires this machine was not

strong enough to permit full load, so the full-load deflection was obtained by reducing the inflation-pressure. This undoubtedly reduces the tire temperature below what it would be for full load and full inflation-pressure. Just what this relation may be for various sizes of tires is a matter that is still being investigated.

Fig. 8 shows several interesting things. The first is how rapidly the temperature rises during the first 60 miles, then less rapidly to its maximum, which is near 150 miles, for an 8-in. tire when running at 30 m.p.h. The smaller the tire the sooner it reaches its maximum temperature. Also observe what the temperature effect is of using an undersized rim on a 40 x 8-in. tire. This is a practice not to be encouraged in general, and one to be distinctly avoided in hot climates. It will be noticed how fast the temperature of this size of tire rises with decreasing inflation-pressures. For the load of 2500 lb. the lowest pressure, 38 lb. per sq. in., represents a deflection nearly 60 per cent more than normal. This is what many tires sometimes have to endure. These curves indicate that the "air thermometer made out of a tire" may be entitled to the high rank usually accorded this type of thermometer.

Fig. 9 shows the temperatures of the various parts of an 8-in. tire as obtained by the several methods described. The reason for the inside couple near the center of the tube giving values lower than those obtained from the pressure and the volume is probably largely due to heat conduction from the type of couple used, shown by Fig. 4, and to the convection currents. These readings from the inside couples were all obtained while the tire was running, except those taken after 5½ hr. of the test when the tire was stopped. Attention is called to the very rapid drop in temperature of the eighth ply and of the cushion in the tire immediately after stopping, at the end of 150 miles. This is a 12-ply tire with a thick tread. This test shows, as did many other similar to it, that the air in the tube is some 15 or 20 deg. cooler

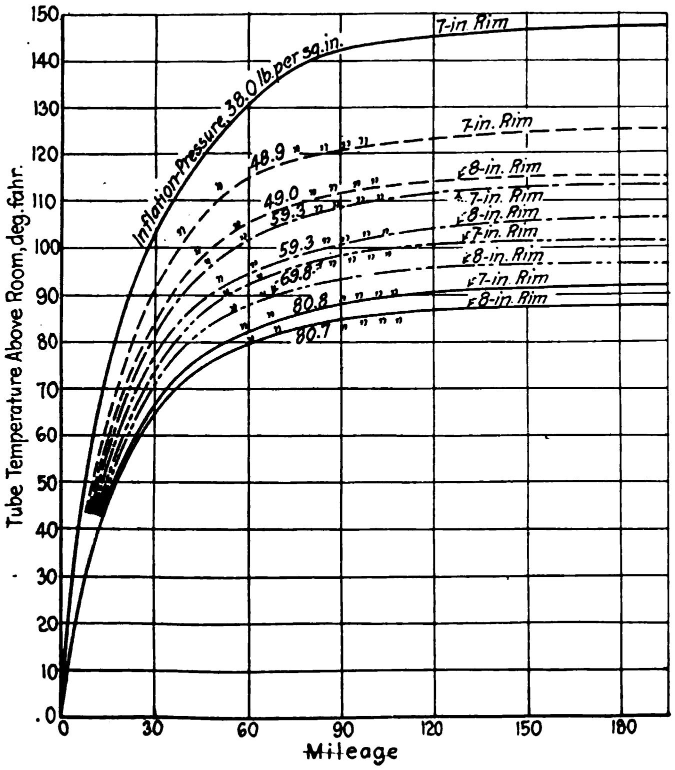

Fig. 8—Temperatures of a 40 x 8-in. Tube Above the Room for Various Mileages and Inflation-Pressures and Two-Rim Sizes

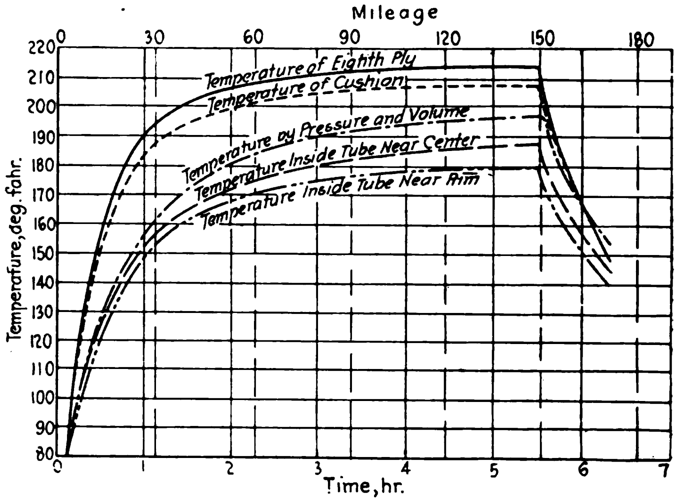

Fig. 9—Temperatures of Various Parts of a 40 x 8-in. Tire as Obtained by Various Methods, the Room Temperature in This Case Being 90 Deg. Fahr.

than parts of the casing. It also shows how quickly the high temperatures are reached by those parts of the tire in which the heat is produced, so that long distances do not need to be covered to produce temperatures that may be injurious to the carcass.

Artificial Cooling

For those sections of the Country where there are long periods of very hot weather it is undoubtedly true that some simple means of keeping large truck-tires cooler than they are under present conditions would be very valuable. It was therefore one of the purposes of these experiments to ascertain if possible some means of doing this. The most successful method of artificial cooling investigated was that in which a small quantity of water is placed inside the tube before inflating it with air. Many such experiments, beginning in 1919, have been run and in all cases the reduction of the average tire-temperature by water has been very noticeable. The hotter the tire the greater this effect, for reasons that will be discussed later.

The theory which prompted trying water inside of the tube is this. First, we should expect a certain reduction of heat generation because of the reduced flexure of the casing by reason of the increased pressure inside of the tube due to the pressure of the water vapor. This vapor-pressure would be small, but nevertheless would act as a sort of safety-valve in under-inflated tires because the hotter the tire the greater would be such pressure. The vapor-pressure rises faster than the air pressure for a given increment of temperature.

In the second place we should expect that the vapor inside of the tube would transfer a large amount of heat from the hot portions of the tube to the cooler parts of it since as the tire rotates the water is kept in a thin band extending entirely around the tread portion of the tube. A section of the top part of a rotating tire with some water in it is shown in Fig. 11. That portion of the tube covered by the water is the hottest. The water absorbs heat from the hot part of the tube and the vapor thus formed carries a large amount of heat from this part of the tube to the cooler portions lying next to the rim and sidewalls on which condensation takes place. There is a very rapid circulation of the air and vapor in all transverse sections of the tube, as indicated by Fig. 10, due to the difference in temperature between

the tread and bead sections of the tire. The faster the tire rotates the more rapid is this circulation. Thus it is easy to conceive of an extremely rapid circulation in every transverse section of a rapidly rotating tire whether it has vapor in it or not. The addition of water-vapor produces more rapid circulation as it increases the difference in densities for equal temperature-differences. It is, however, in the vastly superior heat-carrying capacity of a wet vapor, as compared with dry air, that we should expect to find the main advantage of the vapor. The delivery of more heat to the cooler parts of the tube lying next to the rim and sidewalls means increasing the ability of the tire to dissipate heat to the atmosphere. If this be true, we should find a very marked decrease in the temperature of the hottest parts of the tube, which always fail first, and an increase in the temperature of the coolest parts of the tube, which never fail from high temperature. The average tube-temperature should also be lowered in proportion as the ability of the tire to dissipate its heat is increased. This would be best determined by the temperature of the air in the tube, as the average

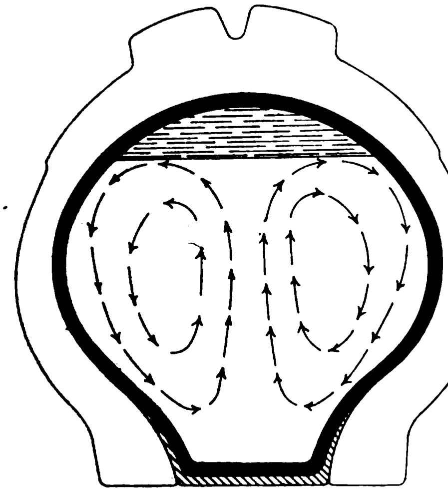

FIG. 10—CROSS-SECTION OF A ROTATING TIRE WITH ABOUT 10 PER CENT OF THE TUBE VOLUME FILLED WITH WATER
The Arrows Show the Path of the Convection Currents

temperature of the tire cannot be so nearly measured by any other means.

In the third place, we should expect a cooling effect by reason of the absorption of heat by the liquid itself, until this liquid becomes heated up to the same temperature as the tube. Here we should expect a cooling effect that would depend upon the amount of liquid used and the distance run without stopping. On the other hand, the effect of the vapor will be independent of the amount of liquid used. The experimental evidence will now be presented to substantiate this theory.

Fig. 11 shows clearly the cooling effect of water inside the tube of a 40 x 8-in. tire in amounts varying from 5 to 25 per cent of the air volume of the tube or from about ¾ to 3½ gal. of water. The larger amounts are advantageous only during the first few hours of the run, as the temperatures with water in the tube all finally approach the same value. This wet-tube temperature is seen to be very appreciably lower than when using the dry tube. These curves were all taken with the same tire, tube, load and speed and also with the same inflation-pressure which was kept low to give a flexure about 30 per cent greater than normal.

Fig. 12 shows what portion of the total pressure in the tubes of two badly under-inflated tires is due to the

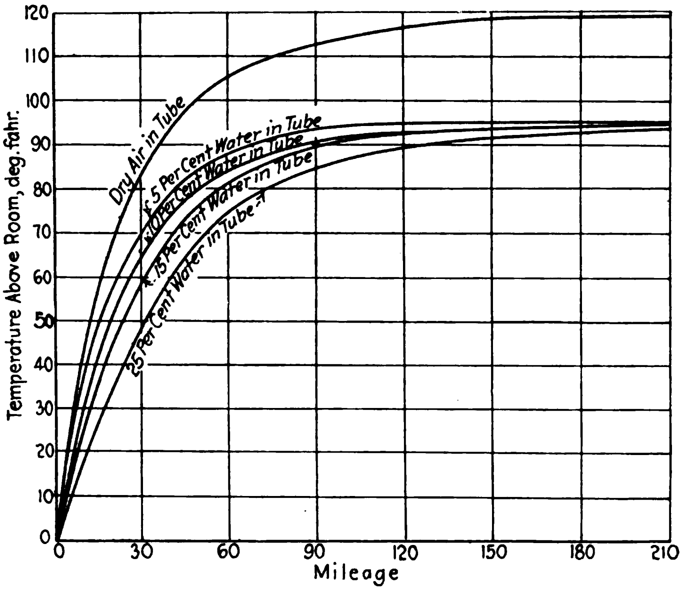

FIG. 11—TUBE TEMPERATURES RESULTING FROM VARIOUS AMOUNTS OF WATER IN THE TUBE

vapor. For each tire it will be noticed that the total pressure is slightly greater with the dry tube until about 30 miles have been run. From that point on the temperature has become high enough so that the vapor-pressure more than compensates for the lower air-pressure in the

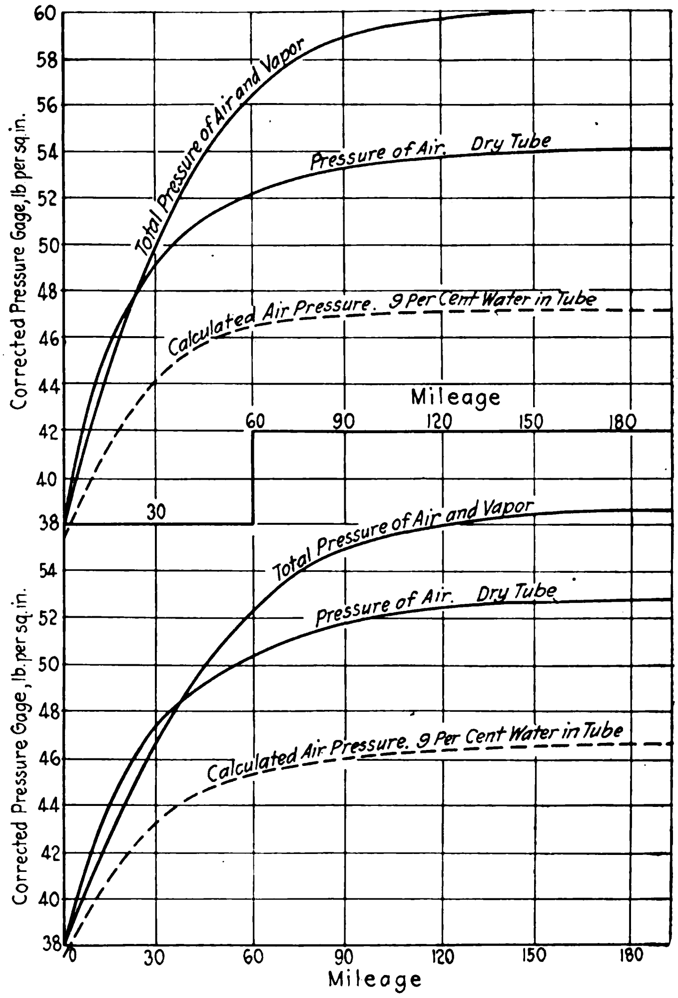

FIG. 12—COMPARATIVE PRESSURES IN THE SAME TIRES WITH AND WITHOUT WATER IN THE TUBES

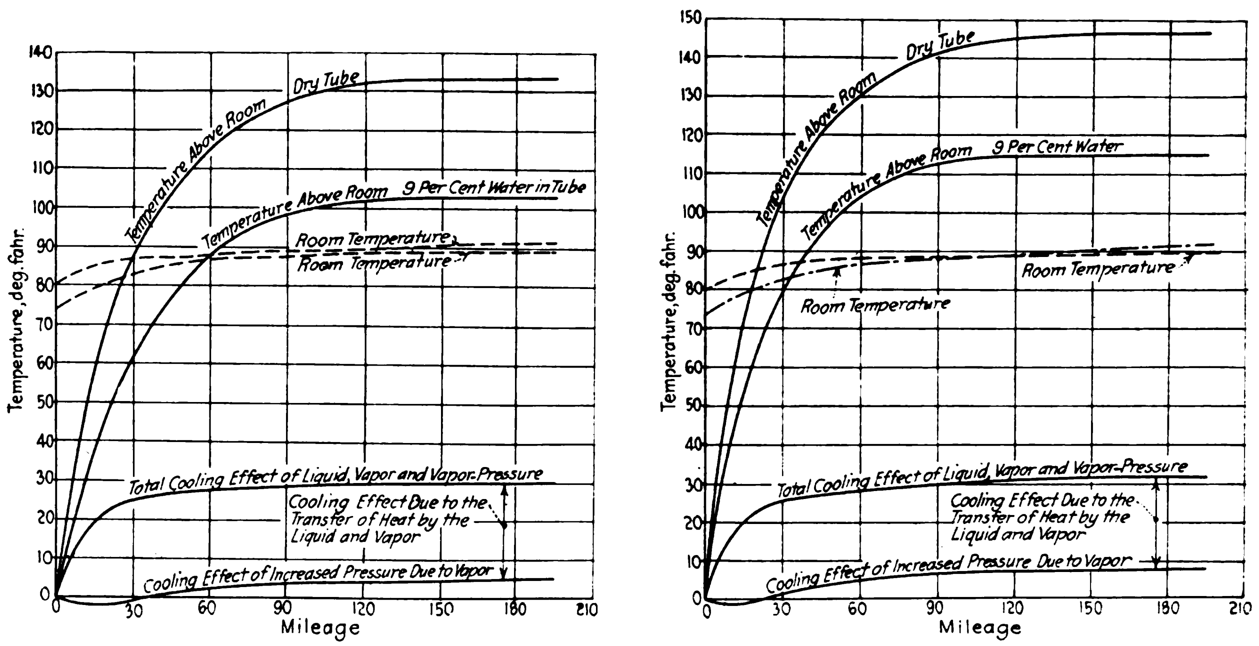

FIG. 13—CURVES GIVING AN ANALYSIS OF THE EFFECT OF WATER IN THE TUBE

wet tube due to its lower temperature. In Fig. 13 the wet tubes are seen to be about 30 deg. cooler than the dry ones after 90 miles have been run. At 30 miles this difference is about 25 deg. With these tires badly under-inflated one of the dry tubes reaches a temperature above that of the room of 147 deg. fahr. and the other one goes to 134 deg. This difference is due to the variation in the design of the two casings. With 9 per cent of water in each of these tubes in the same casings as before we have the tube temperatures reduced more than 30 deg. How much of this cooling is due to the vapor-pressure may be found from the bottom curve in each chart. This is seen to be about one-fourth of the total cooling effect, showing that the vapor must be given the credit for most of the benefit derived from the water in the tube.

The curves given at the bottom of Fig. 14 are extremely valuable as they help to explain so much that was at one time very puzzling and annoying. Reference has been made to the fact that the tire thermometers always gave temperatures much lower than those obtained by calculation from the pressure and volume measurements. With water in the tubes the tire thermometers would for many tests give higher readings than with the dry tubes. I believed that this was because the vapor caused a greater reduction in the correction of the tire thermometer than

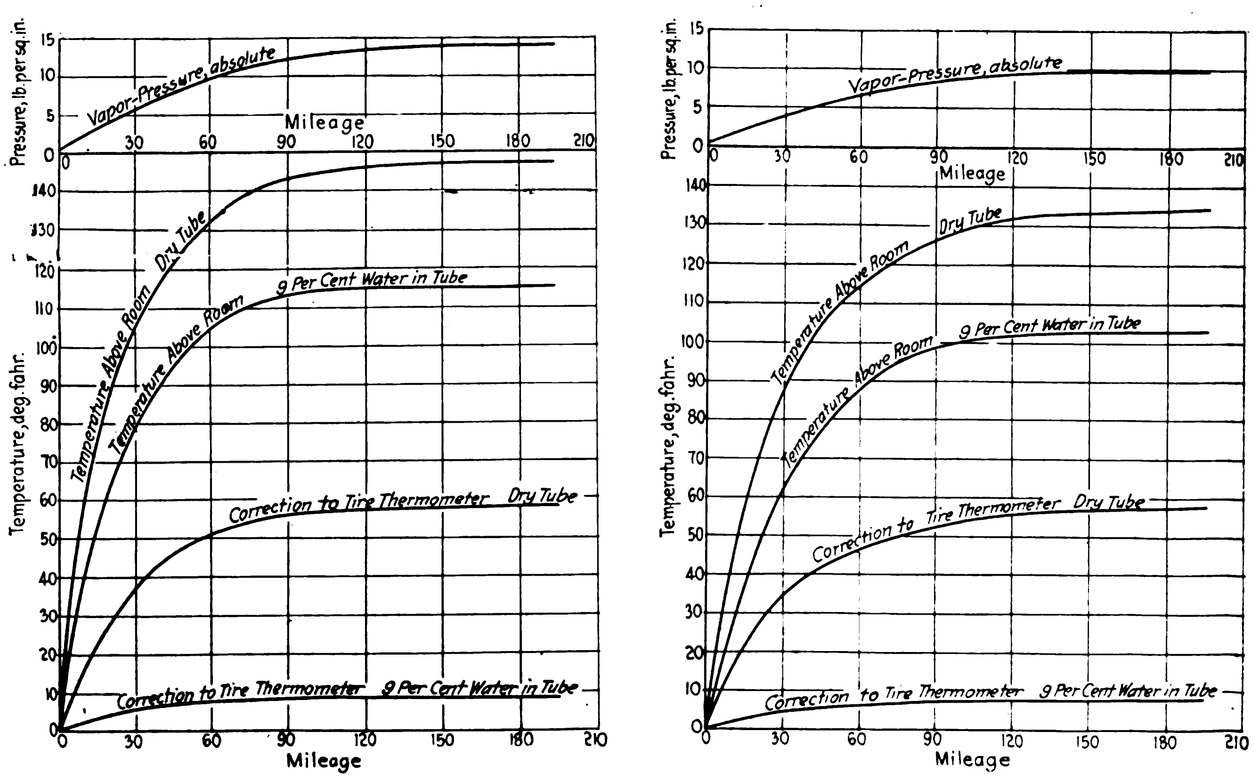

FIG. 14—CURVES SHOWING THE EFFECT OF WATER VAPOR IN REDUCING THE CORRECTION OF THE TIRE THERMOMETER

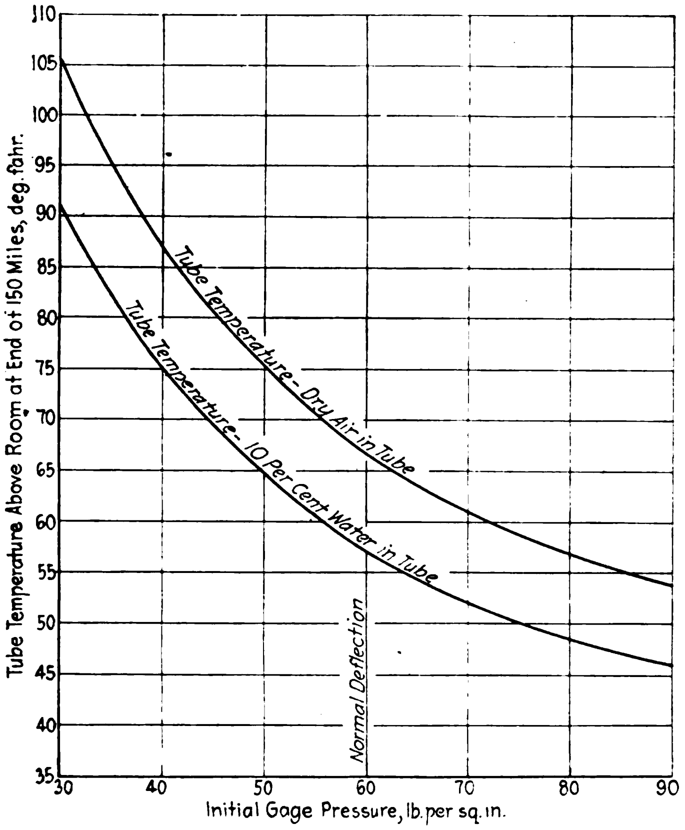

FIG. 15—TEMPERATURE EFFECT OF UNDER-INFLATION ON A SMALL TRUCK-TIRE WITH AND WITHOUT WATER IN THE TUBE

it caused in the reduction of the temperature of the tube. To prove this was not easy. I have already mentioned the large correction of the tire thermometers due to heat conduction from the thermometer well. With wet vapor surrounding this well instead of dry air the amount of heat delivered to it would be much greater; hence we should expect a large reduction in the correction of the instrument. This made it imperative to check very carefully, as already described, the amount of moisture in the air when we were supposed to be using dry air.

In Fig. 15 may be seen the serious temperature effects of under-inflation on dry and wet tubes in a 36 x 6-in. tire. Even for this size of tire the cooling effect of the water is very marked, but is not as great as with the 8-in. tire. Just how rapidly the temperature increases with the size of tire is clearly shown by Fig. 16. This is chiefly due to the increase in the thickness of the walls of the casing as the size increases. It is obvious from this chart that ordinarily no artificial cooling is needed for tires 6 in. or less in diameter, but as we go to 8 in., and above, anything that will keep the tire cooler should be used if it is not troublesome in its application. Putting a small amount of water inside a tube is a simple matter and there is nothing further to do about it until the tire is to be used in freezing climates, when the tube should be taken out and drained. There is no change in the riding qualities of the tire by using a small amount of water. There appears to be no objection to its use and the certainty of its cooling effect on the tube makes it deserving of a trial by all truck owners who operate large pneumatic tires at high speed over long distances in hot climates.

General Conclusions

From these experiments it may be deduced that:

(1) The measurement of the increase in the air pressure of a tire is a very accurate and reliable means of determining the average tube-temperature, provided proper precautions are taken to know whether the air is saturated and to have accurate measurements of pressure, volume changes, leakage and initial temperature

(2) Convection currents inside the hot tube are very large and may produce a large correction to any temperature-measuring apparatus that records only the particular temperature at some definite region, which is small in comparison with the whole cross-sectional area of the tube

(3) These convection currents are due to the variation in density of the air in contact with the hot and cool portions of the tube, combined with the force of gravity and that due to the rotation of the wheel. The first of these forces causes the convection currents to flow in the longitudinal direction of the tube when the tire is standing still, and the second causes much stronger currents in a transverse section of the tube when running

(4) Tire thermometers of the type described are not very well suited to any tube-temperature measurements except the initial one, on account of the corrections due to the convection currents and the still greater correction due to heat conduction to the cool part of the tube to which the well must be clamped

(5) Thermocouples placed inside the tube and mounted so that they will be held near the cen-

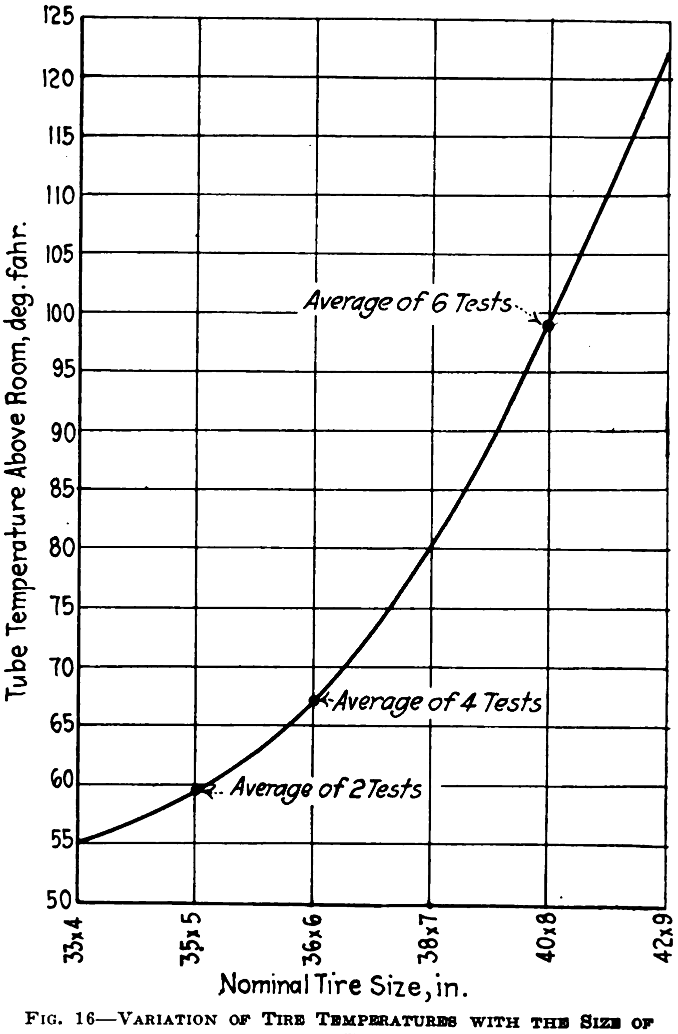

FIG. 16—VARIATION OF TIRE TEMPERATURES WITH THE SIZE OF THE TIRE

ter of the tube are rather frail and are likely to be injured or misplaced during the splicing of the tube. They are also subject to corrections due to convection currents and require considerable care and patience to make and install

(6) Detachable thermocouples of the type described are easily made and used. They are reliable when carefully used, and possess the very great advantage of giving fairly close to the correct temperature of the various parts of the tire that are hottest and whose temperatures are often most desired

(7) The detachable-thermocouple readings must be taken at very carefully measured intervals of time after stopping the tire as the hottest parts of the tire cool appreciably and rapidly immediately after stopping

(8) It appears likely that the detachable thermocouple should have a correction applied to its reading to compensate for the conduction of heat through the wires from the couple to the outside air. This correction has not been worked out completely as yet but will be greatest when this type of couple is used in holes of shallow depth

(9) The pressure in the tube does not begin to drop until a short time after stopping the tire, thus affording time for obtaining accurate pressure and volume readings

(10) The pressure-volume method combined with the detachable thermocouple one appears at the present time to be best suited to determining pneumatic-tire temperatures, as the results are more accurate, comprehensive and easily obtained than those from any other methods thus far developed

(11) For comparative tire-temperatures the results should be given in terms of temperature above the atmosphere surrounding the tire, as this factor is very important and variable. It is also essential to have the same loads, inflation-pressures, speeds, distances, supporting surfaces and sunshine in order to use comparative temperatures of different designs of the same size of tire

(12) The temperature of a tire increases very rapidly with its size due to the thickness of its walls

(13) High temperatures are destructive to tires but the relation between the life of a tire and its temperature for various compounds has not yet been established

(14) Small amounts of water, from 5 to 10 per cent of the air volume, used in the tube with the air result in a cooler tire, the difference in the temperature being marked in the case of a large hot truck-tire

(15) The use of water in tires of passenger-car size is not recommended, since their temperatures are not usually sufficiently high to justify it

(16) Care must be exercised when using water in the tube to drain the water from it before exposing it to temperatures much below the freezing-point

(17) All users of tires should note the rapid increase in tire temperature with the decreased inflation-pressure, so that they may prolong the life of their tires.

In conclusion, I desire to express the opinion that the study of pneumatic-tire temperatures is of much greater commercial importance and possibly of more scientific interest than is generally realized. A quick temperature-test of a tire may yield much more information than a long expensive test to destruction of the same tire. A temperature test is not, however, complete in itself for all tire testing, since tread wear, for example, requires another type of test, which, incidentally, can also probably be made better in the laboratory than on the road. Even after the correct temperatures of a tire shall have been determined, there is much research work yet to be done in order that the full significance of such information may be appreciated. This much, however, is sure: the rise in temperature of a tire is a direct and sensitive indication of the waste of energy due to its operation. The greater this loss of energy the faster the tire wears itself out and the more power is wasted to operate it. This loss can never be reduced to zero, but it can be kept low by selecting the most suitable materials and then very skilfully combining them according to the best design.

THE DISCUSSION

Ellwood B. Spear:—Only those who have endeavored to measure accurately the temperature of a pneumatic or a solid tire can appreciate the difficulties encountered by Professor Ellenwood. He is certainly to be congratulated on his success in his painstaking and important investigation. Tire manufacturers are making an intelligent and well-organized effort to give the consumer better value for his money. They will inevitably fail to get the best results, however, unless the consumer is ready to cooperate by taking proper care of his tires. I believe that one of the best methods of obtaining this cooperation is to demonstrate to the automotive engineer, the tire repairer and the consumer what happens to a tire in cases of overloading, under-inflation and fast driving. The determination of the temperature in the different portions of a tire offers a convenient means for the investigation of the effect of the design, as well as of the part played by the above-mentioned factors.

John P. Coe:—Engineers in the rubber industry are continually confronted with difficulties in scientific work due to the fact that the materials they are using are in a sense intangible compared with most of the materials used in the construction of automotive machines. Professor Ellenwood's work is all the more significant to tire engineers on that account. I have been very much interested in the data he has presented, as well as in the methods he has explained for measuring the temperatures developed, on account of the fact that we have experimented along very similar lines. Our research has not been as complete as Professor Ellenwood's, however, and is covered in most respects by his report. We use a slightly different arrangement for measuring the temperatures of the tire wall in that we attach our thermocouples to an awl point and are able thereby to insert them quickly into any part of the tire desired. We use small wire for thermocouple leads and solder the couple to the point of the awl. This outfit does not last indefinitely but is more durable than might be expected.

It has been very interesting to hear Professor Ellenwood's discussion of the effect of adding water to the air within the tube, inasmuch as we made this discovery ourselves without previous knowledge of his work. It is, however, evident from his report that he antedates us in the discovery.

The Car-Owner versus the Service-Man

By J. F. PAGE[1] CHICAGO SERVICE MEETING PAPER

STATING that there has been, and still is, too little mutual understanding, as well as too much bitterness and dissatisfaction between the automobile public and the service departments, the author outlines methods of improvement.

The service problem is considered under seven headings, which are amplified, and the service-man's problem is discussed with these headings as a basis, assuming that he has equipped himself with a trained personnel, space and adequate tool equipment and an effective system.

Fair and unfair service demands by car-owners are differentiated and a summary is made of some of the reasons that make satisfactory service impossible.

THERE can be no real and lasting benefit to either the car-owner or the service-man without mutual satisfaction in their business relations. If the owner expects perfection in the car and free service, the business of the service-station will not produce satisfactory results and there will be no profit. If the service-man is unreasonable, repair prices exorbitant and work imperfect, the public will neither patronize his shop nor buy the make of car represented by him. There has been and still is too little mutual understanding, as well as too much bitterness and dissatisfaction between the users and the handlers of automobiles. There is right and wrong on both sides.

The public gradually is learning that motor vehicles are complicated pieces of machinery, which are subjected to abuse and therefore require adjustments and repairs not covered by the purchase price. The average car-owner is slow to admit that a car should not run indefinitely with the addition of a little oil and fuel, and does not accept gracefully the normal repairs and loss of use of his car. On the other hand, the service-man must be brought to a sympathetic realization that the use of the car has become a necessity in the owner's life, and that the loss of the car's services may throw his business and social engagements out of joint. In the case of the truck-owner, the result may be a pecuniary loss. The vehicle, whether passenger-car or truck, must give a consistent performance at a reasonable cost to be satisfactory. These are generalities, but the need of patience and a realization of the inevitable on the one hand, and of proper equipment and a sincere effort to please on the other, are obvious, if the service-man is to accomplish his task and the owner is to be satisfied with his car and the service.

THE SERVICE PROBLEM

To get a bird's-eye view of the service problem, it is necessary to consider the demands of the car-owner and the requirements of the service-station to fill those demands satisfactorily. Let us first review the demands of the owner and the sources of service complaints. There are reasonable demands and unreasonable ones. Reasonable demands are susceptible of satisfactory fulfillment, and should be met by the service department efficiently and satisfactorily. Unreasonable demands make it impossible to satisfy the owner and are the cause of the service-man's "grief." The reasonable demands of the owner include:

(1) Courteous and intelligent treatment
(2) Proper mechanical work
(3) Quick service
(4) Dependable promises
(5) Fair prices
(6) Correct and understandable bills
(7) A convenient location of the service-station

Courteous and intelligent treatment means much. In an efficient uptodate service-station, the car-owner expects to deal with a high-class attendant who evinces a real interest in him and in his car. He likes to discuss repairs with a man who knows what he is talking about, and who will use judgment and intelligence in taking orders. He expects the service-man to know whether the owner wishes his car kept in perfect condition regardless of expense, or whether he wants to spend as little as possible to keep it in fair operating condition. He expects the service-man to take time to discuss his problems, and not to hurry away as though the owner's troubles were of little moment. He expects to be given prompt attention, not by a man with greasy hands who wears overalls, but by one dressed in clean clothes who has a pleasant smile.

Proper mechanical work is, of course, the primary object to be attained, but is advisedly placed second on the list of the car-owner's demands because he comes in contact with the service salesman first and looks for proper results from the shop as a secondary matter. The two are equally important; the success of each depends absolutely upon the effectiveness of the other, while both are essential.

The time has arrived when the passenger-car and the truck are absolute necessities of social and business life. Until the day comes when every car-owner will also own an emergency vehicle, it will be demanded that the service-station use every possible means to eliminate delays and speed-up repairs.

With quick service goes the demand for a promise on delivery so dependable that engagements can be made and kept. Dependable promises are more important, by far, than speed in turning out the work. After his first disappointment at the loss of use of the car, the owner will resign himself to the necessary adjustment of his engagements; but when the promised time for delivery has arrived, nothing else will satisfy him. The failure of the service-station to fulfill his promise is, in his eyes, an inexcusable and most annoying blunder, be the reasons what they may. The car-owner will not be satisfied with any "hard-boiled" policy of refusing a promise, nor does he wish to telephone several times about the delivery. The service-man should not talk of "estimated" time for delivery, because this means *the* time to the car-owner, just as clearly as if the service-man had said "promised" time. The car-owner will pay little attention to the technical difference between the words "estimated" and "promised."

Fair prices are a difficult question. The car-owner's ideas are not always in agreement with those of the service-man. In fact, the average owner has very little conception of repair costs. His ideas of fair prices probably

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would put the best repair-shop out of business in a short time. The owner who does not know something about the construction of cars is hard to convince that the same job may require entirely different lengths of time on different makes, due to differences of design or accessibility. Nevertheless, the service department must know the costs and do the work according to the best and cheapest methods if the maintenance charges are to satisfy the car-owner. A fair-minded business-man will expect the prices of repairs to be based on adequate equipment, efficient workmen, good materials and business-like administration.

Correct billing is essential, yet the collection of cost data in a large shop is a prolific source of trouble, resulting in errors of both commission and omission. The terms used by the shop are Greek to many owners. The result of doubt regarding the correctness of the charges is slow pay, claims and adjustments, collection costs and much dissatisfaction. Billing, correctly and promptly done, is necessary to keep the car-owner satisfied and avoid troubles and complaints. This question opens the subject of estimates, flat prices, cash on delivery and the like. Sooner or later a definite policy must be established for repair work in the motor-car industry and the sooner the better. The present lack of uniformity of action and thought by the motor-vehicle distributor and the repair-shop manager is keeping the public in a state of discontent and uncertainty.

Convenience in regard to the location of the service-station has always been important to the owner, but it is a difficult problem for the distributor. Salesrooms as a rule are located in congested districts, where space is high in price and small in size. Many distributors have found it necessary to enlarge their service spaces to such an extent that they have been forced to locate the service-station at a distance from a point central and convenient for the owner. Inconvenience of location may result in repair work being taken to outside shops which are not always so well equipped to do the work as the distributor's own service department. In order to compensate for the inconvenience of location the service-station must give so much better service that the car-owners will be willing to travel the greater distance.

The Service-Man's Problem

Corresponding to the fair-minded owner is the conscientious service-man who knows what the customer expects and sincerely tries to give 100 per cent of satisfaction for every dollar he receives for service. His problem is to provide the three essentials, without which the demands of the owner cannot be met successfully: (*a*) trained personnel, (*b*) adequate space and tool equipment and (*c*) an effective system.

The first person with whom the car-owner comes in contact will be the service salesman. Psychology and human nature are brought out in the dealings of the owner with the service department. To the customer the service salesman is the company and the service will be looked upon favorably or otherwise, depending upon the impressions made by the service salesman perhaps more than upon any other one thing. It is the more or less intangible things that affect the owner's opinion as much as the actual job performed by the shop. The efficient and successful service salesman is a man who has had practical experience and possesses qualities of even temperament and tact. He must be a real salesman, continually selling the owner both the car and the service, and doing so very often under adverse conditions. No two persons are alike. The service salesman must be a student of human nature in addition to his other qualifications and meet the customer's demands for courteous and intelligent treatment. He must know whether the car-owner with whom he is dealing wants perfection, regardless of cost, or the least work at the least cost. Nearly every owner has particular whims and ideas of adjustments. Only by catering to these will the service-man please his customer. How often a shop makes standard adjustments, only to find that the owner is dissatisfied because the adjustment is not one to which he is accustomed! The service salesman has a real problem and many new-car sales depend upon his manner of handling it.

Courteous and intelligent treatment is expected from everyone. The grease boy has as much to do with the customer's impressions as the general manager, often more. A clean shop, well-kept tools, firm but fair supervision and fair pay will bring out the best from the men, but no consistently good work can be expected from dark and greasy workrooms and the worn-out equipment.

Good mechanical work calls for proper tools and skilled hands, with proper supervision. Tools alone will not accomplish the task, nor skilled men without proper tools. The shop force must be supervised by careful and experienced men, who must train the mechanic and instill in him the spirit to turn out good work. Spirit has as much to do with an efficient shop as the skill of the men, which is useless unless backed by the right mental attitude toward the customer and the work.

For quick service, there must be ample space in which to handle cars, adequate equipment, and special tools and devices. It is astonishing how much time can sometimes be saved by a special tool or method. Too many special tools, however, easily may prove to be a hindrance. More time may be wasted in going after and making ready some special tool than might have been required to accomplish the purpose with the tools at hand. A shop equipped to do rapid repair-work necessarily will have idle equipment, because a tool which may be necessary for a certain operation may not be called for again for some time. The shop in a large city has a great advantage over a country shop because of the varieties of special work which may be performed in a moment; whereas the country shop might have to wait a long time. Quick service is dependent largely upon the simplicity and accessibility of the units in the vehicle. The service-man's burdens would be lightened greatly and the owner's satisfaction be accomplished more nearly if accessibility were given the study and importance it deserves. Nothing conduces more to quick service, proper work, dependable promises, fair prices and general satisfaction. Quick service cannot be accomplished unless the service-station has a complete line of parts for all models. The stockman's task is one of the most important in the service organization because, in addition to the necessity for a complete line of parts, there is the equal need for a sufficient turnover and economical administration to enable the department to be operated at a profit. The stock department of a main branch-station should not be considered satisfactorily efficient if less than 97 per cent of orders can be filled upon demand.

Dependable promises can be given in 90 per cent of the cases handled in an efficient service-station. There is no excuse for failure if there is a proper system of routing and following work through the various operations and departments, and the right kind of cooperation between the shop and the service office. Fair prices will depend upon the general efficiency of the shop, reasonable prices for parts and the right proportioning of overhead

expense but, in addition, prices will be influenced very largely by the accessibility of the various units in the car.

The lowest costs and fairest prices cannot be obtained with an indefinite or indifferent attitude on the part of the management. The man in the shop is human and, unless he is given a good reason for exertion, will be inclined to take things as easily as possible. A time limit must be established for each operation, and constant study must be made of ways and means to reduce it. Every item of expense and hour of labor must be accounted for if the true costs are to be known. Prices must be based on costs, and not on mere competitive price-cutting. Let fair prices to the customer be based upon efficiency in one's own shop, and not upon cutting the throat of a competitor.

Correct billing needs the attention of the service-man who hopes to collect his money promptly and in full, and at the same time eliminate the source of a very large percentage of complaints. The bill for service must be clear. The operations performed must be expressed in terms easily understood by a man of average intelligence. The materials and time should be listed clearly with the prices of each unless the prices of complete operations have been quoted in advance. The bill should be ready when the final inspection has been made. The customer should look it over at the time of delivery, when all details are clear and fresh; later they will be confused, complicated and half forgotten.

Unfair Service Demands

There is no demand of fair service that cannot be met successfully by an efficient organization. Unfortunately, all men are not easy to deal with and some are not satisfied because they expect the impossible. Some of the reasons which make satisfactory service impossible are as follows:

(1) The owner who has been oversold by the new-car salesman and comes to the service-man with demands for free work or adjustments which cannot be met; or the owner who expects his car to be something more than standard, a car designed in his own imagination as a "super" car

(2) The man who, upon entering the service-station, expects a salesman to be instantly available, no matter how busy the day, and to spend unlimited time in talking over not only business matters but unrelated subjects

(3) Complaints by the car-owner of high costs and exorbitant charges, founded not on a conception of actual costs but on intangible ideas of what the work should cost

(4) The man who thinks and acts as if a chip on his shoulder and a scowl on his face were necessary to obtain proper service

(5) The owner who expects something for nothing; who argues over the bill or complains about the work, hoping to get a reduction

(6) The very busy person who expects repairs or adjustments to be made in much less time than it is possible to do so

(7) The man who abuses his car, and blames the car for the service

(8) People who make no allowances and expect perfection in a line of business that is exceedingly liable to errors and misunderstandings. The service-man is not infallible, be he ever so skillful and conscientious

Although we have still much to learn and many improvements to make, we are endeavoring to carry out in a rather large way the things that make up satisfactory service. They include three essentials: personnel, equipment and system. I hazard the opinion that the public is becoming more motor-wise and that competition will force improvements in service as well as in production. It behooves those who would have lasting success to include in a car the features that make for accessibility, low repair-cost and quick adjustment, and to organize the service-station along business-like, uptodate lines.

THE DISCUSSION

President Bachman:—The request has been made by Past-President Manly that I outline our system of unit rebuilding. It has something to do with the so-called flat-rate billing that Mr. Page touched upon. Nothing that I may say should be interpreted as antagonistic to such a method, because I am thoroughly in sympathy with it. I think that I was one of the first, in our organization at least, to advocate such a method. However, I think that those who have had my experience with it will understand that it is not wholly satisfactory and that there are problems connected with it, the same as with everything else. As a result of work of this character, the engineering department has obtained some excellent ideas that we have been enabled to embody in later designs and to rearrange current designs in a way that make possible the handling of every vehicle by component parts and units. Mr. J. Whyte mentioned that he was pleasantly surprised to learn that a good service job is an economical production job, and I believe that this is true.

We are in the commercial-vehicle business exclusively. If a man comes in with a car that requires repairs and he needs the car promptly, we have an arrangement whereby we can interchange the unit that causes the trouble with a rebuilt unit. This unit has been put into a condition that we are prepared to guarantee will give the equivalent service of a new unit. Then we take the defective unit and repair it. We have a gang doing nothing but that sort of work; that is, we have specialists on engines, on transmissions and the like, throughout the truck. The complete carrying out of such a system is possible only at large centers; it could not be carried out completely in smaller places. We believe that this kind of service justifies a premium and we make an arrangement whereby the customer understands exactly what the cost of it to him will be. He has the privilege of accepting that or taking another form of service in the repair of his vehicle with a dependable promise as to the time of completion, so that he can arrange his business affairs accordingly.

C. M. Manly:—I hoped you would bring out something about the relative cost of new production and a complete overhauling of the units in that way. The point I have in mind is that if such data can be placed in a really accessible form and circulated, they would have great influence on the car-owner in enabling him to understand some of the real fundamentals that go to make up this question of cost of service. For instance, suppose a man knows that a rear axle costs $100 to $150. If he finds that it will really cost a very large percentage of that sum for the garage-man to take the axle down, look it over thoroughly, put in a few new parts and reassemble it, because it is a special job, and it is really an honest charge, I think it would help to remove much of the friction that Mr. Page has described as being bound to arise in connection with the question of cost of repairs, due to the lack of understanding of the public of the difference between manufacturing and overhauling.

PRESIDENT BACHMAN:—While I cannot answer that question definitely, I think I can give a relative answer. From my correspondence with our service people, I understand that they have found that the additional cost of that method generally has been considered by our customers a satisfactory and reasonable charge for the decrease in time that they are without the vehicle. Of course, that includes many factors.

W. O. HINCKLEY:—What can a service-man who is probably servicing from 20 to 30 makes of car do on that kind of proposition? If he must invest in extra axles and the like to take care of that proposition, it might mean an investment that would run up to $90,000 for overhauling various makes of car. From the service point of view, I do not know how a man could make reasonable charges and carry an overhead such as that would necessitate.

PRESIDENT BACHMAN:—I tried to cover that point on the basis that the efficient handling of such a system is dependent upon volume. If there is only a turnover of a unit once monthly, once in 2 months or something of that sort, the essential value of which runs up into a considerable amount of money, the overhead charges would increase to a point where the system gets out of line. I am dealing entirely with the handling of one make of vehicle in our own service organization. Even there we have to confine the plan largely to those territories where we have a sufficient volume to make it practicable.

WALTER E. ROBERTSON:—I have devoted much study to the subject of automobiles. It seems to me that the principal object in the design of the car has been to make it sell, and that service and economical repairs have been overlooked. With some associates, I am entering the field with an automobile part that we have been experimenting with for a long time. We might have put it upon the market a long while ago but we found a defect in our experimental work and had to scrap many of our patterns. The construction seemed to be perfect and simple at the time, yet we would find opportunities for improvement that were especially to the interest of the user. That has been the principal thing that has retarded our progress. From my observation, insufficient consideration is given to the man that ultimately will experience the troubles.

The expense of upkeep and repairs is tremendous. A man does not realize what the difficulties are in gaining access to the various parts that must be removed or repaired. The commercial side has been given more thought than has been given to the needs of the man that will use the car. The builders should think more of the man who must use the car, of its economical repair and of accessible ways of repairing it. An automobile seemingly is susceptible to getting out of order at the most inopportune time. If the builders would circulate instructions to the users in regard to how certain parts can be removed, how they can be repaired and what time is required for such repairs, it would help considerably. It seems that the trend now is to give more thought to the consumer, which is a tendency that I am glad to see coming to the front.

GEORGE S. CAWTHORNE:—A neighbor of mine overhauled his car in his own garage because the treatment he had received from the service-station had soured him. He asked me where he could buy replacement parts elsewhere than at the regular service-station. This car is very well known and very well liked so far as new cars are concerned, but it seems that the service given by those who handle it is not what it should be. That ought to be overcome very easily by courteous treatment. I think we are coming closer to that as these problems develop.

DAVID BEECROFT:—As an owner who drives his own car and pays his bills, I have to go into certain places for service. I deal with seven men previous to dealing with the man who actually does the work. That is a tremendous waste of time. I deal with metropolitan service-stations and I have often thought how much better it would be if the service representatives considered the psychology of the business man; sometimes they seem to be entirely neglectful of that consideration. The service people seem to have no realization that the time of a business man is of value, and the station is not organized to take care of its patrons properly. That is true not only of cars but of batteries and parts.

A wealthy gentleman who had been buying a certain make of car told me recently that he would not buy that make of car any more because he could not get satisfactory service. He was buying purely on the basis of service. He gave less weight to the matter of design. Whenever we make service a difficult problem we are building a barrier of sales resistance.

An official of possibly the greatest mail-order house in this Country told me recently that, in analyzing its business, the greatest asset was believed to be constituted of the innumerable acts of company service that had been performed for a period extending over a generation. Also, that the service the company had rendered, just as much as the quality of the goods, had built up the business. As an actual instance, about 17 years ago a set of dishes was bought from this company and the buyer was assured at the time that the pattern, an European one, would always be in stock. The buyer wrote recently to see if some of these dishes that had been broken could be replaced. The company replied that, due to the war, the same pattern was no longer available; but, because it had agreed that this pattern would always be in stock, it would take back the old set of dishes and give the buyer another set of standard pattern. That is service, and such service is what built up that great organization.

In a communication addressed to Herbert Hoover, Secretary of Commerce, regarding the automobile industry, the dealers of the Union of South Africa said that they insist on greater standardization. They are 6000 miles from the factories and have experienced the difficulties of carrying too much stock. They use motor vehicles for transportation. They are not luxuries. They said to the builders in America that to sell automobiles in the Union of South Africa in the quantities that the people there want them, the machines must be standardized and it must be made easier to service them and carry the parts in stock. Requests of that nature are coming in from other sections.

The automobile is a utility vehicle. Whenever we ponder that there are a certain number of people in the Country with salaries sufficient to enable them to own a car of a certain quality, we are looking entirely away from the real market. The sale of automobiles cannot be gaged on the basis of the salaries of the people of the world. In 1885 my father kept two road horses on our farm. We had about 20 horses, but these 2 horses did nothing but take us to church or to market and do miscellaneous road work. Those horses were never used to plow or harrow or do similar work. My brother lives on the same farm today and for years has had his automobile and his motor truck. They are just as essential and constitute the same economic portion of his home as did

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The Measurement of the Property of Oiliness[1]

By Robert E. Wilson[2] and Daniel P. Barnard, 4th[3]

Annual Meeting Paper *Illustrated with* Photographs and Ch rt

THE term "oiliness" is defined as that property of lubricants by virtue of which one fluid gives lower coefficients of friction (generally at slow speeds or high loads) than another fluid of the same viscosity. Its importance under practical operating conditions is shown to be greater than is generally recognized. Unfortunately, however, no satisfactory method has ever been developed for the quantitative measurement of this property in comparing different lubricants.

The paper describes the variety of possible methods of measuring the property of oiliness and of throwing light on the mechanism of partial lubrication, including (*a*) the use of a Deeley-type machine to measure coefficients of friction between plane surfaces at slow speds; (*b*) a refined and reproducible method of determining static coefficients of friction between partially lubricated metal surfaces; (*c*) the measurement of the interfacial energy between oil and mercury; (*d*) measurements of the electrical resistance and the rate of formation of adsorbed films on metal surfaces; and (*e*) the clogging of fine metal capillaries through which lubricants are forced. Some other interesting preliminary experiments also are described.

In the light of the results obtained by the above-mentioned methods, it is believed that the static-friction test, with proper refinements, is the best single measure of the properties of oiliness, but that it should be supplemented by measurements of the thickness of the adsorbed films at high pressures, in order to throw more light on the mechanism of the action of different constituents in lubricating oils.

Animal and vegetable oils are almost invariably superior in oiliness to straight mineral-oils. The blending of considerable proportions of these neutral glycerides with mineral oils greatly improves their oiliness, but the same results may be accomplished by adding much smaller proportions of other materials, such as fatty acids or oil-soluble soaps.

These experiments confirm entirely the customary hypothesis that oiliness is due to selective adsorption of constituents in the oil by the metal surface, but the common conception of a mono-molecular adsorbed film that acts merely by masking the attractive forces of the metal surfaces for one another appears to be incorrect. The adsorbed film is shown to be of colloidal rather than molecular dimensions, is a plastic solid rather than a fluid-film, and apparently acts by smoothing over surface irregularities, carrying much of the load, and minimizing metal-to-metal contact and abrasion. The structure and physical characteristics of this film seem to be more important than its thickness in determining its efficiency in lowering friction.

The constituents of lubricants that form these adsorbed layers can be selectively adsorbed and largely removed from the oil by repeated treatments with very finely divided metals, such as iron-by-hydrogen.

IN their previous article the writers have shown that under certain adverse conditions—especially at slow speeds and high loads or where the oil supply is inadequate—a perfect fluid lubricating film cannot be maintained, and lubrication of the rubbing surfaces depends on the ability of a very thin, probably semi-solid, film of lubricant to adhere to the metal surfaces in spite of the pressure tending to squeeze it out and the abrading effect due to motion. This fundamental property of "oiliness," which is possessed in variable degrees by all good lubricants, has been the subject of a considerable amount of investigation, chiefly in England; but, although there is considerable evidence to indicate the presence of some such adsorbed film, its composition, structure, thickness, and the mechanism of its formation, are matters which still remain to be established. Furthermore, no simple and satisfactory *quantitative measure* of the property of oiliness has yet been devised.

For the purposes under discussion, the "oiliness" of a lubricant will be defined as the property by virtue of which one fluid gives lower coefficients of friction (generally at slow speeds or high loads) than another fluid of the same viscosity.

One reason why the property of oiliness has not received the attention it appears to deserve lies in the fact discussed in the previous paper; namely, that practically all well-designed journal bearings operate for at least 95 per cent of the time under conditions of perfect fluid-film lubrication, where the oiliness of the lubricant plays no part. As a result, many are inclined to question the practical value of a detailed consideration of the more exceptional and complicated case of *partial* lubrication, where the speed is too low or the load too high to maintain the normal fluid-film between the metal surfaces, and the property of oiliness becomes important. Since this point of view is frequently taken by experienced lubrication engineers, it seems desirable to point out the following basic reasons why the laws of partial lubrication and the property of oiliness are of very real practical importance, apart from their theoretical interest.

(1) Every bearing must occasionally start and stop, and thus pass through the region of partial lubrication, where the friction coefficients are enormously higher than at normal speeds and some abrasion is certain to result. The resultant roughening of the bearing surfaces is directly or indirectly responsible for a large proportion of bearing failures, although the actual failure may take place only after the bearing has been running for some time at high speeds, when the rough surface causes overheating. By using a lubricant of high oiliness that will maintain a film even when the bearing is not rotating, it is possible to minimize this abrasion and its resultant effects

[1] This is the second part of the paper on The Mechanism of Lubrication, by the same authors, that was presented at the 1922 Annual Meeting of the Society. The first part was printed in the July, 1922, issue of The Journal, p. 49.

[2] Director of the research laboratory of applied chemistry, Massachusetts Institute of Technology, Cambridge, Mass.

[3] Research associate, Massachusetts Institute of Technology, Cambridge, Mass.

(2) Conditions very frequently occur, especially in machinery in the hands of unskilled operators (under which head we may include most automobiles) where the oil supply is temporarily deficient for one reason or another. Under these conditions the only salvation of the bearing is a tenaciously adsorbed film of lubricant which will remain for a long time and prevent seizing and damage to the bearing, even though it gives a higher coefficient than that of the fluid-film which should be present

(3) In many cases, such as the piston-rings in an engine cylinder or the crosshead of the steam engine, there is a reciprocating motion between flat surfaces and the maintenance of a perfect fluid-film under these conditions, especially at the ends of the stroke, is very difficult, and the amount of friction depends very largely on the permanence and lubricating value of the adsorbed film. The precise type of lubrication in the cylinders is, to be sure, not so important in many of the more poorly made engines where, to quote a well-known contemporary, "the threads on the piston-rings do not fit those in the cylinder"

(4) There are many special uses of lubricants where high pressures and comparatively slow speeds are absolutely essential, under which conditions only a strongly adsorbed film can prevent seizing of the surfaces. Important examples under this head are the lubrication of gears, where greases are usually employed to maintain a lubricating film, and in cutting or threading most metals, where lard oil is unquestionably superior to ordinary mineral oils

(5) From the standpoint of the producer of lubricants, a study of the film-forming tendency of oils appears to be indispensable, since this unmeasured oiliness factor is the only fundamental difference between good lubricants, poor lubricants and so-called non-lubricants of similar viscosity. Under these circumstances, it is surprising that neither the producers nor the consumers of oils have developed any recognized test or specification to tell the difference between the amount of oiliness possessed by different lubricants, even though such tests might hamper the remarkable imagination and fluency of the average oil salesman when discussing this phase of the subject

(6) Probably the most important way in which the property of oiliness affects the power losses in practically all bearings, even when operating under conditions of perfect fluid-film lubrication, apparently has never been pointed out. This is the influence that the property of oiliness has on bearing design

As has been shown in the previous paper, bearings are designed primarily to operate in the range of perfect fluid-film lubrication and reasonably near to the point of minimum friction. Unfortunately, it is not possible to operate exactly at this point because it is just on the verge of the dangerous condition of partial lubrication. It is therefore necessary to multiply the value of $z\ N/p$ at the critical point by a factor of safety of, say, 5 to determine the safe operating value, though these conditions give much higher coefficients of friction than the minimum value.

The effect of the oiliness of the lubricant on this matter of design is apparently two-fold; first, it lowers to a considerable extent the critical point[4]; and, second, it decreases the danger attendant upon operating occasionally in the region of partial lubrication. It is therefore quite reasonable to use a smaller factor of safety if a lubricant with a high degree of oiliness is assured. In other words, a bearing supplied with a very oily lubricant could be designed to operate at a value of $z\ N/p$ determined by multiplying a *lower critical value* by a *lower factor of safety*, thus giving a *much* lower operating coefficient of friction than if reliance could not be placed on the degree of oiliness possessed by the lubricant.

In view of the foregoing facts, the research laboratory of applied chemistry at the Massachusetts Institute of Technology believes that the whole question of oiliness and the mechanism of lubrication at low speeds is the most important present problem in the entire field of lubrication, from both a practical and a theoretical standpoint. As the first point of attack upon the problem, the laboratory has been endeavoring for some time to work out satisfactory methods of measuring this oiliness factor, and has also been conducting experiments designed to throw light on the mechanism by which the film is formed.

In this preliminary investigation of the factor of oiliness we offer no apology for the fact that we have not as yet attempted to simulate actual working conditions in bearings and the like. As has already been pointed out, the factors which determine the coefficient of friction in a bearing are many and are separable only with difficulty. Theoretically, under certain conditions a bearing could be used to measure the *viscosity* of a lubricant; practically, the only way to obtain a satisfactory degree of accuracy is to use a viscosimeter that bears no resemblance to a bearing; similarly, in developing the quantitative measure of *oiliness* it is desirable to go to extreme conditions and eliminate all other factors as far as possible. It is also our belief that undue reliance should not be placed on any single supposed test of the oiliness factor in the present preliminary stage of the work, but that several radically different methods of measurement should be tried to determine which results are most uniformly consistent with the known properties of the oil and its behavior in actual service tests in bearings operating under conditions of partial lubrication. This comparison of the results obtained by different methods of testing should also aid greatly in explaining the mechanism of partial lubrication.

Thus far we have made use of six independent methods of measuring some property connected with the adsorbed lubricating film on metal surfaces, not to mention several others that have been tried out and discarded for various reasons. The results of these preliminary studies are admittedly inconclusive in many important details, but a number of new and interesting facts have been brought to light and it is hoped that a detailed presentation of the methods used may stimulate discussion and research in this important field.

In view of the specified definition of oiliness in terms of the coefficients of friction for partially lubricated surfaces, measurements of the coefficient of friction under conditions which emphasize the effect of oiliness are the obvious first approach to a solution of the problem. This means that the measurements should be made in the region of partial lubrication, by using either very low speeds or high pressures.

In undertaking such a study, it is well to recognize at the outset one very serious and apparently inescapable difficulty that besets any method of measuring friction under conditions of partial lubrication. This difficulty is due to the combination of two facts: first, in this region the nature and condition of the surfaces profoundly influence the coefficient of friction; and, second,

[4] As indicated in the preceding paper, insufficient work has been done to settle this point definitely.

some abrasion is certain to result if the test is severe enough to distinguish clearly between good and bad lubricants. It is therefore necessary to use extreme care in the preparation of the surfaces if reproducible results are to be secured, and the surfaces must be refinished and brought back to their original condition after virtually every test. As a result, the time spent in lapping and repolishing surfaces is generally a matter of days, as compared with minutes required for making the actual friction measurements. This results in slow progress, especially in the early stages of a fundamental investigation where the relative importance of the various factors and the proper methods for obtaining the best results are not yet established. This difficulty is even more pronounced in working with journal bearings, because any repolishing or scraping tends to increase the clearance and change the whole behavior of the bearing. The use of flat bearing surfaces for such tests is therefore decidedly advantageous.

Results on a Deeley-Type Friction-Machine

The most promising method described in the literature for studying the mechanism of partial lubrication at slow speeds and comparatively high loads is the Deeley type of machine.[5] Apparently only one of these machines has been constructed previously, that by Mr. Deeley himself, and the meager amount of data secured was insufficient to permit the formulation of important conclusions.

[5] The Deeley-type machine and its use are described in some detail in the 1920 Report of the Lubricants and Lubrication Inquiry Committee of the British Department of Scientific and Industrial Research.

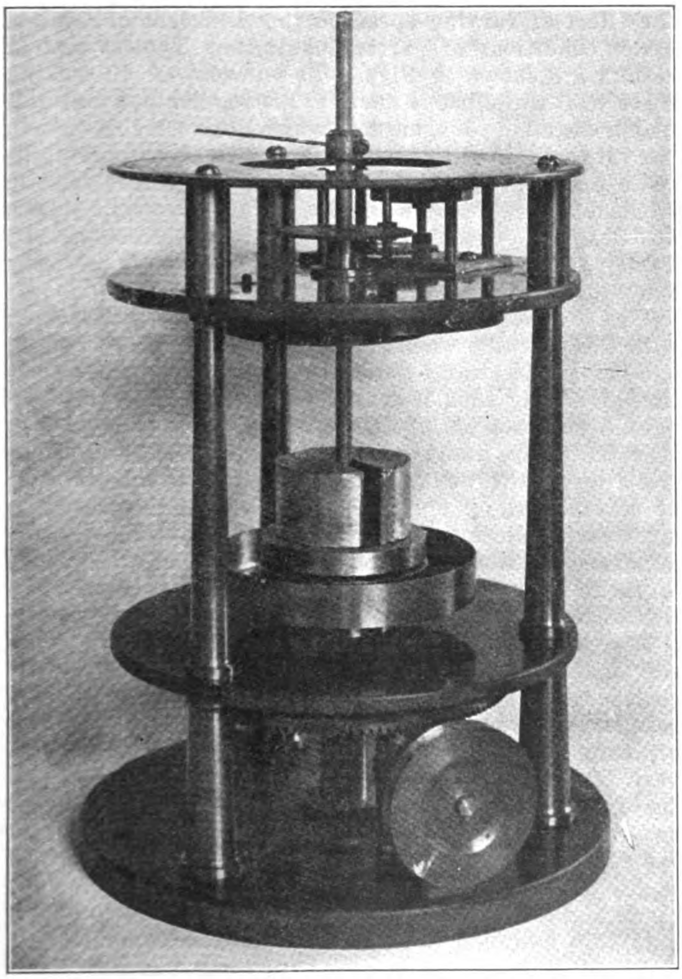


Fig. 1—Deeley-Type Friction-Machine Used in Studying the Mechanism of Partial Lubrication at Slow Speeds and Comparatively Heavy Loads

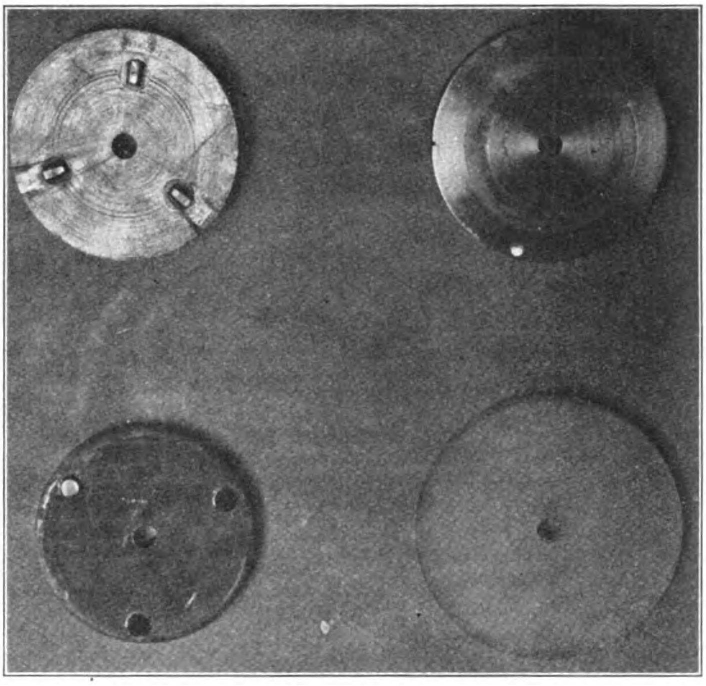


Fig. 2—The Two Essential Elements of the Deeley-Type Friction-Machine, a Flat Plate and the Disc, Which Is Supported by Three Pegs, That Rests Upon It

The construction of the Deeley-type machine used by this laboratory is shown in Figs. 1 and 2. It consists essentially of a flat plate that can be rotated at any desired (slow) speed, on which rests a disc supported by three pegs and pressed down by weights of any desired size. The disc is held in place by a small vertical shaft that is free to turn except for the restraint imposed by a calibrated spiral spring. When the bottom plate is rotated at constant speed, the top disc follows it until the spring overcomes the frictional resistance and slipping begins. The first maximum reading on the graduated scale measures the static coefficient of friction, and the steady position that it assumes as rotation is continued is a measure of the kinetic coefficient at the speed in question. A damping device is added to prevent too violent oscillations of the system when the disc first breaks away.

The general procedure in making measurements is as follows: When glass surfaces are used they are first cleaned by washing with soap and water and then with chromic acid. They are next rinsed thoroughly with distilled water and dried on a clean cotton towel. They are then placed on the machine, the receiver filled with oil, covering the bottom plate to a depth of 1/16 in., and the machine run at high speed and light load for several minutes to rid the surfaces of any adhering moisture film. In making a series of readings the machine is adjusted to the speed required, the load is applied, and as soon as the pointer comes to rest the reading is noted and the machine stopped. In this way undue abrasion is prevented, as the pointer almost always comes to rest after one or two revolutions of the turntable. By the time that a half a dozen readings can be taken, however, the surfaces are so badly scored that it is necessary to refinish them before they can be used again. This rapid wear affects the results considerably, but thus far no way has been found of avoiding it. For this reason the values given below must be considered only approximate.

In using metal surfaces the procedure followed is some-

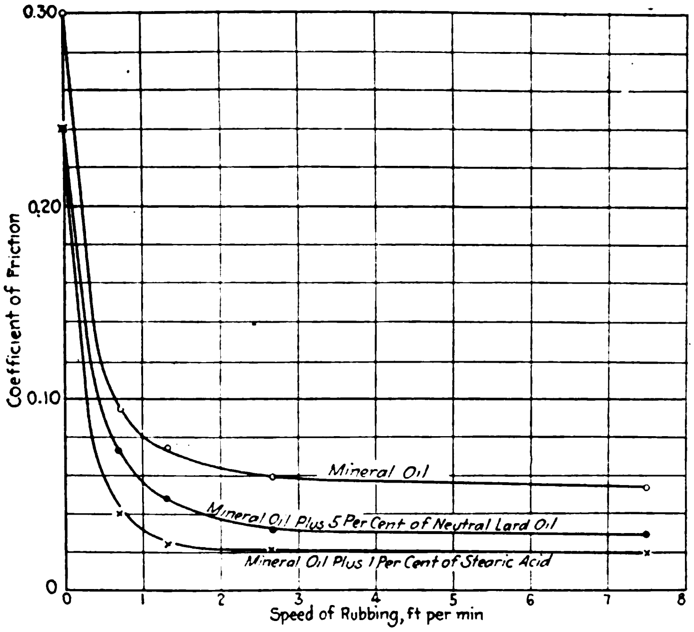


FIG. 3—RESULTS OBTAINED WITH GLASS RUBBING SURFACES AT A PRESSURE OF 300 LB. PER SQ. IN.

what different. The plate used consists of a case-hardened tool-steel disc which, after hardening, is ground true and polished. The tripod is made of cast-iron and has rectangular feet, approximately 1/16 x 1/4 in. They are placed so that the longer dimension is at right angles to the path of travel. These surfaces are ground together first with carborundum flour and then run dry for several days until both surfaces are bright and smooth. They are next oiled and run-in for about 50 hr. more. Following this treatment the surfaces are removed from the machine, carefully cleaned and oiled with the oil to be tested, and the machine run till constant readings are obtained. This is frequently a period of from 8 to 10 hr.

One of the principal questions that it was hoped to settle by using the Deeley machine was whether the coefficient of friction drops off steadily with increasing speed, or whether it first rises from the static value and passes through a maximum at very slow speeds. A statement to the latter effect is made by Archbutt and Deeley[6] and repeated by several other recent writers on the subject, but the experimental data on which it is based seem to be extremely meager[7] and, in view of the difficulty of obtaining reliable results under conditions of partial lubrication, it was felt desirable to check up this statement which seems contrary to both theory and general experience. It was also felt that a study of the static coefficient and the kinetic coefficient at slow speeds would throw considerable light on the best conditions under which to obtain a quantitative measure of the property of oiliness.

Using the above-described machine, an extensive series of observations has been made to determine primarily the effect of varying speed on the coefficient of friction under conditions of partial lubrication. Typical sets of results are shown in Figs. 3 to 6 inclusive. In making these tests an effort is made to distinguish between animal or vegetable oils, most of which are known to possess the property of oiliness in a high degree, and straight-refined mineral-oils of similar viscosity, of which Velocite B, made by the Vacuum Oil Co., is fairly typical. It was also desired to check up the observations of Archbutt and Deeley on the effect of adding small amounts of fatty acids to the mineral oil.

Fig. 3 shows some of the results that were obtained with this machine when using rubbing surfaces of glass, and brings out clearly the effect on the coefficient of friction of adding small amounts of fatty constituents to a light, high-grade paraffin-base spindle-oil. It it noted that over the entire range of rubbing speeds covered, from static up to 8 ft. per min., the oils containing 5 per cent of neutral lard-oil and 1 per cent of stearic acid gave much lower values for the coefficient than did the straight mineral-oil. Stearic acid to the amount of 1 per cent serves to lower the coefficient even more than does 5 per cent of the neutral animal-oil.

The values shown for the static coefficient are undoubtedly inaccurate, due to the great tendency of the well-cleaned glass surfaces to seize and give widely variable results. Glass also does not appear to adsorb a film from ordinary lubricants at all firmly, and hence it gives a very high static coefficient. The glass surfaces used in these tests all showed very deep scratches after one or two readings had been obtained. The very rapid drop in the coefficients as soon as appreciable speeds are reached is also consistent with what might be expected from such plane highly polished surfaces that would require only a very thin fluid-film to prevent all solid contact.

Fig. 4 shows the variation of the coefficient with speed of rubbing for a given oil, rapeseed, at two different temperatures. It is interesting to note that at low speeds the higher temperature gives the higher coefficient, while

[6] See Lubrication and Lubricants, by L. Archbutt and R. M. Deeley, p. 60. Kimball found such a case in some "decidedly anomalous" results on steel against steel surfaces.

[7] See A New Investigation of One of the Laws of Friction, by A. S. Kimball, *American Journal of Science and Arts*, vol. 13, p. 353; also, Journal Friction at Low Speeds, by A. S. Kimball, vol. 15, p. 192

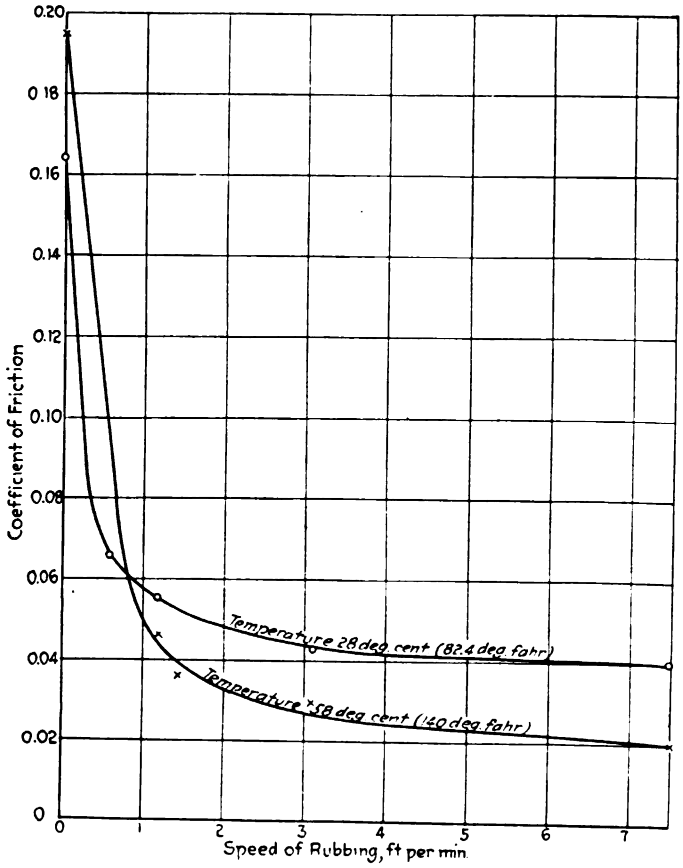


FIG. 4—VARIATION OF THE COEFFICIENT OF FRICTION WITH CHANGES IN THE SPEED OF RUBBING

In This Test Blown Rapeseed-Oil Was Employed with Glass Rubbing Surfaces That Were Subjected to a Pressure of 300 Lb. per Sq. In.

at speeds above 1 ft. per min. the two curves reverse their relative positions. This is readily explainable on the basis of the fact that adsorption is the main factor in reducing the coefficient at lower speeds, and this is always more pronounced at lower temperatures; whereas, at higher speeds, the viscosity of the fluid is the primary factor in determining the friction, and this is lower at the higher temperature.

In Fig. 5 the effect of adding small amounts of stearic acid to the mineral oil is shown, using cast-iron against steel surfaces. The presence of but 0.5 per cent of the fatty acid apparently lowers the coefficient to less than one-half of its former value in the range studied. Increasing the fatty-acid content to 2 per cent effects a further, though less marked, reduction. The relatively enormous effect of such small additions apparently can be explained only on the assumption that the fatty acid is *concentrated* in a surface film by a selective adsorptive force at the metal surface. The curves again show a steady decrease from static friction with increasing speed of rubbing, the drop being, however, much less precipitous than for glass surfaces.

Fig. 6 compares the results on blown rapeseed-oil, mineral oil, and mineral oil plus 2 per cent of oleic acid. The oleic acid appears slightly superior to the stearic acid as an adulterant after motion has started, though the static coefficient is somewhat higher. The straight blown rapeseed gives only slightly lower values than the mineral oil plus oleic acid.

It is impossible within the scope of the paper to present all of the rather extensive experimental results secured on this machine. For example, several series of tests were made to determine the effect of load on the coefficient of friction at speeds up to 80 ft. per min., and with pressures varying from 25 to 140 lb. per sq. in. The curves for different loads were practically identical in shape and fairly close to one another in position. The relative positions of the curves for different loads were not always consistent, but in general the lower loads tended to give slightly lower coefficients at moderate speeds, though most of the differences were not much greater than the experimental error of duplicate runs. Additional measurements were made of the *static* coefficient at loads up to 1600 lb. per sq. in. without noting any consistent variations, though the average deviation of duplicate measurements at the higher loads was about 15 per cent.

As a result of the work on the Deeley-type machine, the following conclusions can be drawn as to the friction between *plane surfaces* at *slow speeds:*

(1) There is no maximum coefficient of friction at very slow speeds, the static coefficient always being higher than any kinetic coefficient

(2) For polished glass surfaces, the drop from the static coefficient to the slow-speed kinetic is extremely sharp and it is doubtful if the curves should really be considered as continuous. On metal surfaces, however, the static coefficients are not so high and are more reproducible, and the curves appear to be continuous, the static coefficient being only slightly higher than those at very slow speeds

(3) Within wide limits pressure has comparatively little effect on the *coefficient* of friction under the above-specified conditions

(4) Animal and vegetable oils show consistently lower coefficients than ordinary refined mineral-oils

(5) The addition of very small amounts of fatty acids, or of considerably larger amounts of neutral vegetable-oils, to mineral oils, produces a very marked lowering of the coefficient of friction, thus confirming the statements of Southcombe and Wells

(6) On metal surfaces the maximum differences in the coefficients of friction for different oils are found in the neighborhood of zero velocity. In other words, the *static* coefficient of friction appears to be the best single measure of the oil-

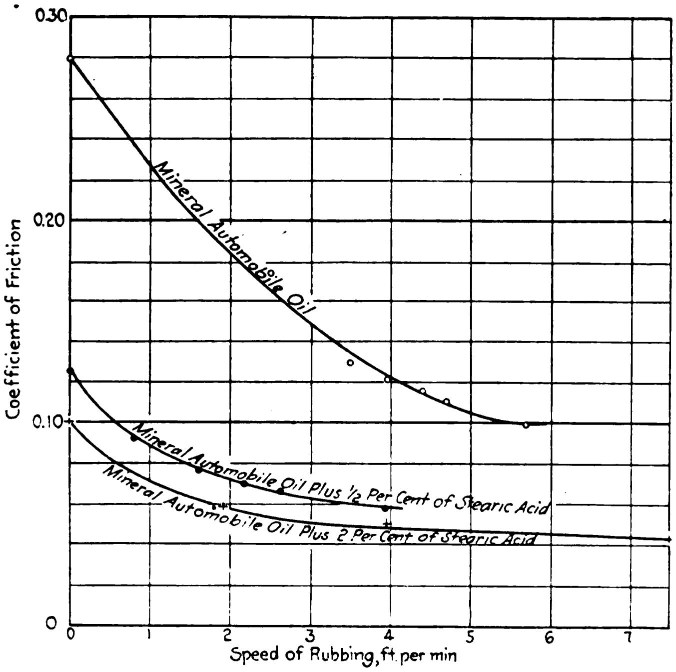


FIG. 5—EFFECT ON THE KINETIC COEFFICIENT OF FRICTION OF ADDING SMALL QUANTITIES OF STEARIC ACID TO A MINERAL LUBRICATING OIL
The Surfaces in Contact Were Cast Iron and Steel and the Pressure Was 30 Lb. per Sq. In.

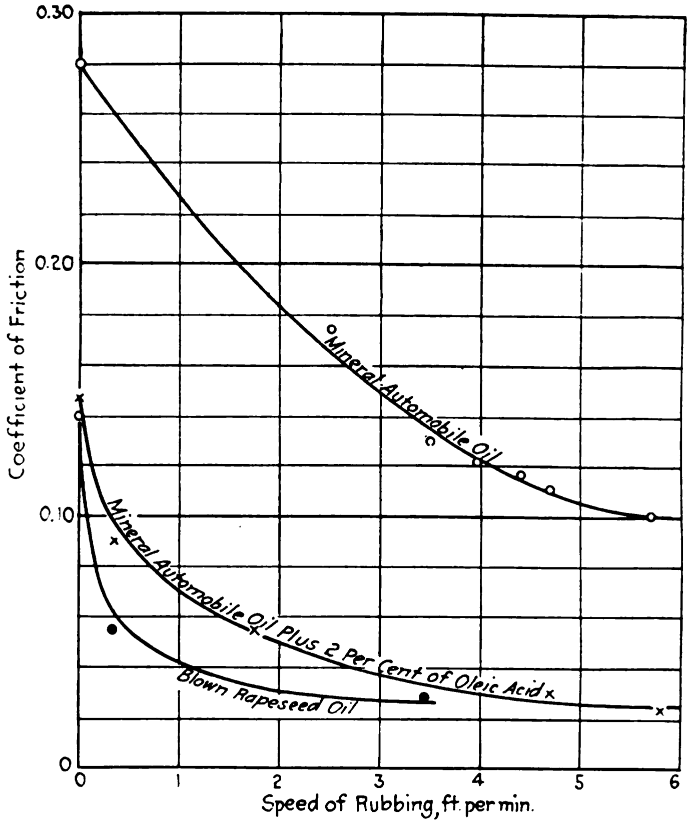


FIG. 6—COMPARISON OF THE KINETIC COEFFICIENT OF FRICTION WITH BLOWN RAPESEED-OIL AND MINERAL OIL WITH AND WITHOUT OLEIC ACID
The Surfaces in Contact Were Cast Iron and Steel and the Pressure Was 30 Lb. per Sq. In.

iness factor of a lubricant, especially considering the greater simplicity of the measurements. Slightly greater *percentage* differences, and greater reproducibility, may be obtained at speeds in the neighborhood of 0.5 ft. per min., but such conditions would require more complicated apparatus and cause much more abrasion than would the determination of the static coefficient

(7) All of the above data confirm the belief that the oiliness factor of lubricants is due to a tenaciously adsorbed film of some constituent of the lubricant, the presence of which diminishes or prevents metal-to-metal contact, even after the surfaces have been pressed together for some time. This ability of the film to withstand prolonged pressure without being squeezed out indicates that it partakes of the nature of a plastic solid[8] with a fairly high yield-point, rather than being a fluid-film such as does the lubricating in a rapidly rotating bearing. To account for its remarkable effectiveness, even on surfaces with very appreciable irregularities, this film would apparently have to be much thicker than the mere mono-molecular film which is frequently postulated

Measurement of the Static-Friction Coefficient

In view of the fact pointed out in conclusion (6) just preceding, and of the time-consuming nature of measurements on the Deeley machine, it seemed highly advisable to develop a rapid and reproducible method of measuring the *static* coefficient of friction for a large number of oils, reserving the use of the Deeley-type machine for making a more thorough study of the more promising possibilities as indicated by these static-friction measurements. A considerable amount of the time therefore has been devoted to the development of such a test, which has proved more difficult than was anticipated, although a fairly satisfactory solution has now apparently been attained.

Some interesting work along these lines has already been done by Langmuir[9], Hardy[10], and Lord Rayleigh[11]. Langmuir made measurements of the angle of slip between non-metallic crystal surfaces, such as mica, quartz and galena, in the presence and absence of mono-molecular films of fatty acids and similar substances, while Hardy worked with various pure organic liquids on polished bismuth and glass surfaces and again obtained evidence of mono-molecular adsorbed films. Lord Rayleigh made a few observations on glass surfaces. None of the sets of results is, however, readily applicable to the lubrication of metal surfaces with ordinary lubricants, though Hardy's methods of measuring the static coefficient were essentially those used in most of our measurements to date.

In the initial experiments in this laboratory, attempts were made to determine the static coefficient between flat surfaces. The results obtained were entirely unsatisfactory owing to the practical impossibility of reproducing the original flatness of the surfaces and the size and shape of the area of contact. It was found necessary to obtain surfaces of as nearly perfect *smoothness* as possible, rather than working toward true flatness. True flats can be obtained only by lapping methods, and consequently always show many tiny scratches which are far more objectionable from the standpoint of accurate friction measurements than very appreciable variations from precise flatness.

It might be argued that excessive smoothness would give results widely different from those obtained with the ordinary types of surface that are used in engineering practice, and such a possibility is indeed quite conceivable. As indicated previously, however, the primary need is for definite reproducible measurements of the oiliness property; reproducible measurements require reproducible surfaces; if surfaces of a definite intermediate smoothness could be thus reproduced they might prove entirely suitable for the purpose in hand, but no procedure to attain this end has yet been suggested, and the only alternative is to make the surfaces so extremely smooth that further changes in degree of smoothness would not affect the results. Furthermore, preliminary measurements with surfaces of intermediate degrees of roughness have shown that the order of magnitude and arrangement of results is nearly the same for all reasonable degrees of smoothness, but that the average deviation of the observations becomes less and less as the surfaces become smoother.

To avoid these serious difficulties, a modified method was developed which consists essentially in the measurement of the static coefficient (from the angle of slip) between two surfaces, one of which is a highly polished and approximately plane surface, while the other is a highly polished spherical segment with a definite radius of curvature, 6 in. By this expedient it is possible to focus all attention on the smoothness of the surface, while minor variations in the planeness or curvature of the surfaces have but little effect. Furthermore, the pressure per unit area is high without requiring the use of large weights, and is reproducible, although its exact magnitude is unknown. Fortunately, however, ten-fold variations in the load, 20 to 200 grams (0.705 to 7.055 oz.) have not been found to affect the results more than 3 or 4 per cent on the average.

To the uninitiated it may seem a fairly simple matter to produce surfaces that have a mirror-like smoothness and yet possess the other necessary properties. The most obvious method at first sight would be to electroplate the surfaces with some metal, such as gold, silver, or copper, that readily takes a high polish, but experiments with such surfaces indicate that even with comparatively small loads the surfaces are so soft that they are actually "plowed" up whenever static-friction measurements are made, and it was felt that the measurements were more a function of the internal properties of the soft metal film than of the adsorbed oil film. Other experiments with soft and readily burnishable metals lead to the conclusion that fairly hard materials must be used to obtain accurate and reproducible results.

One material that takes a remarkably good polish, and yet possesses a high degree of hardness, is speculum metal, a definite crystalline compound of copper and tin in the proportions of 68.2 per cent of copper and 31.8 per cent of tin. This metal is very brittle, but with proper care can be cast into the form of discs, hemispheres or pegs which take an excellent polish. The metal also has the advantage of being practically non-corrodible.

In view of the probable specificity of different metals in building up an adsorbed film, it seemed desirable to approximate most service conditions by making at least one of the surfaces out of steel, preferably fairly hard. Considerable difficulty was experienced in securing suitable pieces of steel that were free from microscopic inclu-

[8] This belief was enunciated and some supporting data offered by one of the authors at the Rochester, N. Y., meeting of the American Chemical Society, in April, 1921. See *Chemical and Metallurgical Engineering*, May 11, 1921, p. 825.

[9] See Surface Phenomena of Ore Flotation, by I. Langmuir. *Transactions of the Faraday Society*, vol. 15, part 3, p. 62.

[10] See Static Friction, part 2, by W. B. Hardy, *Philosophical Magazine*, Series 6, vol. 38, p. 32; also vol. 40, p. 201.

[11] See Notes on the Theory of Lubrication, by Lord Rayleigh. *Philosophical Magazine*, vol. 35, p. 157.

sions of oxide or pits. Furthermore, many steels can be ground to a very smooth surface and yet in the long polishing operation by cloth tend to develop irregularities due to differences in the hardness of different parts of the grain structure, or to form pits due to corrosion. To get a steel reasonably free from non-uniformities, recourse was had to the stock used in making micrometer checks, etc., or to the best quality of case-hardened tool-steel with very fine grain. The second difficulty (corrosion) has been overcome by using kerosene instead of water as the liquid on the polishing wheel. The surfaces are prepared by lapping to the desired form, rough-polishing with 60-min. carborundum on a canvas-covered wheel, and finishing with levigated alumina on a wheel covered with a fine grade of broadcloth.

Static-friction measurements have been made with various combinations of these two metals and also with alloy No. 5, a low-melting alloy (containing bismuth, tin and cadmium) which it was found possible to cast in smooth hemispherical form on watch glasses, but was too soft to give entirely satisfactory results. The best combination appears to be to make the spherical segment out of speculum metal and the plane surfaces of annealed high-carbon steel, both being polished to a mirror-like smoothness. With such a combination of surfaces most of the abrasion comes on the steel, and 25 or 30 observations can be made before repolishing by moving the slider each time to a fresh part of the steel surface.

In measuring the static coefficient the plane surface is placed on a support, the angle of inclination of which can be slowly raised by a motor acting through a belt, light gears and a cord and pulley to prevent the transmission of vibrations. The plane is flooded with oil, the curved slider placed in position on the plane and the angle of inclination slowly increased until slip occurs. A stop is provided to prevent the slider from slipping more than about 1/8 in. The surfaces with tripod slider recently adopted are shown in position for a determination in Fig. 7.

Table 1 shows the results of the preliminary experiments on four different combinations of surfaces with several typical oils that might be expected to differ in the property of oiliness. Each of the recorded results is an average of not less than 10 and generally about 20 separate observations of the static friction coefficient. The average deviations varied from 2 to 5 per cent, being generally below 4 per cent, so that the probable error of the average of 20 observations is generally less than 1 per cent so far as a particular set of surfaces is concerned. It has been found, however, that even in the case of the best combination thus far found (speculum against speculum), differences in the degree of polishing make perceptible differences in the results, even when a high degree of smoothness is apparently approached. Speculum against steel gives slightly greater deviations, but is recommended in order to approach more closely to service conditions. Therefore, too much reliance must not be placed on the precise reproducibility of the different observations before further work shall have been done.

As to the significance of the typical results given in Table 1, it will be noted in the first place that there is again a sharp distinction between lubricants, such as lard oil, which are known to possess a high degree of oiliness, and ordinary mineral oils; and also that small

[12] Since this paper was presented these difficulties have apparently been overcome by using a slider consisting of a heavy metal disc supported on a tripod of three speculum pegs, each 1/4 in. in diameter and polished off to a 1/2-in. radius of curvature. This arrangement gives considerably higher coefficients for dry friction, glycerin, and the like, but insufficient work has been done to establish a series of definite values on the new arrangement.

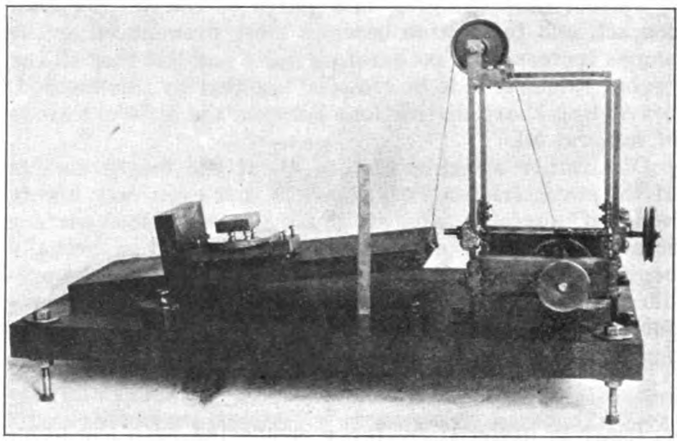


Fig. 7—Arrangement Used To Measure the Static Coefficient of Friction on an Inclined Plane Surface Employing a Tripod Slider

amounts of stearic acid added to mineral oils, such as Velocite B, give values that correspond very closely to the animal or vegetable oils. These results all check up remarkably well with the previously discussed results on the Deeley-type machine, except as to the precise magnitude of the static coefficients. It will be noted that ferric stearate also is very effective in lowering the coefficient, though the soap is rather unstable on standing in the mineral oil.

To show that not all constituents which lower the surface tension between oil and water necessarily lower the static friction coefficient, results are given on Velocite B oil containing 2 per cent of para-cresol. This raises, rather than lowers, the coefficient of friction and indicates that the mere presence of an adsorbed film is not the only essential point, but that its structure is also important. This point is discussed more fully in the following section.

One difficulty with the above method of measurement with the spherical segment on a plane surface is the impossibility of determining the dry coefficient of friction, because in the absence of any liquid the segment begins to twist about on the vertical axis through its point of contact, and slowly works its way down the surface at comparatively low angles of inclination. This can be overcome by using a viscous non-lubricant, such as glycerin, which, as will be noted, gives the highest coefficients observed for these surfaces. It should be said, however, that the higher observed values for the static coefficient of friction are open to some question, since the spherical segment tends to roll slightly[12] as the plane

TABLE 1 — STATIC-FRICTION MEASUREMENTS BY THE INCLINED-PLANE METHOD USING A SPHERICAL SEGMENT SLIDER WITH A LOAD OF 100 GRAMS (3.527 OZ.)

Oil	Coefficient of Friction: Speculum on Steel	Speculum on Speculum	Alloy No. 5 on Steel
Glycerin	0.200	0.200	....
Velocite B	0.182	0.173	0.152
Velocite B, plus 2 per cent of Stearic Acid	0.125	0.121	0.123
Velocite B, plus 0.5 per cent of Iron Stearate	0.120	0.120	....
Neutral Lard,	0.125	0.126	0.093
Velocite B, plus 2 percent of Para-Cresol	0.184	0.178	....
Velocite B Treated with Iron-by-Hydrogen	0.190	0.180	....

is raised, thus bringing new parts of the surface into contact, and this action becomes more pronounced as the angles increase. It is therefore quite possible that all the higher values tend to be crowded together by this method, preventing sharp distinctions between the different kinds of mineral oil.

One rather striking feature about the measurements of the static friction coefficients is that even very highly polished surfaces almost invariably produce surface scratches of an appreciable depth when sliding actually begins, although it appears that these scratches are produced only after the slider begins to move, and therefore play no part in determining the value of the starting coefficient.

Significance of Surface-Tension Measurements

There is probably no factor connected with the property of oiliness that is so frequently mentioned without any clear notion of its true significance, as surface tension. For example, it seems to be assumed generally that a lowering of interfacial surface-tension between the oil and the metal must in some way *necessarily* lower the coefficient of friction, whereas there is no such necessary relationship. Even Southcombe and Wells in their extensive studies along these lines are open to serious criticism as regards their method of presentation.

When any two immiscible liquids are in contact, the molecules in the surface layers are in a partially unbalanced condition and possess a certain amount of potential energy, due to the fact that the molecules of the one liquid do not attract those of the other as much as they attract like molecules. If this were not true, the molecules of one liquid would dissolve indefinitely in the other liquid, as is the case with *miscible* liquids, and there would be no interfacial surface.

It therefore requires work to bring enough molecules of each liquid from the interior of their phases to form the interfacial layers, and the amount of energy thus required per square centimeter of interface formed is called the interfacial surface energy in ergs per square centimeter (which is precisely equal numerically to the more frequently used, but less clearly visualized term, surface tension, in dynes per centimeter).

The magnitude of this interfacial surface energy, as we shall now refer to it, is readily measureable by a variety of means, the most common of which is the determination of the drop weights of one liquid when slowly dropped into another from a capillary of known diameter. In this case the force of gravity is caused to work against the forces tending to prevent the increase of surface and the drop falls when the force of gravity just exceeds the opposing forces, the magnitude of which can then be readily calculated.[13]

If molecules of some third substance are also present in one or both of the immiscible liquids, other interesting possibilities arise. Consider, for example, an interface between oil and water, with some soap present. Soap consists of a long hydrocarbon chain with an active group containing oxygen and some metal at one end. The long hydrocarbon chain is soluble in oil, but the active end with the metal atom is not. Therefore, comparatively little soap will dissolve in the oil layer. On the other hand, the active end has a marked affinity for, and is quite soluble in, water, but the hydrocarbon chain prevents much of it dissolving (except in the form of a colloidal solution where the hydrocarbon chains probably form the interior of colloidal particles with the outside largely made up of the active groups).

It is obvious from these considerations that the one place where a soap molecule can be most "comfortable" is in the interface, with the hydrocarbon chain in the oil layer and the active water-soluble group in the water layer; and we would expect to find a very much higher concentration of soap at the interface than in either of the separate phases, which has actually been proved to be the case. Furthermore, the presence of this type of molecules which are, so to speak, *anxious* to concentrate in the interface, makes it easier to produce more interface. In other words, it lowers the interfacial surface energy more or less in proportion to this concentrating tendency. For substances in the molecular state, the precise relation between the lowering of surface energy and the increase in concentration has been worked out on thermodynamic grounds by Willard Gibbs, but his fundamental equation does not apply quantitatively to the concentration of *colloidal* particles at the interface, though it is frequently and erroneously so used. For the purpose in hand, therefore, a lowering of the interfacial energy is simply a measure of the tendency of certain constituents to become concentrated at an interface, and, by measuring the amount of such lowering, the extent of such concentration can be determined roughly.

In an exactly similar way, there is an interfacial energy between any liquid and any solid, and the tendency of any molecules to concentrate at this interface will cause a lowering of this interfacial surface energy. Unfortunately, however, in spite of many references to this quantity as though it were definitely measureable, no method has ever been developed for

[13] See *Journal of the American Chemical Society*, vol. 42, p. 2534, for a detailed discussion by Harkins of these measurements and calculations.

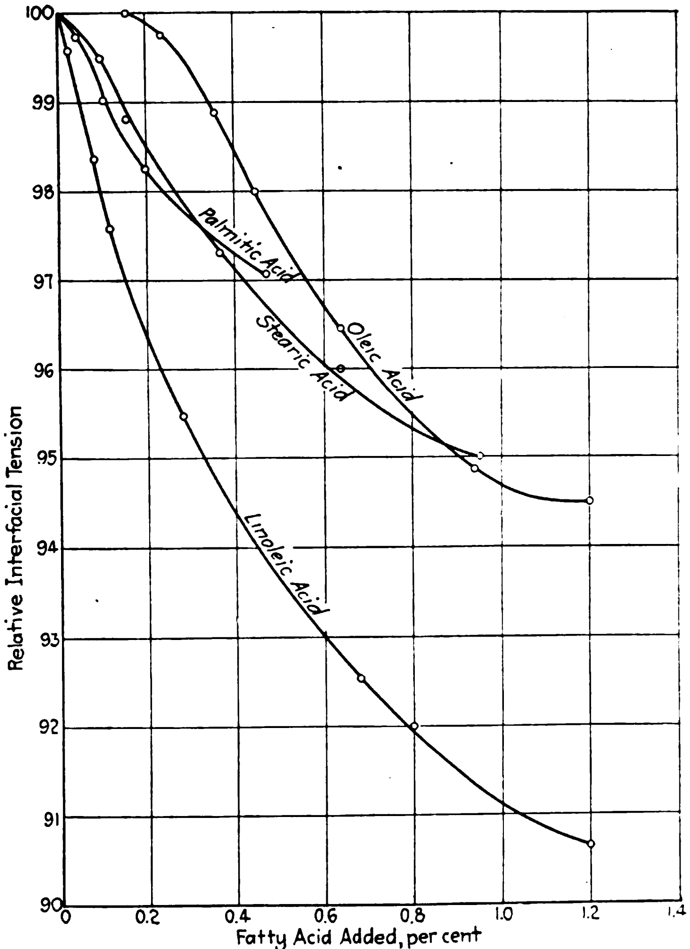


FIG. 8—EFFECT ON THE INTERFACIAL TENSION OF ADDING FATTY ACIDS TO MINERAL OIL

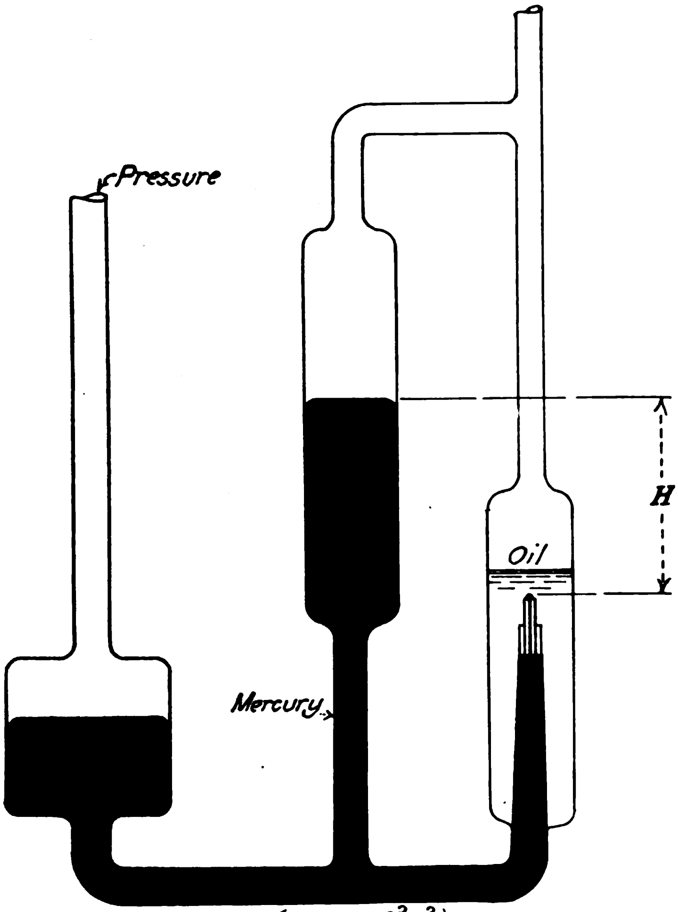
measuring the interfacial surface-tension between a solid and a liquid.[14] On account of this difficulty many have assumed erroneously that anything which lowered the surface tension between any two liquids would also lower the surface tension between one of those liquids and any metal. This is probably true in a very general and qualitative sense, but the two effects are by no means sufficiently parallel for the assumption to be of real value in making comparisons between different constituents of an oil, because it is obvious that the tendency of molecules to concentrate at a metal surface depends upon the attraction between these molecules and those of the metal, which would certainly be greatly different from their attraction for water molecules. In spite of this fact, the writings and patent claims of Southcombe and Wells indicate their belief that anything which lowers the surface tension between an oil and *water* will decrease the coefficient of friction between partially lubricated metal surfaces.

It does seem conceivable, however, that the lowering of interfacial energy between oil and metal surfaces could be determined approximately by making measurements between oil and mercury, the only pure metal that is liquid at ordinary temperatures. At any rate, the approximation would be much better than that obtainable by using water as the second liquid. This laboratory had just completed the construction of an apparatus for such determinations when the interesting results of Bhatnagar and Garner[15] were published, and accordingly only a few preliminary measurements have thus far been made.

[14] It should be said that there are some bare possibilities of arriving at an approximate measure by very special means that never yet have been carried to the limit.

[15] See *Transactions of the Society of Chemical Industry* (London), vol. 39, p. 185T.

[16] See Temperature Coefficients of Surface Energies of Liquids, by F. M. Jaeger, *Verslag. Akademie van Wetenschappen*, vol. 23, p. 330.



$$Y = \frac{Hr}{2}\left(1-\frac{2}{3}\frac{dr}{H}-\frac{d^2r^2}{H^2}\right)$$

d = Density of mercury

r = Radius of capillary

FIG. 9—APPARATUS EMPLOYED TO DETERMINE THE INTERFACIAL TENSION

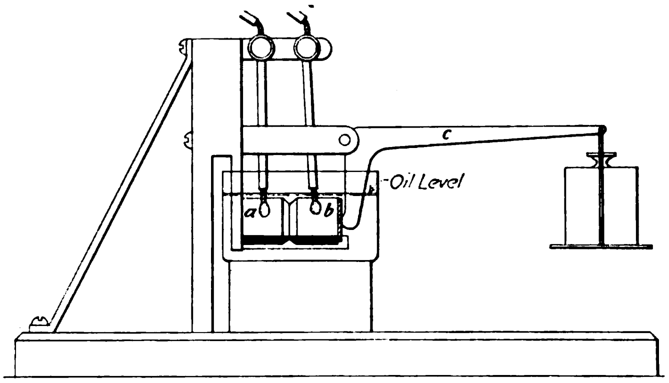


FIG. 10—"CONDUCTIVITY" APPARATUS USED FOR DETERMINING THE ELECTRICAL RESISTANCE OF ADSORBED OIL FILMS

Bhatnagar and Garner's results with four different fatty acids dissolved in mineral oil are shown in Fig. 8. It will be noted that each has its characteristic curve, though linoleic acid seems to be more highly adsorbed than the other three. Palmitic and stearic acids give almost identical results, as would be expected from their chemical similarity.

The method developed by this laboratory is somewhat simpler, being based on the method of Jaeger.[16] The essential portion of the apparatus is shown in Fig. 9. The interfacial tension between the mercury and the oil is measured by the head of mercury, H, necessary to cause the drops to separate from the capillary which is immersed in the oil. The method is much more rapid than, though probably not quite as accurate as, that of Bhatnagar and Garner.

Even granting, however, that the results against mercury surfaces will parallel roughly those against other metals, it must be emphasized that these results do not have any direct bearing on the coefficient of friction between partially lubricated surfaces, in spite of frequent assumptions to the contrary. *Lowering of interfacial energy is only a measure of the tendency of some constituent to concentrate at the metal surface;* whether or not this results in a lowering in the friction coefficient *depends almost wholly on the nature and structure of this adsorbed film.*

This fact is brought out rather strikingly by some of the static-friction measurements just discussed, in which the effects of 2-per cent additions of para-cresol and of stearic acid to mineral oil are compared; both lower the surface tension against mercury; both therefore are concentrated at the metal surface; but while the stearic-acid film greatly lowers the coefficient, the para-cresol film appears to give even slightly *higher* coefficients than the plain oil.

On the basis of these and some other preliminary results, plus certain theoretical considerations, we believe that a partial lubricating film, to be effective in reducing friction coefficients between metal surfaces under high pressures, must possess the properties of a *solid* rather than of a *fluid*-film, and must be of *colloidal* rather than of ordinary *molecular* dimensions. Para-cresol apparently fails in one or both of these specifications, while stearic

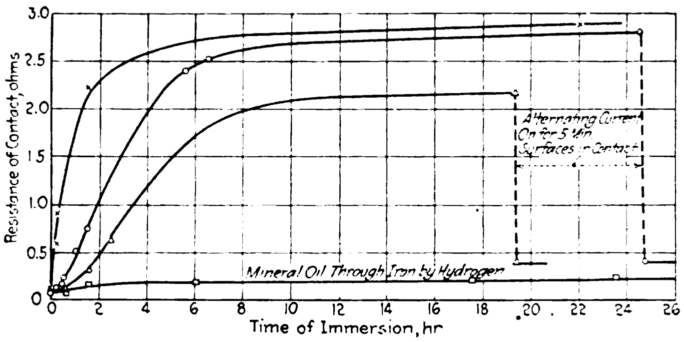


FIG. 11—EFFECT OF THE TIME OF IMMERSION ON THE CONTACT RESISTANCE OF MINERAL OIL

acid possesses them in a high degree, as is demonstrated by some later experiments.

ELECTRICAL-RESISTANCE MEASUREMENTS OF ADSORBED FILMS

As pointed out previously, interfacial-energy and static-friction measurements show the indisputable presence of adsorbed films on metallic surfaces in contact with lubricants. They do not, however, give any definite information as to the thickness or physical characteristics of that film. The writers have therefore conducted some experiments designed to give some idea of the rate of formation and the thickness of the adsorbed film by means of its electrical resistance, with results that were far more striking than could have been anticipated. The procedure employed is outlined briefly as follows:

A "conductivity apparatus" was constructed, as shown in Fig. 10. The oil-film under examination is held be-

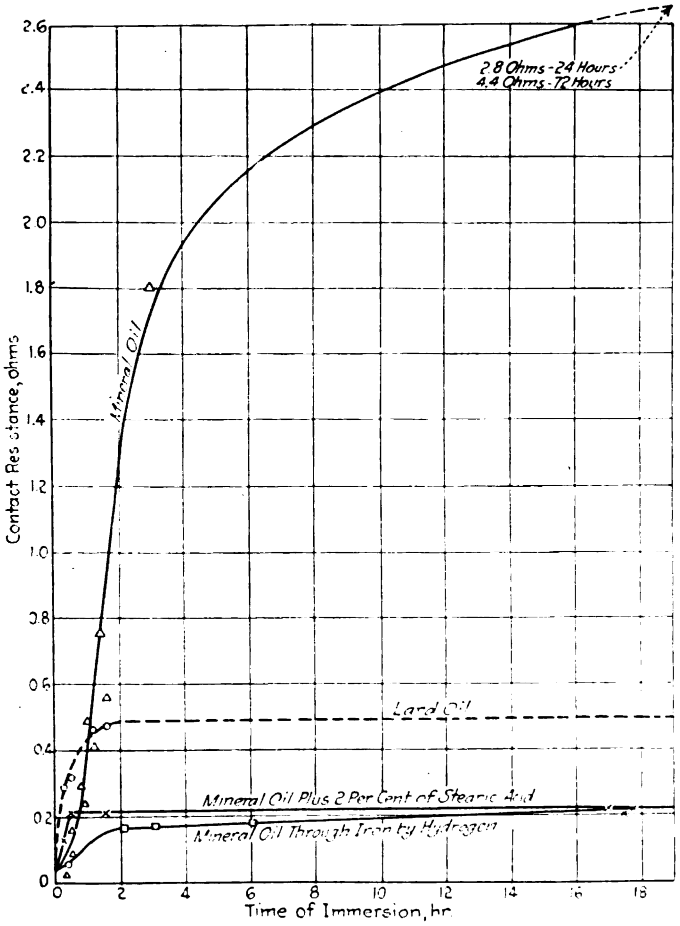


FIG. 12—EFFECT OF THE TIME OF IMMERSION ON THE CONTACT RESISTANCE WITH VARIOUS LUBRICATING OILS

tween the two hardened-steel surfaces, *A* and *B*. The contact faces of these surfaces (originally ½-in. micrometer checks) are lapped and polished true until they adhere tightly when dried and lightly pressed together. The resistance of the contact between these two surfaces was measured by a Wheatstone bridge arranged so that balance could be obtained by either a battery supplying direct current and a galvanometer or an alternating-current microphone hummer and phones. The tests were carried out in the following manner:

The surfaces were carefully polished and cleared,

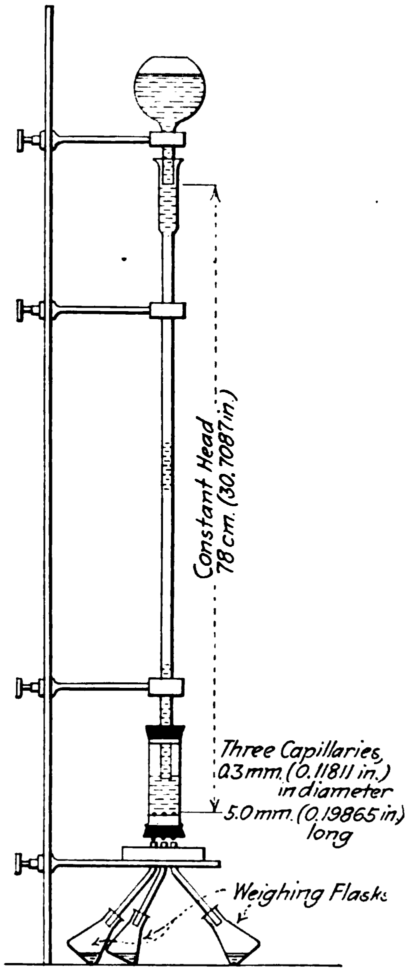


FIG. 13—APPARATUS DEVELOPED TO STUDY THE CLOGGING OF CAPILLARY TUBES

placed in the apparatus as indicated, and a definite moderate pressure applied through the lever *c* and the "dry" resistance noted. This was very low, usually about 0.03 ohm when all traces of air had been removed from the surfaces by gentle rubbing under pressure. Part of even this was undoubtedly due to the leads and connections. The load was then removed, the surfaces separated, immersed in the oil under examination and again placed under pressure. Some little time (about 15 min.) was required for the contact resistance to drop off to a substantially constant value. Usually the first value thus obtained was two or three times the "dry" resistance. The surfaces were again separated, immersed in the oil for a longer period of time and then placed together, and the resistance again noted. This time the observed values were very much higher. This procedure was followed at intervals for about 24 hr., at the end of which time there was in general very little tendency toward increasing resistance. The results of a number of experi-

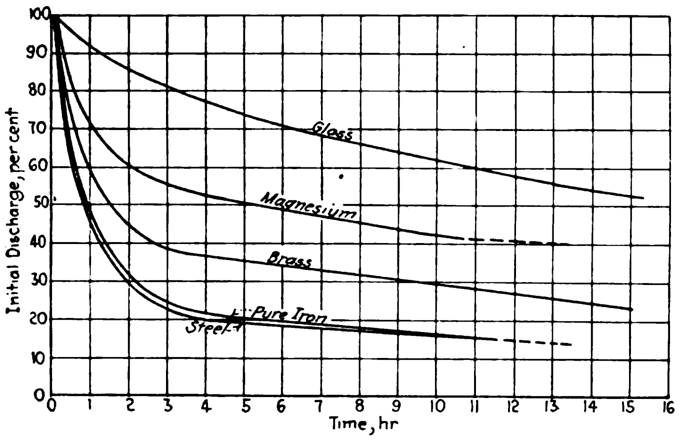


Fig. 14—Film Formed by Untreated Velocite B on Various Capillary Tubes

ments are shown in Figs. 11 and 12. It is evident that the film increases fairly steadily in thickness for about 6 hr., after which the curve tends to flatten off. The different curves in Fig. 11 are for the same mineral oil, Velocite B, and were obtained by regrinding and polishing the surfaces for each series of observations. The divergence indicates the lack of reproducibility, at least in these preliminary experiments. Part of the trouble is due to the difficulty of separating the surfaces after they have been in contact for some time without scraping off part of the film. This is especially true of the more tenuous portion formed after the first hour or two, which is so sensitive to rough handling that it probably plays no important part under service conditions. The first 10 or 20 per cent is much more tenaciously held, although even then vigorous rubbing or wiping will remove all but a comparatively small proportion of the total film thickness. Experiments are now being made with much higher pressures to permit distinction between lightly and firmly adsorbed films.

One phenomenon accompanying these experiments that we have thus far been unable to explain satisfactorily, is the evident destruction of the film when subjected to an alternating electromotive force, even though the voltage and current be extremely low. As indicated in Fig. 11, after the film had attained a resistance value of from 2 to 3 ohms, measured by means of the battery and galvanometer, the circuit was changed so as to impress on the film an alternating electromotive force of very low voltage, but of approximately 1000 cycles per sec. on the film, in an effort to check the measurements by the microphone hummer. Although the current thus passed was less than 1 per cent of the amount used in the direct-current measurements, the resistance fell off with great rapidity to the low value of about 0.4 ohm. Some more

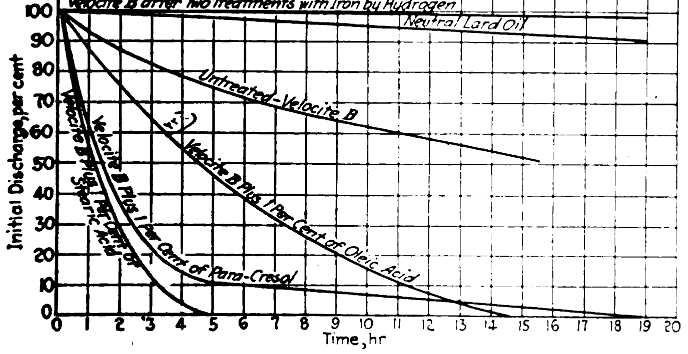


Fig. 15—Flow of Various Oils through a Glass Capillary Tube

recent experiments have shown that a *heavy* direct current has much the same effect, probably due in this case to heating. Since the observations do not vary appreciably, regardless of the length of time the ordinary direct current of 0.002 amp. is applied, the effects of the very small alternating current cannot be due to straight electrolysis, though they might be attributed to the action of rapidly alternating electrical stresses on a gel-like adsorbed film.

One of the most interesting features of these experiments is the surprisingly long time required for the film to build-up in the case of the straight mineral-oils. The most reasonable explanation of this observation appears to be that the film-forming constituents of the mineral oil are present in comparatively small proportion, and that considerable time is required for them to diffuse to the metal surface. To test this out, the same oil was shaken up for several hours with two successive portions (20 per cent by weight each) of very finely divided iron (made by reducing ferric oxide with hydrogen). The iron

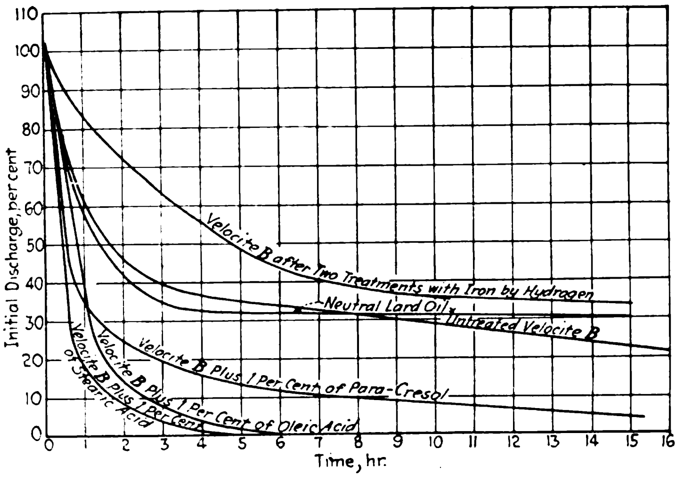


Fig. 16—Flow of Various Oils through a Brass Capillary Tube

was then separated by centrifuging and the resistance measurements repeated, with the results shown in Fig. 12. It will be noted that the resistance rises to only 8 per cent of its former value. This might, of course, be due to a decrease in the specific resistance of the film, rather than to a decrease in its thickness, but careful tests did not disclose any iron soaps or suspended iron in the treated oil. It would therefore appear that a considerable proportion of the film-forming constituents had been removed by selective adsorption on the large amount of iron surface offered by the iron by-hydrogen. It is hoped also that by extraction of the adsorbed mate-

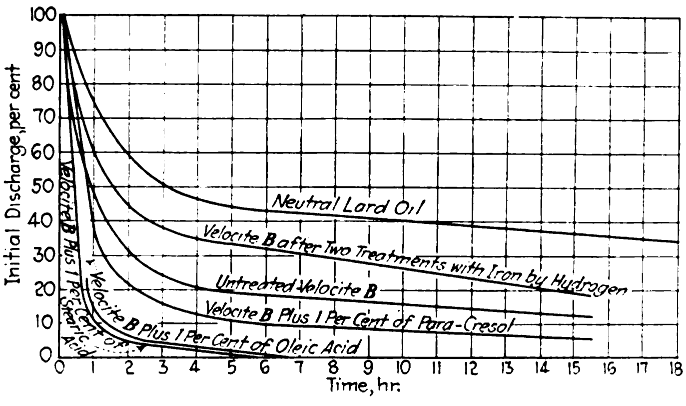


Fig. 17—Flow of Various Oils through a Steel Capillary Tube

rial plus the oil held in the pores and between the particles of the iron, followed by further concentration by readsorption on a smaller amount of surface, it may be possible to isolate and identify the "essence of oiliness" (?) thus separated. It will be noted that the oil thus treated showed an increase in its static coefficient of friction (See Table 1).

Fig. 12 shows the results of similar experiments with lard oil and with Velocite B plus 2 per cent of stearic acid, compared with the results on plain Velocite B plotted on a smaller scale.

It will be noted that the first two (which from other experiments appear to possess the greater oiliness) reach their maximum resistance much more quickly than the mineral oil, although its magnitude is much less. It is obvious, however, that the resistance cannot be taken as a direct measurement of the thickness of the film because of wide variations in the specific resistance of the different oil-films, which is, for example, probably much greater for the mineral oil than for the other two. Neither is it possible to calculate the film thickness by assuming that the specific conductivity of the adsorbed film is the same as that of the oil in bulk, because the active constituents in the adsorbed film probably have conductivities of a much higher order of magnitude than the oil in bulk.

Efforts were made to arrive at a more accurate figure for the film thickness by measuring the electrical capacity of the film, since the dielectric constants of different constituents in the oil vary but little, in marked contrast to their conductivities. This was not found feasible, however, on account of the previously mentioned disrupting effect of alternating electromotive forces on the adsorbed films. Apparatus is now being constructed with which it is hoped to measure the thickness directly by a system of mechanical and optical levers. Meanwhile, other methods described in the next section were tried in an effort to determine how thick such adsorbed films might become under favorable conditions.

Clogging of Metal Capillaries

In view of the above-mentioned uncertainties as to the thickness of the adsorbed films, it was thought that a study of the rate of clogging of very fine metal capillaries when an oil with film-forming tendencies was run through them, might throw some light on this subject. Experiments were first made with capillaries approximating in dimensions those in the standard Saybolt viscosimeter, maintaining a small constant head at all times. These results showed that the rate of flow through glass and metal capillaries tended to decrease gradually over a period of hours, even though the temperature was accurately controlled. The change was, however, less than 5 per cent and it was difficult to control the temperature accurately enough to distinguish sharply between different thicknesses of film. It did confirm, however, the presence of a film of appreciable thickness and seemed to account for many of the difficulties that have been observed in attempting to get check measurements on lubricants of high viscosity in the ordinary viscosimeters. As a check, glass capillaries of the same dimensions were run in parallel and they showed only a negligible clogging during the period of the run.

To magnify the effect, recourse was therefore had to much smaller capillaries, about 0.3 mm. (0.118 in.) in diameter and 5 mm. (0.197 in.) long, where the thickness of the film might become an appreciable part of the total diameter of the capillary. To get an accurate comparison between different types of metal and glass, three jets of almost identical dimensions were used in parallel on the same oil in all tests, the apparatus being constructed as shown in Fig. 13, on which all dimensions are given. Using such small capillaries, it was found that the results were quite erratic until the oil was filtered through 200-mesh wire-gauze to remove all dust and foreign particles, of a size likely to affect the results.

Most of the comparisons were made between capillaries of glass, steel and brass of the same dimensions, through which the same oil flowed simultaneously. Observations of the rate of flow were made by weighing the efflux during 10-min. periods. The efflux during successive periods showed a marked decrease, as is indicated by Figs. 14, 15 and 16. To make the results comparable in spite of variations in the viscosity of the oil, and the precise size of the capillaries, all the data are plotted in terms of percentage of the weight flowing in the first 10-min. period, before appreciable clogging had time to take place, and this initial flow was always very nearly the same for all three capillaries.

Inspection of the three charts indicates, first, that the rate of clogging of the glass capillary was in all cases very much less than that of the metal capillaries, indicating a less tenacious adsorption or a thinner adsorbed film. The only cases in which the decrease even approached that of the metal capillaries was where various organic acids were added to the mineral oil. The results with the glass capillary do, however, show that the very rapid changes in flow are not due to any change in the viscosity of the oil as a whole. This is also shown by the fact that the results obtained after thoroughly cleaning the capillaries always gave results that checked with the original rate of flow.

Proceeding, then, to the consideration of the results with brass and steel, which more nearly measure the specific oiliness property in which we are interested, it will be noted that, under the conditions of these tests, all the oils build up a film of surprising thickness, of the order of magnitude of 0.1 mm. (0.004 in.). It also indicates that the thickness of the film is greater for Velocite B than for neutral lard-oil, and this fact, coupled with the static friction data, indicates that the thickness of the film, at least under comparatively low shearing-stresses, is by no means the only factor of importance in determining the "oiliness" of a lubricant. On the other hand, the effects of addition agents such as stearic acid, oleic acid and para-cresol, which other tests have shown to be adsorbed at the metal surface, are uniformly found to increase the rate of formation and thickness of the film, and in the cases of the oleic and stearic acids to stop the flow entirely after about 6 hr.

The results on the Velocite B treated with iron-by-hydrogen confirm the previous conclusion that this treatment removes a considerable proportion of the film-forming constituents. Similar treatments with Fuller's earth did not affect the thickness of the adsorbed films except possibly making a slight decrease in the case of glass. It will also be noted that the rate of building up the film is very similar to that indicated by the resistance measurements, being rapid for a few hours and then flattening off. The film thicknesses are, however, undoubtedly much greater in the capillary experiments than in the flat disc experiments, or in lubricating practice, due to the very low rates of shear in the former case. Similar experiments are now being conducted at much higher rates of shear.

To prove that the clogging effect was due primarily to a selective adsorption by materials that were present in relatively small proportions, the capillaries were allowed to soak in Velocite B, although the oil was not

passed through the capillaries. Upon measuring the flow after a 6-hr. soaking it was found to be 92 per cent of the normal initial flow, indicating that clogging under these conditions was inappreciable compared with the amount that took place when a large amount of oil was brought into contact with the metal surface.

To determine whether this method would confirm some unpublished results of Dr. A. E. Becker of the Standard Oil Co. which showed cast iron to be greatly superior to wrought iron or magnesium in its film-forming properties, capillaries were constructed of pure Armco iron and of metallic magnesium. The results on Velocite B with all five of the capillaries tested are shown in Fig. 17. The iron and steel check very closely, while the magnesium corresponds more nearly to glass, possibly due to the presence of an oxide film that partially saturates the normal free field of force of the metal.

From a practical standpoint, these experiments indicate the desirability of making viscosimeter capillaries from a comparatively inert material, such as glass, quartz, or agate, rather than from metal, which apparently has a considerably greater tendency to clog with ordinary lubricating oils.

Miscellaneous Experiments

Adsorption of Stearic Acid by Iron-By-Hydrogen:—To throw further light on the action of the iron-by-hydrogen in removing some film-forming constituents from oil, and to demonstrate that the fatty acids were adsorbed at metal surfaces, quantitative experiments have been made on the adsorption of stearic acid from oils by very active pyrophoric iron. The iron powder was made by reducing ferric hydroxide gel with hydrogen at 450 deg. cent. (842 deg. fahr.), the process requiring 2 weeks to give complete reduction at this temperature. The experimental procedure consisted of treating a solution of stearic acid in Acto, a rather fluid, colorless paraffin oil, with several successive portions of pyrophoric iron, allowing the solution to stand, with intermittent shaking for about 24 hr., whizzing out the suspended iron particles, and analyzing the residual oil for stearic acid. The results of some of these experiments are given in Tables 2 and 3.

Some experiments were also made in which incompletely reduced pyrophoric iron, or pyrophoric ferrous oxide, was used in lieu of the pyrophoric iron. The oxide appears to possess a much stronger affinity (probably chemical) for the fatty acid than the completely reduced metal, as is shown in Table 4.

Although careful tests were made, in no case was there any evidence of the presence of appreciable amounts of iron stearate in the solution, although it may have been formed and immediately adsorbed.

Increase in Apparent Volume of Finely Divided Iron:—A few tests were made to determine whether the presence of the adsorbed film would increase the apparent volume of finely divided iron after centrifuging a definite weight for a definite time. The iron used in this case was very finely divided, but not pyrophoric. The apparent volume of duplicate weighed samples centrifuged from benzene checked within about 1 per cent. Other samples of equal weight, shaken with oils and centrifuged 25 min. (much more than necessary to give substantially constant volume), gave *increases* in apparent volume as shown in Table 5.

Cutting Tests:—A few tests were made to determine the effect of varying degrees of oiliness on the behavior of cutting lubricants. A piece of mild steel was given a deep rough-cut with a Stellite tool, taking due precautions to keep the oil as the only variable. There was a marked difference in the smoothness of the resulting cuts, the lard oil being the smoothest, Velocite B plus 1 per cent of stearic acid a close second, plain Velocite B much rougher, and Velocite B heated with sulphur worst of any. Even more striking results have recently been obtained by Bingham and others at the Bureau of Standards.

While these tests were a satisfactory confirmation of the previous conclusions as to oiliness, there seemed to be no way to make the comparisons quantitative, and further work was accordingly discontinued.

Methods Approximating Service Conditions:—There are three very promising methods of investigating the effect of oiliness in a manner approaching service conditions, two of which were mentioned in the previous paper. They are:

(1) Investigation of the effect of oiliness on the critical value of zN/p in a normal full bearing. This laboratory is now undertaking such tests, using a Sperry gyro-compass and applying high loads by processing the gyro

(2) Investigation of the length of time bearings will continue to operate with a moderate coefficient of friction after the oil feed has been discontinued. Viscosity is also a variable affecting these results, but practically can be eliminated in a series of comparative tests between oils of similar viscosity but varying composition

(3) Investigation of the power losses in gears under heavy loads and lubricated by oils. Some inter-

TABLE 2—ADSORPTION OF STEARIC ACID

Stearic acid originally present, per cent	0.600
Stearic acid left after treating 50 grams of solution with 10 grams of pyrophoric iron, per cent	0.280
Stearic acid removed per gram of iron, grams	0.016

TABLE 3—ADSORPTION OF STEARIC ACID

Original concentration of stearic acid in oil, per cent	1.220
Concentration left after treating 100 grams of solution with 20 grams of pyrophoric iron, per cent	0.840
Concentration left after treatment of residual oil (90 cc.) with a second 20-gram portion of iron, per cent	0.600
Weight of stearic acid removed by first treatment per gram of pyrophoric iron, grams	0.019
Weight of stearic acid removed by second treatment per gram of pyrophoric iron, grams	0.011

TABLE 4—ADSORPTION OF STEARIC ACID

Stearic acid originally present, per cent	0.530
Stearic acid left after treating 50 grams of solution with 10 grams of pyrophoric ferrous oxide, per cent	0.040
Stearic acid removed per gram of iron oxide, grams	0.025

TABLE 5—INCREASE IN APPARENT VOLUME OF FINELY DIVIDED IRON

Shaken With	Increase in Apparent Volume, per cent	Duration of Contact, hr.
Velocite B	8	5
Velocite B	14	24
Acto	8	24
Acto, plus 2 per cent of Stearic Acid	8+	24

esting results have been obtained by this method[17] in England,[18] which confirm the superiority of vegetable and animal over mineral oils under these conditions of partial lubrication

The general conclusions that have been drawn from this work are covered adequately in the abstract at the beginning of the paper. In closing, we desire to acknowledge the courtesy of the General Motors Research Corporation and of the Standard Oil Co. of New Jersey in permitting the publication of these results; and the invaluable assistance of Harry Myers and Tyler Fuwa of this laboratory, who have made all the static-friction measurements and capillary-clogging measurements respectively.

THE DISCUSSION

CHAIRMAN H. C. MOUGEY:—I notice that stearic and oleic acids, lard oil and iron stearate were used. The iron stearate, according to the data presented, seemed to be very good. Has Professor Wilson any opinion as to the relative value of adding acids, glycerids and soaps to oil?

PROF. R. E. WILSON:—We realize that most of the things we are now trying are out of the realm of possibility for practical use but believe that if we study enough of them, we shall find something possessing the desirable properties and none of the undesirable ones. The glycerids must be used in larger amounts; they are not adsorbed so readily as the fatty acids and we know exactly why that is. The glycerids are much less corrosive and it may be something of the glycerid type that will afford the ultimate solution. I have in mind something like an ester with a small molecule on one end and a large molecule on the other. Here we would get away from the corrosive tendency of a fatty acid, which is certainly undesirable, yet have something that is adsorbed much as a fatty acid would be. I believe it is not necessary to have a half-way compromise between something that will gum up, or oxidize readily, and something that will lower the static-friction coefficient. I think we can dodge that dilemma because we know from other considerations that unsaturated compounds are not so highly adsorbed as certain oxygen-containing compounds.

J. E. POGUE:—How about the addition of solids to the lubricants?

PROFESSOR WILSON:—That is one of the very important things on our list, but we have not reached it yet. We know when we bring a third-solid phase into the system, we shall introduce other troubles; so we want to be sure of our ground on two solids. However, the addition of solids such as graphite may very likely be found helpful.

T. E. COLEMAN:—Could any of the fatty or organic acids be made so that they would be unemulsifiable in the crankcase?

PROFESSOR WILSON:—That is related very closely to one point that I mentioned. The reason that some of these additional agents form emulsions with water in the crankcase is that they lower the surface tension between the water and the oil. It is entirely concelvable that we could get something that would lower the surface tension between the metal and the oil without doing the same between the water and the oil but in general anything that does one thing will have some tendency to do the other; so probably we shall always have to watch for that difficulty.

[17] Tests with greases are not satisfactory from this standpoint, because so much of the friction is in the mass of grease itself, rather than between the gear teeth.

[18] See 1920 Report of the Lubricants and Lubrication Inquiry Committee of the British Department of Scientific and Industrial Research.

R. W. A. BREWER:—Can you say anything about the effect of camphor?

PROFESSOR WILSON:—No.

G. W. COGGESHALL:—Did you try different metals in that last test where you measured the electrical resistance of different types of oil? Would the same thing occur on platinum, for example, which would not react chemically?

PROFESSOR WILSON:—Not yet. Mr. Barnard informs me that he did try to use brass on one of the oils containing fatty acids but he got corrosion and was not able to make the measurements. In regard to whether the same thing would happen with platinum bearings, there is no doubt that some of the materials used did react chemically with the metal surfaces. Adsorption is, however, the *first* thing that takes place. The chemical reaction is slower. We get our best results as the coefficient of friction on a material like brass within an hour or so after the oil is added. Later if we see any etching of the surface, very frequently we get poorer results that we can trace directly to the fact that the surface is rougher. We are evidently dealing primarily with a physical adsorption which is determined by the chemical structure of the molecules of the lubricant.

C. M. MANLY:—Have any experiments been made to indicate what effect the porosity of cast-iron has in producing the "magic" quality that seems to be obtained by straining oil through cast-iron filings? Cast iron has shown remarkable properties in connection with bearing surfaces.

PROFESSOR WILSON:—I think porosity could not have had much effect on the result because the greatest thing we had to fight against was porosity. We could do a fine job of polishing on a surface, but 95 per cent of the castings we made from different metals are, under the microscope, full of tiny holes. When we polish out one row we come to the next; so we had to do all our work on carefully selected non-porous surfaces. I think the abnormal results frequently observed with cast-iron are almost certainly due to the graphite that is segregated out around the grain boundaries and are something akin to the lubrication action by colloidal graphite.

CHAIRMAN MOUGEY:—In the case of a number of commercial oils the attempt is being made to capitalize the idea of adding fatty acids, glycerids, soaps or materials of like character. Has any one had experience in the use of these oils in service to indicate whether the theoretical value is borne out in practice, or whether troubles develop?

PROFESSOR WILSON:—One thing I did not mention about the static-fraction coefficient bears on that question. Some members may have seen the remarkable claims for "mineral" lard oil or something of the kind made by heating mineral oil with sulphur. Colloidal sulphur is said to be in it but cannot be found. We tried the effect of heating mineral oil with sulphur and found that the compounds formed raised the static coefficient and also increased the type of cut that is made in machine-tool work, at least in our tests.

J. WILLARD LORD:—There has been some talk about reclaiming oils and several firms are engaged in that work. Do you think that oil reclamation is feasible?

PROFESSOR WILSON:—I have no definite opinion. I see no reason why reclaimed oil should not be just as good as the original if the different kinds of oil are kept separate.

DR. W. H HERSCHEL:—I think we should use caution in keeping the two kinds of lubrication, complete-film lubrication and incomplete-film lubrication, separated. It

is customary to speak of glycerin as a liquid that has no lubricating properties. It should be remembered, however, that this applies only to the special field of incomplete-film lubrication. On a friction test, I succeeded in getting the lowest friction with a 60-per cent sucrose solution. This shows that under certain conditions sucrose, or glycerin would show the same thing and could be used as a lubricant. I do not advise using it because it would not be durable.

C. O. BECH:—I had an experience with a bearing that was fitted so closely that although one could revolve it freely when it was mounted on a shaft, after putting on one or two drops of oil the bearing would seize. What is the explanation of that? Has it any connection with the clearance necessary for oil between the bearing and the journal?

PROFESSOR WILSON:—What do you mean by "seize"? What oil did you put on?

MR. BECH:—An ordinary lubricating oil was used. The bearing would bind to the extent that one could not turn it by hand.

PROFESSOR WILSON:—If you have a very small clearance, air is a wonderful lubricant, provided the speed is high enough and the load is low enough to keep it in. It may be that the film was so extremely small that the rate of shear of the oil was very high, but I cannot see why the bearing should seize.

MR. BECH:—Perhaps I should not say "seize." It seemed to me that a distortion took place in the bearing, due to an attraction of some kind, and that this caused binding or increased friction. The torque necessary to turn the journal on the shaft was increased from 5 to approximately 75 lb-ft. This information may be worthy of some consideration in assembling shops of plants building machinery of various kinds.

PROFESSOR WILSON:—I should say that the difference was due to the much greater viscosity of oil as compared with air.

O. P. SELLS:—After being started, the bearing could be spun easily with a couple of fingers. But it took a man's strength to break it loose. What was the reason?

PROFESSOR WILSON:—That is exactly what I showed; that the static coefficient at zero speed is enormously higher than at any other speed. It is 0.2500 for lubrication where there is not a good film. With an oil of high film-forming tendencies it will be around 0.1000. As soon as the speed increases it will drop to 0.0200. A little later it will go to 0.0020. A number of bearings in actual operation did so. The Sperry gyroscope bearings run as low as 0.0020 and one can actually get as low as 0.0007 by using air under heavier loads and high speeds. In other words, the friction coefficients vary four-hundred-fold under those conditions. There is also the tendency of certain oils to solidify when cold and actually build up a gel structure that requires great force to break it loose. This is the main factor in cold weather.

THE CAR-OWNER VERSUS THE SERVICE-MAN

(*Concluded from page* 142)

the two horses we had in 1885. When we think of a farmer owning an automobile we must forget the question of the amount of his income. That is not the question. The fact is that he must have the automobile, just as he must have his mower or his binder. That man wants a job that is standardized to his economic needs as much as possible, and we should keep that need in mind.

We must keep broad thoughts in mind. We must look beyond the four walls of our offices and factories. There are great uncultivated land areas and coal and petroleum deposits south of the equator. How can we cultivate these great areas of land and develop these natural resources? All of the great manufacturing nations of the world are north of the equator, and they will have to supply the machinery.

Mr. Hoover sent me the letter he had received from South Africa and I wrote to several parts manufacturers. I said, in effect, that perhaps they had thought only of factory production methods within their own four walls and considered only what is best for the manufacturers. I advised them to enlarge their vision to include the world, because that is where the products that they manufacture can be sold. They all replied with the acknowledgment that they had been in error, that they believed they had concentrated too narrow mindedly in the designing of their products and that they must, if they are to utilize the full extent of these possibilities, think more in terms of the fields in which their products are to be sold.

I think these service meetings are very helpful. I hope that this annual meeting in Chicago will be a service meeting for some years to come, and that the owners and all other interested people will attend.

POWER LOSS IN PNEUMATIC TIRES

THE rubber laboratory of the Bureau of Standards is equipped with a special dynamometer for determining the power loss in automobile tires. Many standard makes of tire have been tested on this dynamometer. Some interesting figures have been secured. It is stated that an average 4-in. fabric tire, under conditions of normal load and air-pressure, will absorb approximately 0.90 hp. due to rolling resistance at a speed of 25 m.p.h. Under the same conditions the power loss in a 4-in. cord tire is approximately 0.60 hp., while a 5-in. cord tire represents a loss of 1.20 hp. It is estimated that from 80 to 85 per cent of this loss is in the carcass, the tread contributing 10 to 15 per cent and the tube probably less than 5 per cent.

RECLAIMING LUBRICATING OILS

AN investigation of the possibilities of increasing the quantity of used lubricating oil that can be reclaimed is being carried out by the Bureau of Standards. Experiments with a commercial type of oil reclaimer that involved increasing the temperature at which the oil was allowed to stand from 180 to 200 deg. fahr., increased the amount of oil recovered from 80 to 89 per cent. The data obtained show that the reclaimed oil is similar to new oil except as regards sediment. Further investigation is necessary to determine a satisfactory means for the removal of this sediment, which is probably carbon. In this connection it should be borne in mind that this increase in the amount of oil recovered may represent an increase in quantity at the expense of quality.

Discussion of the Minneapolis Tractor Meeting Papers

THE discussion of the papers presented at the Minneapolis Tractor Meeting of the Society consisted without exception of the remarks made at the meeting. Those who presented the papers were asked to reply to the discussion, and their comments are included in the discussions. A brief abstract of each paper precedes the discussion, with a reference to the issue of THE JOURNAL in which the paper appeared.

In addition to discussion given below the discussion of Tractor and Plow Reactions to Various Hitches, by O. B. Zimmerman and T. G. Sewall, was printed with the paper in the July issue of THE JOURNAL. The discussion following the presentation of the paper by R. C. Schoen entitled Practical Road Construction will, it is expected, appear in an early issue of THE JOURNAL, together with the paper.

THE VALUE OF STANDARDS IN TRACTOR MANUFACTURE

BY P. M. HELDT

THE history of the systematic introduction of standards in mechanical manufacture is outlined, instances being given of the need for such standardization and specific reference being made to its value to tractor builders.

After mentioning the necessity of manufacturers maintaining a broad-minded attitude toward standardization, the author discusses steel and other standards, such as tractor hitches, belt speeds, connections between parts or machines that are made in different plants, screw sizes, lug attachment and general matters relating to the subject. The hope is stated that tractor manufacturers will appreciate the full advantage of cooperative effort and of mechanical standardization as one of its expressions.—[Printed in the April 1922 issue of THE JOURNAL, p. 270.]

THE DISCUSSION

CHAIRMAN O. W. YOUNG:—This is a live topic. There are many reasons why engineers and designers should keep it in the forefront of their thought. We have not done very much in the standardization of tractors, but it is a fertile field. We little realize how much S.A.E. Standards have entered into tractor production uptodate, even in small things. There is no reason why they should not go farther toward reducing production costs. The exhibit of the Society at the tractor show contained much food for thought.

One thing that ought to be emphasized whenever this subject is discussed, is that standardization, no matter how far it is carried, never means the sacrifice of individuality or originality in the product. It is surprising how many people still adhere to that old idea. Standardization used to be resisted. When the subject was first broached in connection with tractor design, the designers threw up their hands and said, Conditions vary all over the map and until we get together on some sort of common design we cannot do much standardizing. But standardization of details is the principal factor affecting production costs.

CHESTER S. MOODY:—We have used the S.A.E. Standards in our work. Naturally I shall talk particularly from the material end. We have run into a number of things with those standards. We have had trouble with some of them; others point a way toward a better development of our present standards. The manufacturer who starts with the heavy-type machines is apt to get into trouble as he has a tendency to run to larger sections. He must be careful to apply those standards to sections up to 1½ in. maximum. In our first use of the S.A.E. Standards in metals, we had a 3½-in. section. That is something a new man should know, and there should be information concerning the technical end, the effect of the different alloys on the penetration of heat-treatment.

In tractors that are carrying the heavy loads and high stresses we find that the bearing sizes sometimes are the determining factor. We should have reliable information as to the bearing pressures that different materials can stand.

I cannot say too much in favor of the standards we have had. Our experience has been altogether very pleasing. We have been able to buy steel quickly and it has expedited production in many ways.

The advantage of another factor, that of possibly narrowing our specification, comes in as production increases. One thing that is hard to standardize is the quality of the steel of a given analysis. Our protection is to pick reputable makers. Steel is something that we cannot standardize; I should not say we cannot; we hope that we may some day.

CHAIRMAN YOUNG:—Mr. Moody, by saying that you have to select your sources to assure the quality of the steel, do you mean that ordering from a number of different sources from the same specification would give you a wide range of variation in quality?

MR. MOODY:—That is exactly what I mean. We have had that particular experience. I was not able to specify where the material was to come from, but merely to give the S.A.E. Standard. The trouble came particularly in sizes. If I had received a higher-priced steel I might have expected different results. The trouble occurred not only in straight carbon-steel but also in alloy-steel. It is very common among those who are trying to produce high-grade work to specify sources, that is, probably two or three different sources; they are not necessarily confined to one.

A. W. S. HERRINGTON:—The trouble is largely a matter of organization. I believe it is not only possible but that it is good practice for the purchasing department to have an approved list of purveyors. It is then merely a matter of inspection upon receipt of the materials. You do not have to accept any pay for them if they are not satisfactory. You have that under your control.

W. M. MANSFIELD:—From the implement-manufacturers' standpoint the Society's specifications are a sort of penalty. We have to pay more for material. There are many parts that can be made of mild steel, or of a higher-carbon steel where the range is not important, at a less price than under S.A.E. Standards. Recently we have abandoned the Society's specifications on many parts that carried no particular load, and have gone to the cheaper grade of steel.

O. B. ZIMMERMAN:—That is the same question that has been before our Materials Committee. We find that we can adopt the S.A.E. Standards, so far as specifying No. 1010 and so on is concerned, but we can allow the manufacturer a 15-point range on that specification. Say, for instance, the Society's specification has a 10-point range. We will allow 2½ points above and 2½ below. In that way we can get away from the extra price. One point that it is well to note in the standardization is that the steel manufacturers charge an extra of $5 per ton when a lapping past the 20-point carbon is specified, and $15 per ton above the 50-point. Now in looking up your specifications and working them over, if you will be careful not to lap over that range and keep under the 20-point with one and over the 20-point carbon with the other and under the 50 and over the 50-point carbon, you can save considerable money, especially if you buy in a large tonnage.

P. M. HELDT:—The Society's steel specifications were prepared originally to meet automobile requirements. In a line of work where the requirements are less severe no one would recommend S.A.E. Steel. The requirements are probably the most severe in airplane work; next, in automobile work. These specifications are the best that could be written to meet those very exacting requirements.

G. A. YOUNG:—There is a great need for standardization in the tractor field in not only reducing the cost of the tractor itself, but also facilitating the renewal of tractor parts. The inherent difficulty in tractor standardization, of course, lies in the fact that the industry is really in the experimental stage and it is hard to say definitely what is right and what is wrong. We are getting to the point where we can standardize engine accessories with some degree of confidence that such standardization will not hinder development.

It would seem advisable to make the height of hitch adjustable between certain limits. Belt speeds will probably be reduced to two standards which in time the builders of belt-driven machinery will adopt. Insofar as possible basic machine parts in tractors should be standardized. However, I venture to make the assertion that there are many parts which will not be standardized for some time yet and I think one of these is the question of the lugs.

JOHN MAINLAND:—Personally we can see no reason why two belt speeds are not sufficient and three at the most. In fact, in the days when the steam engine was used almost universally to drive farm machinery there was only one belt speed. Although this never was considered a standard, all manufacturers adhered so closely to it that it might have been considered standard, and in selling threshing machines; clover hullers; silo fillers and corn huskers, the belt speed was never considered, because it was taken for granted. The situation at the present time is very different indeed and our first consideration today, in selling a machine, is to get the power with which it is to be operated in order to put on the correct pulleys to give the correct results.

Practically, all machines of the type mentioned have a very small range within which the number of revolutions of some part of the machine can operate successfully, and this is constant or practically so no matter what the belt speed. The fact is, that some means must be found of getting this speed no matter what the belt speed of the driving unit is.

It is true that the same belt speeds are not the best for all machines, but any tractor, unless its sale is very limited, must meet the varying conditions. Now, if one belt speed is such that it is more convenient for one kind of machine it is less so for the other and you cannot meet all machine conditions to the best advantage with the same belt speed. On the other hand, all belt speeds in tractors must be modified to meet the requirements of the whole range of power-driven agricultural machinery. If there was only one belt speed, this would not be so very difficult, because it would be definitely known what was required, but with the present range of belt speeds in tractors, this is a very complicated process because such a large equipment of pulleys and the like must be carried in stock.

We would like to correct one mistake in Mr. Heldt's paper. He says as follows:

> It is possible that this number of different speeds is necessary temporarily because of the requirements of the various types of power-driven machinery now on the market. The designers of these machines had nothing to guide them in determining the pulley sizes.

This statement is not true. The power-driven farm machinery now on the market was almost uniformly designed for a constant belt speed and at present the builders of them are compelled by the impossible number of different belt speeds in tractors to do the best in their power to have their machines conform to the speeds forced on them by tractor builders.

THE RELATION OF THE TRACTOR TO THE FARM IMPLEMENT

BY G. DOUGLAS JONES

STATING that the trend of tractor development must be toward the small tractor that is capable of handling all of the power work on a farm, the author quotes farm and crop-acreage statistics and outlines diversified farming requirements, inclusive of row-crop cultivation.

Tractor requirements are stated to be for a sturdy compact design to meet the demands of the diversified farm, which include plowing, seeding, cultivating, hauling and belt-power usage, and these requirements are commented upon in general terms. Consideration is given to farm implements in connection with tractor operation, and the placing of cultivating implements ahead of the tractor is advocated.—[Printed in the March 1922 issue of THE JOURNAL, p. 177.]

THE DISCUSSION

O. B. ZIMMERMAN:—This paper produces an interesting reaction on the speaker. After designing tractors for

about 16 years and internal-combustion engines for nearly 30 years, the analysis that Mr. Jones has given seems to me almost like a single patent-medicine. To meet all the possible requirements of a farm tractor with a single unit with all the necessary attachments, with all the necessary adjustments, does not look to me like a feasible proposition. We have, as I see it, a minimum of two types of machine to fulfill all the requirements. One of these must meet the plowing situation in which we have to deal with enormous strains, and the other one, the simpler proposition, concerns power cultivation, the handling of mowers, binders, and tools of that kind. My general analysis is that these two units will persist eventually; that in the course of development it may be possible to obtain an all-around outfit, but with the thought that I have given to the question I must say candidly that I do not believe the proposed solution is possible.

G. DOUGLAS JONES:—Mr. Zimmerman is, I think, holding very steadily to the plow line. I feel that the plow some day will be displaced, will be removed, and means of handling soil, other than by dragging a piece of steel of an angular shape through the ground, can be used to far better advantage than the method used today.

MR. ZIMMERMAN:—We hope that will come about, but I do not see it.

CHAIRMAN O. W. YOUNG:—The general specification that Mr. Jones lays down is pretty broad. Certainly we have not the type of machine to fill that bill to do a small percentage of the work implied as necessary. We know what Mr. Jones' company has done; we know what several other firms have done in an endeavor to give us a general-purpose machine. This is certainly one of the uppermost questions, speaking broadly, that is before the industry today. Some admirable progress has been made toward producing a general-purpose machine.

J. S. CLAPPER:—I agree with Mr. Jones in many things he has said about the tractor, although he has described the "ideal" tractor and it may be a long time before we get to it. When one considers that 45 per cent of the tilled land in the United States goes into row crops and that 55 per cent of the value of all crops produced is in row crops, one has a better idea of the need of a machine capable of doing more work than has been done. I believe we have reached a point where we cannot longer convince the farmer that a machine designed for plowing and beltwork is economical for diversified farming. It is all right in the wheat belt. I think much progress has been made in developing machines capable of doing 80 to 90 per cent of the farm work now being accomplished by horses and mechanical power. Until we can demonstrate to the farmer that the tractor will replace more than 20 per cent of his horses, I am afraid we will not get far in motorizing the farm.

I do not speak of the grain-belt section, for it has been demonstrated that the tractor is the most economical way of farming, but because the crops there are not row crops. I believe that the builder, instead of producing a single-purpose machine and trying to convince the farmer that he must shape his work to fit that machine, must study actual conditions and, if possible, give the farmer a machine that will do the work that he has to do on his farm.

PROF. J. B. DAVIDSON:—I think we may recognize two schools of designers that are working on the problem of the application of mechanical power to farm work. One is trying to displace the horse with a tractor that will perform all of the functions of the horse. It is along this line that present progress has been made. It is perhaps the first step. But I believe the other school, that is trying to capitalize the advantages of the internal-combustion engine, ought to have encouragement. For instance, present practice has not capitalized fully the advantages of light weight and automatic operation.

Future history probably will record that the designers who finally solved the problem got away from the pulling unit. We are doing many things now that are inconsistent from an engineering standpoint; for instance, we transmit the power of the engine to the drawbar and then pick it up again on the bull-wheel of some machine and transform it into a rotary motion. It may be necessary to have the heavy weight for traction to plow with, but if it is a matter of weight we might load our tractors with concrete or some other ballast to secure the extra weight. Perhaps we will build the engine in units so that it could be split into halves or quarters and a part of it used for lighter jobs.

CHAIRMAN YOUNG:—The diversity of opinion on this subject shows how big it is. It is stimulating to hear the different angles. We are probably thinking too much in terms of mechanism rather than in terms of operation. We have visualized mechanisms and tractors and means and have not done the fundamental research to meet the real issue, which is to accomplish the work rather than to consider the means whereby it is done. It is an infinite field.

F. N. G. KRANICH:—A thorough appreciation of the entire unit should be considered, as has been mentioned. The machinery that goes with the tractor is, after all, the important part of the outfit because it does the work with which the farmer is concerned. The tractor is only incidental and a means to an end.

Mention was made of the fact that the implements might be hitched to the front end of the tractor, but it is also possible that they might be hitched behind or even underneath and be just as useful. In either case, particularly with reference to hitching them to the rear, the operator may even ride on the machine where he can be in close touch with all that goes on. This would enable him to be more certain of its correct operation. With implements hitched to the front, the operator may be located too far from the implement proper and not be able to judge its work so clearly. The ears of the operator play a greater part than his eyes in telling whether the machine is functioning right. When he is located on the machine itself, trouble can be heard more easily than when he is located either behind or too far in front of the drawn machine. These points should be given serious consideration.

Using the power of the engine on the tractor as a means for driving the implement to which the tractor is hitched has already received considerable attention from engineers, but even then a thorough appreciation of the work that the implement does is the first consideration in designing what we term a general-purpose tractor.

A. W. SCARRATT:—The specification that has been laid down is practically ideal for a tractor of not more than two-plow capacity. Mr. Zimmerman states that power-farming has to include at least two types of machine and remarks have been made about different methods of tilling the soil. But there have been rotary tillers and other kinds of implements devised to turn over the soil that have never met with success because you cannot get the farmer away from his old way. So long as he persists in thinking of plows as the means of turning over his land, his idea must be accepted. If you are to plow land, machines large enough and strong enough to do the work must be used, and that puts them out of the cultivator class.

I think just such a machine as both Mr. Jones and Mr. Clapper have mentioned should be developed to take care of the farmer whose field does not cover more than 160 or possibly 200 acres and who farms in a diversified manner. When you get beyond that size of farm and come to the large farm where huge volumes of grain are produced, some of these considerations are eliminated automatically and one must have a tractor of a different kind, and that kind of machine cannot be a cultivator.

CHAIRMAN YOUNG: — That is true. Mr. Jones approached the subject purely from the standpoint of the general-purpose machine. That has distinctly one field; it applies to the requirements of a majority of potential tractor sales. From a manufacturing standpoint this is one of the most important phases of the subject because a small machine must be produced in large quantities to meet this market. The general-purpose tractor, the small machine below two-plow capacity that must be used as a cultivator, still remains the big market.

W. M. MANSFIELD:—At a recent meeting of the Society in Chicago the thought was brought out that engineering is a compromise. In designing a general-purpose machine we undoubtedly shall have to compromise to a great extent. It has been estimated that plowing constitutes about 20 per cent of the work on the average-sized farm in the corn belt. Possibly we shall have to compromise a little in the matter of weight and power in this 20 per cent to get a unit of the right capacity for the other 80 per cent. The general-purpose machine that is power-driven gives an opportunity to operate implements directly from the engine rather than through the bull-wheel as on the binder and mower.

A. ANDREWS:—In talking at the tractor show with a wheel manufacturer I asked him whether the hub was cast last or first. He replied that the hub and the spokes were cast first and that a tension was produced artificially before the rim was welded. He said that anyone who has done manufacturing appreciates that handling the work, rather than doing it, is the big item. Most people do not look at it that way. That is the problem on the farm; how a farmer will handle the farm is the real problem. When that is known we can easily find out what kind of machine is needed to take care of him.

CHAIRMAN YOUNG:—That always will be an open question in power farming. Certainly the methods of operating will change. They will be influenced to a great extent by engineers and manufacturers who take a step in advance by producing a machine that will do things in a little different way. The farmer will meet them half way or go them one better by using that machine in ways not thought of by the manufacturer. Then there will be another step by the manufacturer, and so on. This is going on now. We little dream how tractors will be used by the farmers. We are sometimes startled to see a tractor used for purposes different entirely from those for which it was intended. It would be very helpful if engineers would spend a little more time in the country studying or investigating some of the interesting things that the farmers devise.

A. H. BATES:—There is a wonderful field for the small tractor but, from an economical standpoint, I believe there is a decided tendency toward larger units. In fact, there is a persistent demand for a unit as large as one man can operate. It is easily shown that such a unit can be operated at a greater profit to the farmer than a small unit.

CHAIRMAN YOUNG:—There is no question about that from the standpoint of purely operating expense, but that situation might change over-night if prices were cut. Such a contingency might open up an entirely different viewpoint because the first cost of the tractor enters so largely into the fixed expense applying against it.

MR. BATES:—The point I wished to bring out is that the operating expense of the tractor is largely in the wages paid to the operator. If a large unit of three times the capacity of a small one can be operated for the same labor expense, it is self-evident that the cost of production with that machine will be less.

CHAIRMAN YOUNG:—If the farmer kept his books as the manufacturer does, that is the probable viewpoint he would take of it, but I do not think he does much bookkeeping.

MR. JONES:—Mr. Bates' statement is so broad that it gives an unlimited range. Conceive for a moment a tractor of 50 tons being operated more economically than a tractor weighing 1 ton. It is impossible. If labor cost were the only factor to consider, how pleasant it would be for the farmer. In reality the maintenance cost is really outrageous in many cases, and that is only one of the big costs. To eliminate the maintenance cost you would have to reduce the tractor to a compact, simple machine requiring little attention, but one that will keep up and give the farmer value for his investment. When we consider that plowing is 20 per cent of farm work, and that that is the load, I cannot see why we do not cease plowing and handle the soil as easily and probably better some other way and increase our crops to the volume they have in Europe. Why can we not develop our soil and treat it by mechanical processes by inexpensive, small machines that still have the power? We all know we are coming to a light machine. The airplane has shown that. We do not require the large machines of the past to do the same work today. I cannot see wherein a large machine is cheaper to operate than a small one.

MR. BATES:—I would like to enlarge on that. A small machine, a two-plow machine, weighs approximately 2400 lb. A number of six-plow machines that weigh under 7000 lb. are being produced at the present time. The cost of production should not be more per pound for one than for the other, except as the large production of the smaller machine might have some effect. Now, if we grant that, and also that one man can operate either machine, I think it will show that the cost of operation of the larger machine will be less. I have figures on two, three, five and six-plow machines. The man who came nearest to making a financial success was the man who operated the largest machine.

MR. JONES:—What was the limiting factor of the size of the farm? That must be considered first. I have taken 150 acres as the size of the farm wherein volume tractor sales must be taken into account. That is the point I would like to make very plain; that our volume sales are for the 150-acre farm. There we can operate with four horses, do fine work and still keep going. So why have six bottoms? Taking the crops throughout the year, when one portion is disced for corn, you do not plow it that same year, for the plowing has already been done. When the winter wheat has been put in and has been harvested, that too is out of the way. If rotation is kept up, which is absolutely essential to good fertility, plowing is limited; a large gang of plows is not needed. There was a time when a certain company had 55 bottoms in one unit.

A. R. SANDT:—Referring to the original and the maintenance costs, I do not believe that it is impossible to design and produce a tractor that will have a low maintenance cost. One can be produced that will have a low

maintenance cost and also have a low selling cost. One thing we fail to impress upon the farmer is that when he is buying a tractor he is buying a production machine. The tractor is a production tool and prices will have to be raised so that a machine can be turned out that will give the farmer a minimum maintenance cost.

E. R. GREER:—It is a mistake to figure that, because a farmer has a 50-acre farm, he must have a machine to do everything on that farm. I do not see any reason why he should not have a machine for row-crop cultivation and why a group of farmers should not have a suitable machine to do the plowing for everybody. A little two-plow machine will not plow more than about 5 acres per day. You have to get more speed, and cover more acres per day, to do it right if you have a big plowing job to do. Why can they not have machines in each community to do the heavy work, and have light machines to do the cultivator work?

PROFESSOR DAVIDSON:—I do not know of any successful community ownership, but the contract proposition works very successfully in the West. A large number of tractor contractors on the Pacific coast plow for a certain sum per acre or per day. This plan provides personal responsibility for the machines.

The question of the larger plowing-unit and the smaller cultivating-unit must be a compromise, unquestionably, because more power is needed for plowing than for cultivating. If it is a compromise, then this "ideal" machine proposed by Mr. Jones will have more power for cultivating than is necessary, and will be a little limited in power for rapid plowing. Such a tractor will come into competition with the horse in the intertilled-crop area where the best farmers use five horses and are able to plow perhaps 4 to 5 acres per day. If a farmer can do his intertilling or cultivating with a unit that will increase his capacity in plowing over this amount, it will appeal to him.

CHAIRMAN YOUNG: — A compromise machine apparently is one that would not be too heavy for cultivating, not merely from the standpoint of soil-packing but from the standpoint of economy of operation, and still would be large enough to be economical and profitable as a plowing unit.

MR. GREER:—I did not mean to give the impression that I advocate a machine being owned by the community when I said there could be one machine in the community, but rather, as suggested, that some one contractor should do the heavy work. In our operations in the city many of our customers have speed-trucks but contract with us to do the heavy work and heavy hauling.

MR. ANDREWS:—Almost every farmer has a Ford, and it can be proved that an attachment can be furnished for $100 which will do all the plowing on a 150-acre farm. The farmer can buy a cultivator with a motorcycle engine, similar to what Mr. Parrett had at the show, at a production price; so he can afford to have two machines. Here you have the lowest fixed charge that will take care of the work.

MR. SCARRATT:—What about the maintenance charges on the Ford?

MR. ANDREWS:—That is a matter of personal opinion on the part of different individuals.

MR. CLAPPER:—A good many people talk about cultivating machinery who do not know what is required in cultivating row crops. They have not been on a farm and actually spent time in cultivating different crops. We are discussing two different subjects; one is the smaller farm, where the volume of tractor business is coming from, the small farm where there are only 20 to 30 acres in corn, probably 40 to 50 acres in wheat and oats, and all the ground is not plowed at the same time but at different seasons of the year. When it comes to the large farms of the wheat-belt section, the cultivator is out of place. The average farm where there is diversified farming, where the principal crops grown are row crops, is in the central eastern part of the United States. That is where the "ideal" machine that Mr. Jones explained will be used if it can be built or developed, and I believe it can be developed and can be produced profitably. A great many people are wrong as to what is required to cultivate row crops. You cannot take a Ford car and put an attachment on it and hook a cultivator behind it. You have not the clearance in the first place.

MR. ANDREWS:—I did not say that; I said just for plowing.

MR. CLAPPER:—There is more to cultivating row crops than the average fellow thinks, unless he has had experience.

CHAIRMAN YOUNG:—Whenever this subject is opened up it is always a grand free-for-all. While we have skipped around and touched on many different phases of the subject, they all bear on the same thing. We have been consistent in that we have talked, to a great extent, of the relation of the tractor to the implement or its operation. This subject is one on which you never start at the beginning or finish at the end; you grab hold somewhere in the middle and hang on as long as you can and go as far as you can.

EUROPEAN INDUSTRIAL OUTLOOK

UNLESS we entirely misread the ascertainable facts of the economic life of the chief European countries at the present time, what is really occurring is that Europe is rapidly curing herself from the bottom upward, so to speak, instead of from the top downward. At the top, in politics, public finance and the direction and administration of all governmental affairs, we see confusion, conflict of purpose, uncertainty and blundering; but at the bottom, in industry, trade and those things that concern the mass of society, we see everywhere steadily increasing activity, confidence and certainty of motion. Despite all handicaps, the tide of production, whether in agriculture or in industry, is continuously rising in all parts of Europe except perhaps Soviet Russia; and as they produce more, the peoples of Europe consume more, live better and establish their entire economic life on a firmer foundation. This being the case, let us squarely accept the probability that it is better for them that projects for great international loans to them should fail rather than succeed. And to this probability may perhaps be added another, namely, that it will ultimately be found that the politicians have got nowhere with their schemes for settling such vexed questions as that of the German reparations, that of the opening up of Russia to the trade of the world, and so on, but that somehow or other through the silent operation of the great forces of industry and trade, as carried on by the undistinguished mass of the people, in the several countries, inevitable and therefore acceptable solutions of all of these vexing questions will finally have been arrived at. —A. R. Marsh in *Economic World.*

Some Fundamental Characteristics of Present-Day Buses

By R. E. PLIMPTON[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* PHOTOGRAPHS AND DIAGRAMS

THE author enumerates the distinctive features of buses designed for city, for inter-city and for country service and comments upon them, presenting illustrations of these types of bus. Steam and electric motive power are discussed and the chassis components for bus service are considered in some detail. The general types of bus body are treated, together with the influences of climatic conditions and local preferences.

Comfort and convenience factors are discussed at some length and the problems of heating, lighting and ventilation are given constructive attention. Fare-collection devices and methods are commented upon, and the State and local legal regulations are referred to in connection with their effect upon bus operation.

Illustrations are included and a table showing condensed specifications for city buses is presented.

ANY division of the types of vehicle used for buses must be arbitrary, but for convenience we can consider them as applied in city, inter-city and country service. The city bus is designed for use on good pavements. The seating capacity is high, 20 or more passengers, but high speed is not essential. A maximum of 20 m.p.h. is often considered ample. Ability to thread through traffic and to move passengers quickly, on and off, is required. Ordinarily, overload capacity should be provided, so that standees can be accommodated during the rush hours.

The double-decker, a dense-traffic vehicle, has been developed highly through years of service in London and New York. Table 1 gives specifications for the latest types used in these cities. The L-type bus of the Fifth Avenue Coach Co. is distinguished by the low floor, the first step being at the curb and only one other being required to reach the floor of the bus. Increased seating capacity, without any increase of weight, has been secured in many recent double-deckers. A sample now operated in Chicago seats 69 passengers, with a total vehicle weight of only 157 lb. per seat. Previously, about 200 lb. per seat had been considered good practice. This Chicago bus is probably the largest double-decker ever operated commercially. Fig. 1 shows the plan of the lower deck of what is said to be the largest double-decker in England, a Leyland 63-seater carrying 34 passengers on top. Notice the space for a loading well at the lower entrance.

Covered upper decks have been tried out in New York City and Chicago, but have proved popular only in bad weather. Double-deckers in London, England, and Toronto, Canada, have been fitted with blankets for each top-deck seat, in the effort to build up the load factor, these being illustrated in Fig. 2. The so-called double-single-decker has been tried in England; the entire top deck of this vehicle can be detached for winter service,

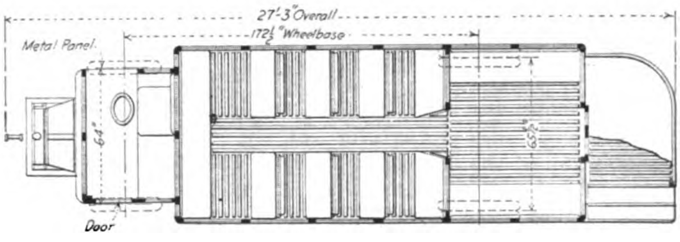


FIG. 1—PLAN OF THE LOWER DECK OF AN ENGLISH DOUBLE-DECK BUS CARRYING 63 PASSENGERS

the stairs being removed and the hatchway closed by a trapdoor. One vehicle of this type has a capacity of 29 passengers as a single-decker, and of 54 passengers, half below and half above, when both decks are in use. However, the single-decker shown in Fig. 3 has by far the largest field as a city bus. A few of these designs are described in Table 1 and there are many others of a similar type under way by other builders. Long wheelbase to give loading capacity, a wide rear gage and a narrower gage on the front to give stability and a small turning-circle, low frame-height to give quick passenger move-

FIG. 2—TARPAULIN LAP ROBES USED ON THE UPPER DECK OF THE LONDON BUSES

[1] M.S.A.E.—Associate editor, *Bus Transportation*, New York City.

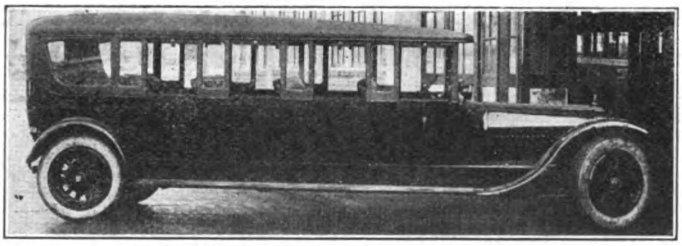


FIG. 4—CALIFORNIA STAGE ON A REBUILT AUTOMOBILE CHASSIS HAVING A CAPACITY FOR 14 PASSENGERS ON FIVE CROSS-SEATS

ment as well as stability, long springs and special devices for easy riding are the more important features of what promises to be a design that will be used widely in and about our cities.

The design for a light, fast, single-deck, one-man bus is being considered for city service by the engineering association of the American Electric Railway Association. It is of interest to know that experts on city transportation favor three sizes, for 21, 25 and 29 passengers, with a maximum floor-height of 26 in. A smaller size, for 16 passengers, has also been discussed.

Most city buses have a central aisle, with cross-seats on each side. In a modified form the seats near the service door are of the longitudinal type, thus providing a space for standees or for those making short trips. This is being applied in short-haul work, while there is a tendency to use the unmodified cross-seat type in suburban work, and even for inter-city service.

THE INTER-CITY BUS

The inter-city bus, such as is shown in Fig. 4, is smaller and faster than the city bus. From 12 to 20 seats are provided for the long runs of 20 miles and upward. These vehicles are known as stages in the West and particularly in California, probably because of familiarity with the term to designate a horse-drawn omnibus. The stage usually is built with three or four rows of seats running all the way across the body, each having its own side doors. It resembles an enlarged touring car and, of course, that is what it frequently is. Comfortable riding is desired, rather than the ability to handle crowds. The floor level is kept as low as possible without decreasing the road clearance below 7 in. The center of gravity is kept down to eliminate sway, with its consequent danger, and undue wear on tires and wheel bearings.

The average speed in Western inter-city service is high. Orders to drivers on a large system in California are to make 32 m.p.h. on well-paved highways and 35 m.p.h. is the legal speed limit. However, there are stretches where it is hard to keep down to this speed. For example, on the desert floor west of El Centro, Cal., a broad concrete pavement stretches away for 30 miles. There are no cross-roads and no habitations are in sight; there is no obstruction to view and very little traffic. With a high-powered car, the driver who will keep down to 35 m.p.h. is unusual. The actual speed over such stretches is from 40 to 45 m.p.h. and, occasionally, 50 m.p.h. The chassis and body should be of the minimum weight, consistent with long life and durability. The improvements being made in highway systems will, it is believed, lead to the use of even lighter chassis than are used at present.

FIG. 3—SINGLE-DECK, 25-PASSENGER BUS FOR CITY SERVICE

The stage used in the West is, as a rule, a rebuilt product. A high-grade touring car is bought second-hand, the frame is lengthened 4 to 5 ft., and a heavier differential, heavier springs and heavier tires are installed. It is realized that these vehicles are expensive, in both first and maintenance costs, since they are always in the "special" class; but Western operators have used them because they could not buy a chassis that would carry the long bodies up steep grades and at high speeds demanded by Western operating conditions. One design at least, that is included in Table 1, is now on the market, and there are indications that others will be avaliable soon for inter-city service, not only on the Pacific coast, but also in many other parts of the Country where the large rebuilt touring car has been used heretofore.

THE COUNTRY BUS

The country bus may start nowhere and end nowhere, as is said of some of them, but usually it runs through rural districts to a trading center or a point where connections can be made with some other form of transportation. This service does not require particularly high speed, but the bus should be able to keep going over poor roads, even when they are covered with mud or snow. Travel is light in this service; so the vehicles used are small. Touring cars of five and seven-passenger capacity may be operated when the roads are particularly bad, giving way to larger buses, up to 18-passenger capacity, during the summer months. One trip a day is often the custom, to town in the morning and returning in the middle of the afternoon. Thus, the operator has the time to pick up light freight for delivery along his route; or he may be a star-route contractor for the Post Office Department, in which case he may carry passengers and freight so long as his contract is properly performed. Therefore, the country bus should be built to carry both passengers and freight. Most operators seem to use the standard passenger vehicle, car or bus, and carry the packages inside or outside, wherever there is vacant space. A space for packages is placed at the front of the vehicle in some designs, with the rear given over to seats.

A convertible vehicle would be particularly useful in consolidated school service, where many buses are now operated only 3 or 4 hr. per day. The seats are removable in one design, so that the school bus can be changed quickly to a light delivery truck. Several combination bodies are on the market, designed to carry both passengers and express, but they have not been used to any extent by bus operators so far.

A transferable body has been suggested for school use, although it might be applied by the operator of country buses also. This would be mounted on a motor-vehicle chassis when the roads were good, but transferred to a horse-drawn vehicle, on wheels or runners, when they are

impassable. The fact that this has been proposed seriously indicates the need for properly designed country buses. The field for them is enormous, and it should be possible to provide operators with a light, sturdy, economical bus for use during the 12 months of the year. Some engineers contend that good roads must come first. Yet bus service is being given on thousands of miles of unimproved roads. The life of the vehicles used is short, and some operators believe it economical to turn them in yearly for new ones. Here is a real problem waiting for automotive engineers to solve. The bus for city service is receiving attention from a large number of sources, but its country cousin is, as yet, almost neglected.

Steam and Electric Motive Power

Before taking up the details of chassis and body used for the standard gasoline bus, other types that have been applied will be noticed. The gasoline-electric bus shown in Fig. 5 is used commercially in England, some 175 of the Tilling-Stevens chassis being operated in London, and

TABLE 1—CONDENSED SPECIFICATIONS OF TYPICAL CITY BUSES[2]

Unit	London General Omnibus Co., Type S	Fifth Avenue Coach Co., Type L	Imperial	Fifth Avenue Coach Co., Type J	Mack	Republic	White	Duplex	Fageol
Number of Seats..	54	51	30	25	25	25	25	23	20
Approximate Floor Height at Entrance, in....	31	18	26½	25	32	28	28	27	18½
Wheelbase, in....	179	175	195	172	180	168	198	160	188
Front Gage, in...	72½	67	66½	68	58¼	66	58½	56	70
Rear Gage, in....	70	72	71	71¼	60	66	60¾	62	70
Total Weight, lb..	9,380	10,150	9,500	6,900	8,850	7,900	8,900	6,500	6,800
Body Weight Only, lb........	2,800	4,000	3,000	2,100	2,700	2,400	3,500	2,300	1,600
Engine, Bore and Stroke, in.......	4.26x5.51	4x6	4⅛x5¼	4x6	4¼x5	4⅛x5½	4¼x5¾	4x5¼	4¼x5½
Type of Valves...	L-Head	Sleeve	I-Head	Sleeve	L-Head	Sleeve	L-Head	L-Head	I-Head
Engine Speed, r. p. m.........	1,050	1,400	1,600	1,400	1,275	1,800	1,650	1,600	1,800
Horsepower......	34	50	47.½	50	37	50	50	42	62
Fuel Tank Capacity, gal........	25	40	30	35	25	30	35	25	28
Fuel Feed.......	Vacuum	Gravity	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum
Carbureter.......	Zenith	Zenith Special	Zenith	Zenith Special	Schebler	Float-Feed	Own	Stromberg	Zenith
Cooling System...	Pump. Tubular Radiator	Thermo. Tubular Radiator	Pump. Tubular Radiator	Thermo. Tubular Radiator	Pump. Cellular Radiator	Thermo. Tubular Radiator	Pump. Cellular Radiator	Pump. Cellular Radiator	Pump. Cellular Radiator
Ignition.........	Magneto	Magneto	Magneto	Magneto	Magneto	Magneto	Magneto	Battery	Battery
Clutch..........	Single Dry Plate	Single Dry Plate	Multiple, Dry Disc	Single Dry Plate	Multiple Dry Disc	Multiple Dry Disc	Single Plate	Multiple Dry Disc	Multiple Dry Disc
Transmission Speeds.........	3	4	4	4	4	4	4	4	4
Control.........	Center	Side	Center	Side	Center	Center	Center	Center	Center
Universal-Joints..	Fabric and Metal	Fabric	Metal	Fabric	Metal	Metal	Metal	Metal	Metal and Fabric
Steering Gear....	Worm and Nut	Worm and Nut	Screw and Nut	Worm and Nut	Worm and Gear	Worm and Nut	Worm and Sector	Worm and Nut	Worm and Nut
Front Axle.......	I-Beam Section	I-Beam Section	I-Beam Section, Dropped Center	I-Beam Section	I-Beam Section	I-Beam Section	I-Beam Section	I-Beam Section	I-Beam Section
Rear Axle.......	Underslung Worm	Internal Gear	Internal Gear	Worm, Semi-Floating	Double Reduction	Internal Gears	Double Reduction	Worm	Underslung Worm
Wheels..........	Cast Steel	Steel Disc	Steel Disc	Hollow Spoke	Artillery	Artillery Cushion	Cast Steel	Artillery	Disc
Front Tires......	Solid, 41.30x4.73	Solid, 34x5	Cushion, 36x6	Cushion, 34x4	Cushion, 34x5	Cushion, 34x4	Solid, 36x4	Pneumatic, 35x5	Pneumatic, 36x6
Rear Tires.......	Dual Solid, 41.30x4.70	Dual Solid, 34x5	Dual Cushion, 36x6	Cushion, 34x6	Dual Cushion, 34x5	Cushion, 34x8	Solid, 36x7	Pneumatic, 38x7	Pneumatic, 36x6
Spring-Suspension	Semi-Elliptical, Volute Auxiliary	Semi-Elliptical, Progressive	Semi-Elliptical, Compensating Underslung	Semi-Elliptical, Progressive	Semi-Elliptical, Rubber Blocks	Semi-Elliptical	Semi-Elliptical	Semi-Elliptical, Snubbers	Semi-Elliptical, Underslung
Braking System..	Two, on Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Two, Expanding on Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Two, Expanding on Rear Wheels	Two, Expanding on Rear Wheels

[2]The first two are double-deck buses; the remainder are single-deck buses.

others in the smaller towns. This consists of a gasoline engine direct-connected to a dynamo. The latter is electrically connected to an electric motor, which drives a worm rear-axle through a propeller-shaft. The control is by a speed regulator of the multiple-contact type, operated by varying the resistance in the shunt field of the dynamo, and by shunting the series field of the electric motor.

Storage-battery buses are used in Chicago for carrying passengers between the railroad depots. In another city the street railway is now trying out a battery vehicle in short-haul feeder service. This bus is fitted with pneumatic tires and makes about 12 m.p.h. in ordinary operation. The limited speed and battery capacity have worked against the battery-driven vehicles, although they undoubtedly have a field for short runs on good pavements.

The trolley bus shown in Fig. 6, a road vehicle using the overhead system of the electric railway, is making headway on this side of the Atlantic, as shown by recent installations in Toronto, Canada; Minneapolis, and Los Angeles, Cal. The system on Staten Island, New York City, is to be extended by the use of 15 trolley buses now ordered for the New York City Department of Plant and Structures. Some of these will be used on a City Island line, in the Borough of the Bronx, New York City.

Many of the trolley buses look like rail cars fitted with rubber tires. However, there has been a tendency of late to follow closely gasoline vehicle design, thus using standard parts and making it possible to change from one type to the other by installing a gasoline engine and transmission apparatus.

A front-drive design has been worked out for the buses used in Leeds, England. The electric motors are under the front, and can be removed for repairs. It is claimed that this construction improves adhesion and traction effort, reduces the consumption of electric energy and makes possible a 14-in. floor level.

The steam-propelled bus is not in commercial operation at present, although several experimental vehicles are being developed for such service. The Clarkson steam buses, which were popular in London because of their speed, quietness and ease of control, stopped operation in 1919, mainly, it is said, because of the increase in fuel cost.

Chassis Components for Bus Service

The principal parts of the bus chassis savor strongly of motor-truck practice, and bus individuality is gained by such features as more powerful engines, longer springs, battery and lighting generator and the like. Four-cylinder engines, with L-heads, are used on the majority of buses. A few designs for city service have sleeve-valve units. Some I-head types are being supplied, mainly for the fast inter-city work. Many operators are talking in favor of six cylinders, because of the smoothness and flexibility they assure.

The data given in Table 1 indicates that engine speeds of from 1400 to 1800 r.p.m. are common, at least for city buses. These speeds need not be maintained for long periods, but they are essential, as is also quick acceleration, for threading city traffic, passing other vehicles on the open road and maintaining schedules when stops are frequent. Quiet operation at slow speeds is essential also.

Governors are not common on vehicles designed for bus operation. A driver to whom the lives of passengers can be entrusted will not require a governor to safeguard the vehicle.

In many localities the law requires that the gasoline tank be filled from outside the bus. Tanks therefore are being placed at the rear of the frame, or at the side between the wheels. Another arrangement is shown in Fig. 3, where the tank is placed under the floor of the body, and filled through a port-hole in the side of the body.

Ignition is mainly by magneto, this being favored by many operators because a magneto is easy to replace. With the growing practice of installing a battery and a lighting generator, battery ignition is likely to be given more consideration.

Thermosyphon and pump circulation are both used on buses. The latter system is used in the majority of the designs listed in Table 1. Radiators with replaceable cores are being developed for bus service. They illustrate the tendency to provide parts that will decrease the time required for adjustments or repairs.

The exhaust system is usually fitted with a valve connection so that the body-builder can install a heating system. The end of the exhaust pipe, and this also applies to heater outlets, should be placed at the extreme rear so that the gases are discharged back of the body and not underneath it.

A smoothly operating clutch is an essential on buses. Jars and jerks not only increase maintenance costs, but they mean also continual discomfort to passengers. The service is so severe that this unit should be of liberal size.

Bus transmissions as a rule are of the four-speed sliding-gear type. They should be designed to stand up under a service involving 10 or more stops per mile and upward of 50,000 miles of operation in a year. The strain is intensified if the driver jumps to second or third speed when starting.

The maximum speed should be just high enough to maintain schedules from a transportation point of view, with a fair margin allowed to make up lost time. Passenger comfort requires steps between gears that are as nearly equal as possible. The low-gear speed must have power enough also to pull the vehicle through loose dirt, mud or snow.

Universal-joint design is especially important in the long-wheelbase bus. Bearing supports as well as extra joints are used to overcome the whipping effect of the long propeller-shafts. The angularity of these shafts is comparatively slight, so that full advantage can be taken of the shock-absorbing properties of the fabric joint.

The rear axle for the city buses is being made wider, of 72 and 74-in. gage, than on heavy motor vehicles. The underslung type of worm-gear axle is being used to a considerable extent to gain low floor-levels, while front axles with a center dropped sharply down between the pads and the yokes. are being provided for city service.

Frame construction and proportions have been influenced appreciably by the development of the city-type bus. The law limits the overall width, so the bus body has had to be lengthened to get capacity. The overhang beyond the rear axle and in the back of the frame is limited also by considerations of safety and good design. All this has led to the long wheelbase, which seems to be longer with every new design. Frame sections have had to be made deeper on account of the great distance between the wheels. The frames have been widened at the same time to care for the 7 and 8-ft. bodies they must support. The demand for low floors has again been taken out on the frame, which is made with a kick-up over the rear axle so that the central part can be kept close to the ground. On one new

design, shown in Fig. 7, this kick-up is found at both the front and the rear axles.

Owing to the severe service and the need for the utmost safety, the steering-gear and the brakes should be heavier even than for truck designs. Easy steering is important, to save the driver's strength.

A propeller-shaft brake is used on a number of bus models, undoubtedly because of its simplicity and the ease of securing equal braking on the rear wheels. On the Pacific coast four-wheel hydraulic brakes have been tried, for reasons of safety and to increase the life of the rear-wheel tires. The four-wheel air-brakes now being developed for heavy motor vehicles should be useful for large buses also.

Riding comfort is a bus requirement that has received close attention in the new designs. While underslung springs are installed principally to lower the bus floor, they undoubtedly improve the riding qualities of the vehicle. The change in wheelbase length that occurs on one side when one wheel strikes an obstruction is much less with the underslung construction, because the center of the rear axle is almost in line horizontally with the front eye of the spring, instead of being several inches below it, as with the conventional overhung construction.

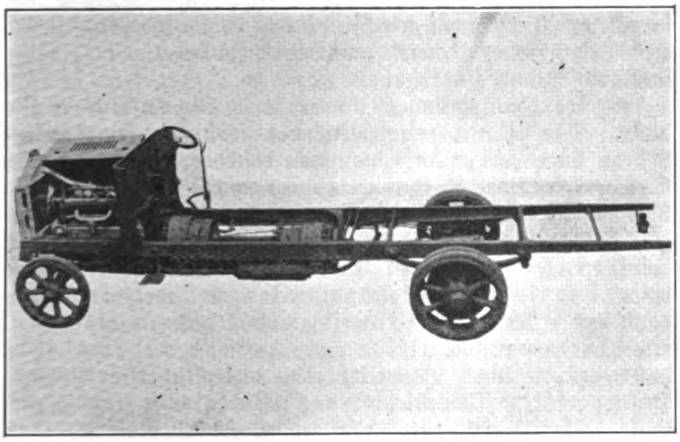


Fig. 5—Chassis of the Tilling-Stevens Gasoline-Electric Bus Used in England

Most of the bus designs have an auxiliary suspension that comes into play with heavy loads or severe road-shocks. The simplest is the so-called progressive spring, with extra leaves mounted below the main spring but at a flatter camber. Much the same effect is secured on the London buses by a volute spring attached to a bracket on the frame; or, the ends of the main spring are carried in sheaves, rubber blocks or small springs mounted inside the hangers. While these devices are used particularly on the rear springs, the front axle also is being better sprung, not only to protect the engine, but to carry the heavier loads thrown forward by the long wheelbase. Another method is to use the shock-absorbers developed for private passenger-cars. The pneumatic type, for example, is fitted to some of the California stages.

Pneumatic tires are almost universal on the small buses, but on the heavier types the large single pneumatic tires have the disadvantage of throwing the body high up into the air. This is being overcome by the application of dual tires on the rear wheels, which permits the use of six tires of the same size. The development of the "doughnut" design will help here, provided the brake diameters can be decreased to correspond.

Many operators in the cities are using solid or cushion tires, mounted on a resilient rim or cushion wheel. This combination seems to work out satisfactorily on fairly good pavements, although it has proved too heavy for the rear-axle mechanism on certain buses working over rough roads.

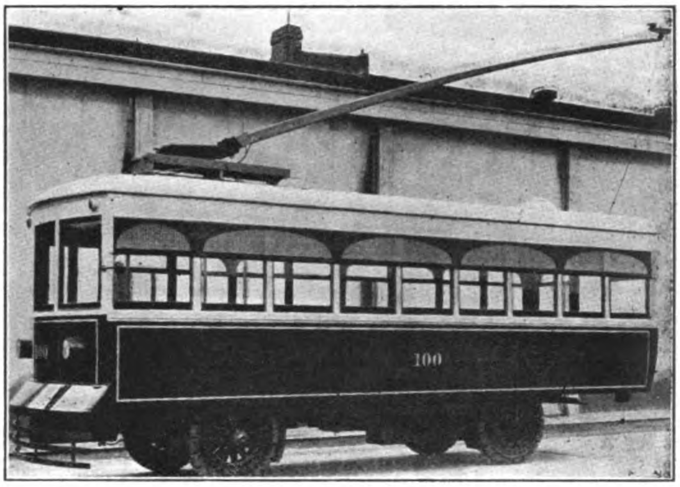


Fig. 6—A Bus of the So-Called Trackless Trolley Type

Bus Bodies

The old horse-drawn stages undoubtedly furnished the inspiration for modern bus bodies, and later technique seemed to be based also on the bodies built for motor trucks. At any rate, many of the first bodies were heavier than necessary for either economy or comfort. The timbers were too large, the ironing was too heavy and too much of it was used. Then, cheaper bodies were demanded; the pendulum swung too far the other way; bus bodies loosened up and became noisy; they began to fall to pieces and repelled instead of attracting passengers.

At first, quantity production of bus bodies was difficult because chassis mounting-dimensions were not standardized. Frame-widths and lengths, the location and the dimensions of rear wheels, the position of drivers' seats and controls and the height of the frame from the ground all seem to be different on every chassis. Bus bodies are now being turned out in quantities, however, partly because the sills can be altered to fit different chassis, and partly because of the great increase in the demand. The price and the weight have been decreased greatly, and, at the same time, there have been many improvements in detail construction.

The mounting on the chassis usually is designed to keep the floor as near the ground as possible. Many bodies now are set directly on the frame, so that the floor is only its own thickness higher. Longitudinal sills are placed between the frame side-members and supported between them. The part of the body projecting beyond the frame may be carried on cross-straps that extend the full width and are supported at the ends by pressed or built-up brackets riveted to the side-members; or, the support may consist of channels carried across

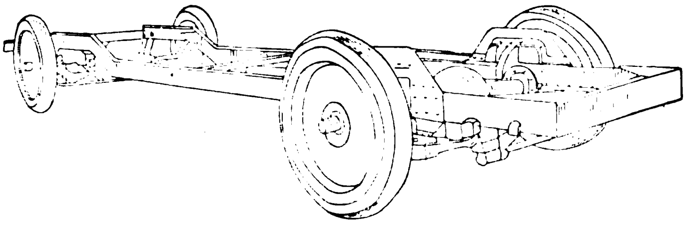


Fig. 7—Bus Chassis Having a Kick-Up at Each End Resulting in a Floor Level That Is Only 19 In. Above the Ground

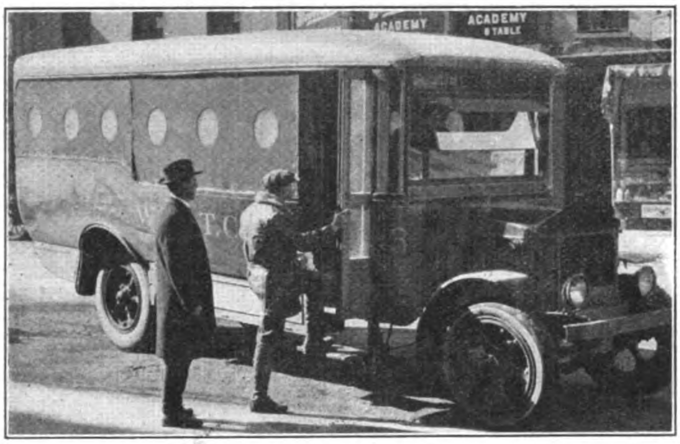


FIG. 8—CONVERTIBLE BODY USED FOR ALL-YEAR SERVICE, THE ROOF POSTS BEING ENCLOSED BY DOUBLE CURTAINS IN THE WINTER

under the main frame and bent so as to carry the outside edges of the body. These are riveted to the side-members by structural shapes.

In general, bus bodies are built up on a hard-wood framing, of ash or oak. Dies for the all-metal body would be too expensive, even should such construction stand the weaving and twisting to which semi-flexible chassis-frames subject it.

Panel material may be wood, sheet-metal or composition. Sheet-metal is applied usually as a sheathing, as is also sheet-aluminum. A wood veneer, protected on the outside by sheet-steel, is sometimes used. This material, it is claimed, has the resiliency of wood, and the easily finished surface of sheet-metal. This veneer, or plywood, is also made up as roofs, or used as a ceiling to furnish a smooth surface over roofs of tongue-and-groove board.

Climatic conditions and local preferences both seem to determine the type of body used in any given section. The open body is popular in England, where it is fitted to so-called motor coaches or char-a-bancs. The seats are unbroken across the body and are reached by individual doors. As a rule, the roof can be folded back. In this Country such a vehicle is favored in the South and on the Pacific coast; also, it is used in large cities and in the National parks where sight-seeing is the reason for presence of the bus. Sight-seeing buses work mostly in warm weather, but in some parts of the Country, as in New York State, an open convertible body, such as is shown in Fig. 8, is used the year around. This has a permanent roof supported on pillars that are built into the sides of the seats. The open sides are covered by double curtains during the winter.

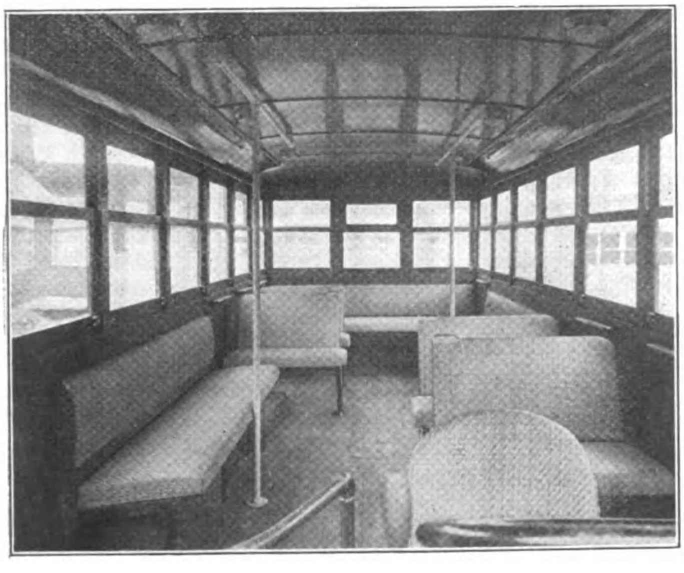


FIG. 9—THE SEATING ARRANGEMENT OF A BUS FOR CITY SERVICE WITH VERTICAL AND HORIZONTAL BARS TO TAKE CARE OF THE STANDEES

The closed type of body is found generally through the North, East and Central parts of the Country. This body, with two longitudinal seats, is popular with operators because of its wide central aisle, large standee capacity and good ability to load and unload passengers. The longitudinal type is not so comfortable on long runs of 30 min. or more, where the passengers settle down and are not boarding or alighting at frequent intervals. In country service and for large single-deckers in city or inter-city work, it is being replaced by an arrangement such as is shown in Fig. 9, in which both cross and longitudinal seats are used.

COMFORT AND CONVENIENCE FACTORS

The success of most transportation systems depends upon their ability to attract a group of riders who travel not of necessity but because the service given is attractive and its use is a pleasure. This profit-making group is attracted by good-looking buses with pleasing lines, and also by steps, doors and seats that are comfortable and convenient.

The first "convenience" factor is on the outside of the body. The color of the paint has transportation value. Where buses are numerous, each route or line can use a distinctive color, so that patrons can recognize it easily. The electric railways operating buses are naturally painting them the same color as their rail equipment. In Bridgeport, Conn., all of the buses are painted the same color, but the belt rail is painted a distinctive color on each line. This stripe running around the outside identifies each route clearly.

To secure high visability, the buses of the Toronto Transportation Commission are painted a sagamore red. It is believed that this color induces both employes and passengers to keep the paint on the vehicles in better condition, and that accidents are reduced because the operators of other vehicles exercise greater care.

The would-be passenger, having picked out the right bus, is now ready to enter; but first he must allow passengers to alight, since it is not practicable to provide door space for two conflicting streams of travel. If he is to enter comfortably the actual step-height will not be more than 15 in., although from 4 to 6 in. can be gained by curb-loading. Tests on trolley-cars have proved that 15 in. should be the maximum step-height. Three steps, with one 16-in. and two 14-in. risers, showed an average loading and unloading time of 2.50 sec. per passenger. This was cut to 1.75 sec. per passenger, however, when the change was made to one 15-in. and three 10-in. risers, each of the four being slightly wider than the three used in the first test.

The width of steps should be at least 12 in., although this often is limited because of the necessity of keeping within the body lines, and clear of the chassis frame. The steps should be shod with safety tread, for the good of the passengers and to prevent wear. In current designs, the steps are of permanent construction, the outside folding type can be used but is not particularly satisfactory as regards durability or safety. Two steps only are needed with the low-frame buses, and these can be arranged to fit in the body very nicely.

The simplest form of door, assuming that the entrance is covered at all, is the collapsible gate used in

Southern California. But more substantial construction is required in many parts of the Country. Several types of door are available. The sliding door frees the full width of the entrance, but makes necessary an expensive opening in the body. Two-leaf folding or jack-knife doors are therefore in more general use. These fold inward as a rule, as the outward folding type is frowned upon by the law-makers. Hinges at the front permit carrying the opening mechanism across to a handle that can be operated by the driver. The opening of the door may automatically turn on the step light at night.

The service door seems to be one of the most difficult parts of the body to keep in good working order. The trouble is usually in the door itself, although the manual operating mechanism may get out of order at times. On many buses the driver spends a good part of his time, often when he should be watching the road ahead of the moving vehicle, in trying to force a sticky door open or closed.

Doors should be at least 30 in. wide, although in practice they vary from 22 to 35 in. This can be compared with the exit-entrance of the one-man type of trolley-car, which is about 52 in. wide; but, as was said before, space is too valuable in the bus to provide a door that will carry two streams of passengers.

Seats in the bus should be arranged in accordance with traffic conditions rather than the design of the vehicle chassis. The latter sometimes has to control, however, as in the placing of longitudinal seats over the wheel-housings. The result is a well at the rear of the vehicle, which must be reached through the neck of the bottle formed by the central aisle. This well is sometimes needed, but it is often used when the space should be filled with seats. Fig. 10 shows seating layouts for city and suburban service.

The standard cross-seat is 15 in. wide and 32 or 34 in. long. The central aisle is from 14 to 24 in. wide. The 34-in. seat is the better, since the narrower seat will not in reality allow the use of wider aisles. At the same time it may be uncomfortable for two persons. The real aisle-clearance is between the hips and shoulders of the seated passengers. With the 15-in. width, seat centers should be at least 28½ in., which gives only about 11-in. knee-room. These seat centers should coincide with side-post centers.

Seat construction has settled down to the spring type for cushions; springs are sometimes, although rarely, used in the backs. A patented seat, which is said to have given satisfaction on a 30-passenger bus, is supported on two pedestals, each of which is a pneumatic cylinder or dashpot. This really amounts to a shock-absorber directly applied to each seat.

Seat covers are of rattan ordinarily, with plush or leather for de luxe designs. Rattan is low in first cost and can be kept clean more easily. It is somewhat slippery, but this can be taken care of by the seat design; by raising the cushion at the front and recessing the back slightly. This gives more knee-room and comfort, and tends to hold the passengers firmly in place, even over rough going.

Floor coverings of carpet or linoleum are found on some vehicles, but for wear and hard service the floor is slatted, in the aisles and between the seats. The slats can be attached to flat strips of steel, from which they can be removed when worn. This scheme, which has been used in England, is said to make cleaning easier also.

The driver's comfort is equally as important as passenger comfort, and is taken good care of in the new de-

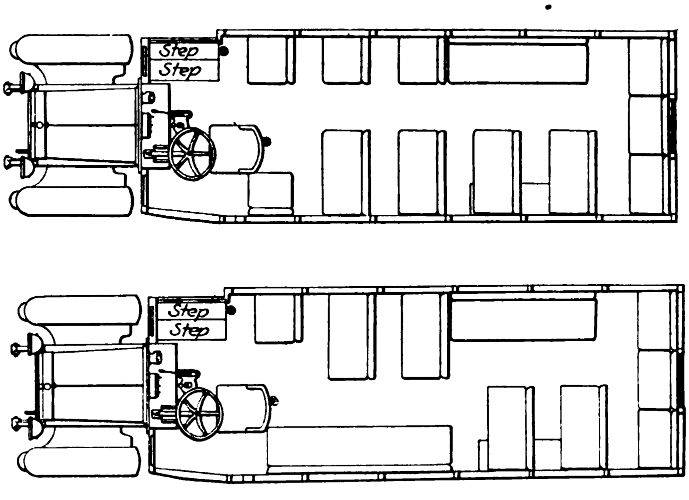


Fig. 10—Seating Layout of a 20-Passenger Bus (Above) for Inter-City Service and (Below) for City Use

signs. The seat is of bucket shape with good springs, and sometimes has an adjustment for height. A railing is placed so that passengers cannot crowd against the driver, wind and sun-shields are attached on the outside and a curtain or bulkhead prevents light from the rear being reflected on the wind-shield.

Heating, Lighting and Ventilation

A heating system such as is shown in Fig. 11, using the exhaust gases from the engine, is installed on most buses put out within the last year or so. Previous to that, however, many buses carried small stoves during the winter months, which of course was unsafe and unsanitary. The first exhaust systems were makeshifts, with the gases led through pipes inside the body. While this was often effective, or at least provided satisfactory heating, it was a clumsy arrangement, hard to control and difficult to keep gas-tight. It was used, however, because operators claimed they could not get heaters of sufficient capacity, at least for large buses.

The problem seems to have been solved by the use of two heaters, placed close to the front of the body. The

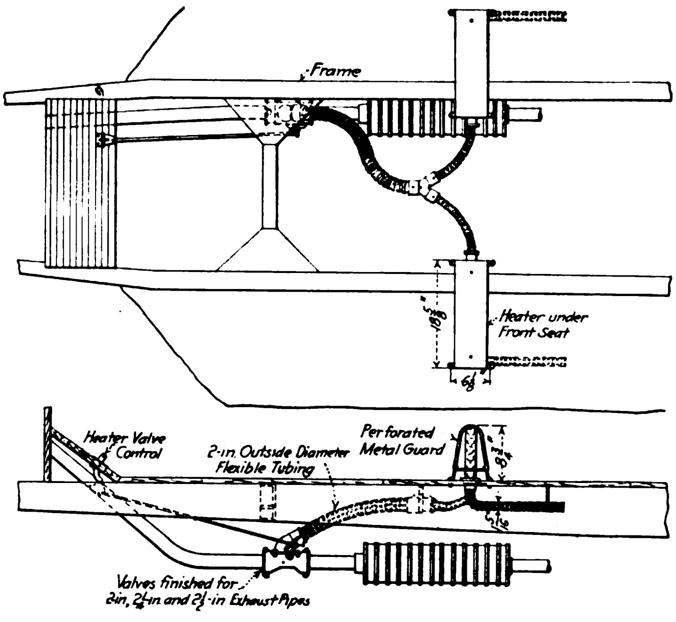


Fig. 11—Installation of Heaters in a 25-Passenger Bus Showing Connections of Flexible Tubing and Valve Control Operated by the Driver's Foot

inlets for these join in a single connection to the exhaust pipe, through a valve controlled by a lever inserted in the floor-boards near the driver. Because of the compact arrangement, no provision is made for insulating the tubing leading to the heaters.

Good interior lighting is essential where passengers have the evening newspaper habit, although in country service, where the buses are used at night only on Saturday, it may not be necessary. Good lighting is needed also for displaying advertising cards, which promise to be a considerable source of income to the bus operator.

The 2-cp. bulbs used in so many buses may help in collecting fares, but are of little service to passengers. It is recognized now that larger generating and battery capacity must be installed. The 300-watt generator system for the Fifth Avenue Coach Co. buses, although in some respects a special job, indicates that the larger equipment is available. The 12-volt battery system seems to have found a place, although the 6-volt design is used widely.

The present tendency is to mount dome reflectors on account of the low head-room. Prismatic and opal reflectors have been designed however for bus service. These require practically no more head-room space and are said to be much more efficient than the domes. The large buses are being fitted with six to eight bulbs, of from 21 to 32 cp.

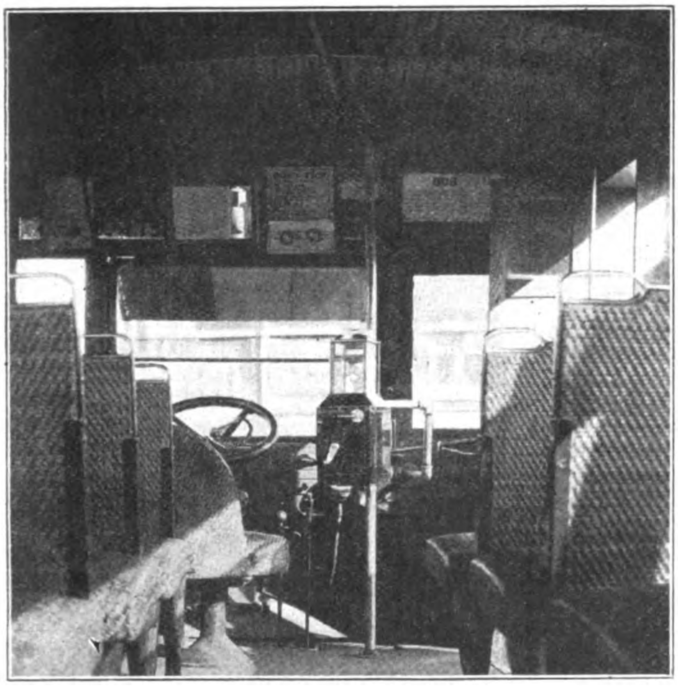


FIG. 12—REGISTERING TYPE OF FARE-BOX MOUNTED ON A STANCHION AT THE RIGHT OF THE DRIVER

Identification at night is a requirement peculiar to the bus. Many designs are fitted with a bulls-eye, of special color, at the top of the roof in front. This permits the would-be passenger to pick out the bus when it is some distance away, and the bulls-eye lights up the destination sign when the bus comes closer. Purple and green running-lights serve the same purpose.

Bus ventilation, in its simplest form, depends upon the service door for the admission of fresh air, and upon vents at the sides or the rear to exhaust polluted air. Most doors admit a slight amount of air even when closed, unless they overlap at the top and bottom, when they are practically draftproof. Air is admitted also through openings across the front of the body, above the driver.

The latest practice is to insert ventilators along the center of the roof, with a screen or grating in the ceiling. These ventilators may be automatic in operation, not requiring any adjustment; rain cannot enter but the air exit is always ready for service. Windows are being equipped with adjustable curtains and sash, safety catches, rubber weather-strips, anti-rattlers and upper stationary sash with Florentine glass. Wire grating or wood slats are placed outside to keep each passenger wholly inside the bus.

Pockets for windows are of two types. The overhead pocket requires good head-room, and heavy top construction with a higher center of gravity. Then there is the side-pocket type that takes up valuable space needed for seats and aisles. The overhead opening seems to be favored, undoubtedly because the windows are easier to handle.

FARE-COLLECTION DEVICES

The collection of fares, while more of a traffic than an equipment matter, is giving much trouble to bus operators. The necessity for simplicity and accuracy has led to the application of mechanical devices, such as illustrated in Fig. 12, to save work for the driver, insure his honesty and speed-up service.

A money-changer, fastened to the side of the bus or at some other convenient place, is the first step; next some type of fare-box, with perhaps an overhead register that the driver pulls to record fares. The latest arrangement is what may be described as an instantaneous registering lamp-post type of fare-collection device. This records each fare as deposited, and is driven by a flexible shaft connected to one of the accessories on the bus engine. The use of such a mechanism should make unnecessary the regulations that forbid the driver to collect fares while the vehicle is in motion.

STATE AND LOCAL REGULATIONS

Laws and ordinances by the hundred, almost by the thousand, have been passed to "supervise and regulate" the bus. The safety and comfort of the traveling public are protected by a host of these regulations, many of which are more honored in the breach than in the observance. In common with other users of motor vehicles, the bus operator is limited as to weight, dimensions and speeds. In addition, he often must pay special taxes that influence the type of equipment.

To assure safe operation, presumably, the carrying capacity of the bus is limited. In some States, in New Hampshire, for example, the manufacturer's rating is taken as the legal limit, and no standees are permitted. New Jersey allows 50 per cent of the seating capacity to be carried as standees. Connecticut allows two more than the number of seats provided. The District of Columbia permits one standing passenger for each 1.5 sq. ft. of floor area, this being calculated on the area remaining after deducting the space occupied by seats and 10-in. knee-room for longitudinal seats. In Maryland the maximum carrying-capacity is the total length of seats divided by 16, except when the result exceeds the carrying capacity. This method, in effect, bars out standees.

An emergency door that can be opened from the inside in case of accident is required in a number of localities. Other equipment insisted upon, for safety reasons, includes speedometers, inside lights, extra tires, skid chains, warning device, and a fire protection device.

The requirements for heating, lighting and ventilation are usually indefinite; when enforced they depend upon the interpretation of the authorities. Closed buses in the District of Columbia must have a minimum temperature of 40 deg. fahr. between November 1 and April 1. Interior lights, of 2 cp. or more, are demanded in Ohio and other States. In Maryland each bus must have two ventilators.

In many States the taxes are based upon seating capacity, and the larger vehicles pay more per seat than the smaller. Florida exacts from the buses used as common carriers a fee of $5 per seat for 7-passenger or less capacity, $7.50 per seat for 7 to 17-passenger capacity, and $10 per seat for more than 17-passenger capacity; in addition there is a tax of 75 cents per 100 lb. of gross weight of the vehicle and the load. Connecticut and Maryland also have systems whereby the fee shoots skyward for the larger buses.

The regulatory authorities may even supervise the purchase of equipment, approve drawings and specifications and compel the improvement of those already in service. It is too early yet to judge what the result of this supervision will be. At present it seems to make for better buses. With wise regulation, and with improved highways that will permit lighter vehicles and more economical operation, the bus, already with such a wonderful development, will have an even more wonderful future as an instrument of transportation not only on the streets of our cities but also in inter-city service.

THE DISCUSSION

R. E. PLIMPTON:—What I have tried to do in this paper is to bring out the transportation viewpoint, the viewpoint of the operator who is operating mainly the smaller lines. What I call here "the country bus" is really the small vehicle that often is operated over rough roads or in service where the traffic is light.

The conclusion of the paper refers to regulation very briefly, only as it affects the design of the vehicle, but that thing is coming more and more. President Bachman referred to the competitive feature, or possibilities of the bus, but I do not see any competition there at all. Mr. Green has referred to the bus as a public utility. I think that our institutions are firmly grounded on the regulation of public utilities and that the bus is coming more and more into that class. Already, more than half the States in this Country have some form of state regulation and, while they started in with restricting the lines and the fares charged, they will continue that and will get more into equipment, not only for safety, but for the comfort of the passengers; and, as that goes on, I think engineers and builders ought to keep more closely in touch with the regulatory authorities, so that they really will be informed. At present, the State authorities are taking many steps connected with design that they are sorry for when they realize what the steps mean. They require certain types of body, or certain types of tire, perhaps, and then, after they pass the law or the regulation they find out that it will not work, and that is not a good thing for anybody.

To show what some of the smaller operators are up against when they go into the business and also to show the viewpoint of a small city, I will read an extract from a Newburgh, N. Y., newspaper. They have about 50 buses now in Newburgh. Outside of New York City, I suppose it is one of the largest centers of bus operation in New York State, in spite of the fact that it is a comparatively small city. The city lies on the Hudson, so that one side of the territory is cut off; but, in spite of this, they have all this bus operation. This newspaper says:

> Any one contemplating going into the bus-transportation business will do well to figure the ultimate cost before going just far enough to have to turn back with a loss on his hands. First, there is a cost of one bus, which may be anywhere from $1,000 to $10,000. There is required a liability bond of $10,000, which costs $592. In addition, there is a New York State license fee, which costs $67.50. There is also a Federal license fee of $20, and $225 for property insurance on the bus. Thus it will be seen that it will cost, exclusive of the price of the bus, $904.50 before a fare has been collected, to embark in the bus business. The bus-transportation business is no plaything for men who have any regard for their money. The day when a man could run a tumble-down car in and out of Newburgh and call it a bus because no one cared enough about what he called it to put up a kick is probably gone forever. A man involved in the bus business in this section is regarded as a transportation unit and that line of industry is evidently not to be meddled with by those not knowing what they are about. If you can afford it, there is money in the bus-transportation line. If you cannot afford it, there is not.

Chairman C. F. Scott, of the Meetings Committee, told me some time ago that he took it as a matter of fact that practically every city and town in this Country had some form of bus transportation. I am inclined to amend that by saying that practically every city and town has had experience with bus operation, which is a different thing, because the dead bus lines mount up into considerable numbers. Of course, that applies in the city also. I think the reason for that is not altogether due, although it is partly due, to inadequate design of the vehicles; it is due also to a lack of transportation knowledge on the part of the operators. It is only a matter of time before they will acquire that knowledge as a result of these failures.

MERRILL C. HORINE:—Does not the second item in Table 1 of Mr. Plimpton's paper, the approximate entrance-height in inches, really mean the floor height above the ground? The Mack engine bore is 4¼ in., and the front axle is of I-beam construction. If he means by entrance height the height of the floor at the entrance, that will make it clearer; but by entrance height I should say is meant the first step, which of course cannot be as high as shown.

G. A. GREEN:—Referring to Mr. Plimpton's remarks in regard to the possible future employment of six-cylinder bus engines and six-wheel buses, our theory is to concentrate on the reduction rather than on the addition of parts. As to the six-cylinder engine, for normal bus operation in the average American city, we believe a four-cylinder engine of about 300-cu. in. capacity is adequate; and assuming proper design, speed and load variations can be taken care of by carbureter adjustments or gear-ratio modifications. If this assumption is correct, then certainly it is folly to add two more cylinders, thus increasing first cost, weight, number of parts and fuel bills. Of course, these remarks apply regardless as to whether the engine design is poppet or sleeve valve. But it should be pointed out that the much desired silence and smoothness of operation are comparatively easily achieved with four-cylinder engines of the sleeve-valve type.

In regard to the six-wheel bus, here again there are additional parts and, in all probability, additional weight and cost. In our judgment the value of this construction has not been proved. In short, it would seem a more logical plan to concentrate our energy on the improve-

ment of the four-wheel system that has given and is giving an exceedingly good account of itself, rather than expend any great amount of time and money on the development of a theory that in our opinion has not yet emerged from its early experimental stages.

In Mr. Plimpton's paper, a suggestion is made in regard to painting buses different colors for different routes. From our viewpoint this would be unwise, for with a public utility there should be the fewest possible restrictions placed on vehicles. The underlying thought is that any vehicle should be capable of operation by any driver on any route at any time.

GEORGE W. CRAVENS:—I think this whole matter comes back to the principle stated by Mr. Green; first, know your business, and then design something suitable to your purpose.

I am interested in several companies, among which is one that is experimenting with and will bring out a steam bus. There is no doubt that there is much to be said in favor of steam from the standpoint of acceleration and smooth operation, but I will not take time for that. However, in this bus-transportation problem you have a similar problem to that of the street-railway man in the last analysis. You must take care of your traffic at the least cost per passenger-mile.

In our investigation of motorbus requirements it seemed to us that standardization of body sizes and wheelbase boils down to approximately 25, 40, and 55 passengers as being good standard sizes. How nearly does that agree with Mr. Green's and Mr. Plimpton's determinations?

Secondly, I believe the time has come when it is necessary to clarify the bus situation. That has been done in the passenger-car and to a certain extent in the truck and tractor fields. It is time for a company to go into the business of building a vehicle from the ground up that will be consistent as well as suitable for its purposes. The time of makeshifts and of trying to redesign something you have to fit a new condition you know nothing about has passed. I believe that eventually some company similar to the Fifth Avenue Coach Co. will go into the business of constructing buses for sale that will be built along standard lines, and you will take them or go without them, the same as is now necessary when we buy a truck or a passenger car.

MR. GREEN:—We have analyzed the seating-capacity situation as nearly as we can, and as a result of our analysis we are building vehicles carrying, with the single-deck bus, from 25 to 29 passengers, and with the double deck, from 50 to 55 passengers. Undoubtedly there are fields for other vehicles, but we think the largest field is for buses of these capacities. Of course, there is also a field for the so-called speed wagon, seating around 15 passengers, but with vehicles of this character, the driver's wages consume a large proportion of the revenue, so the natural tendency will be to increase the seating capacity gradually to make up for this.

CHAIRMAN B. B. BACHMAN:—I used the term "speed-wagon" in the way that it has been used in advertising literature, to describe the vehicle of 1500-lb. and under carrying capacity that is used for general-utility delivery-purposes, rather than for passenger transportation.

HORACE L. HOWELL:—With reference to the question as to the proper seating capacity of a bus, I should consider that it would depend chiefly upon the character and class of service; that is, if it were a high-speed proposition, the single-deck high-speed type of bus would be the answer, with a seating capacity similar to that of the Fifth Avenue Coach Co., 25 or 29. Should the bus be required for urban transport service at slow speeds, a double-deck structure would seem more proper if the overhead clearances would allow it. In this case the number of riders would govern the seating capacity.

The first successful type of bus used by the London General Omnibus Co., London, England, was known as the B-type bus. It had a seating capacity of 34 passengers. That bus was used for a number of years and approximately 1000 of them were commandeered by the British Government for transport service in France during the recent war. A number of them are in operation today; but, owing to the growth in the riding habit and the consequent depletion of the fleet, the construction program of the London General Omnibus Co. called for a larger vehicle with a greater seating capacity. The company therefore established a program for its future operations, to meet the increased riding habit, by constructing the K-type bus having a seating capacity of 46 passengers and, later, the S-type bus having a seating capacity of 54 passengers. The operating statistics of the company show that the cost of operation for the larger passenger-carrying vehicle is not much more than it was with the B-type bus and it may be expected that, on account of the added seating-capacity and because of the comparatively small difference in operating costs, the consequent operations will be much more successful from a financial standpoint.

CORNELIUS T. MYERS:—In reference to capacity, it would seem that wherever bus lines have become established and their patronage grows, the larger seating-capacities can be employed profitably. In a great many districts very likely we find that the smaller bus will be the first installation, not only on account of first cost but because the regular service cannot be given as traffic has not been developed.

MR. PLIMPTON:—One of the most encouraging things in connection with the whole bus situation is that there are several companies now specializing in the construction of vehicles to be used as buses. We must bear in mind that although the traffic is light there is a great volume of other work done by the operator of the country bus. He very often carries light packages and freight; he may deliver milk or newspapers or do errands for people; and he usually has some other business. But for the territory he serves he performs a real service. In Italy and France this has been recognized and Government subsidies have been applied to this "country service," just to keep the buses going.

In connection with Mr. Horine's query, what I had in mind was not the step-height, but the approximate floor-height; that varies somewhat, but that is what the figures were intended to cover.

Mr. Cravens suggests three sizes of bus for standardization, the smallest being of 25-passenger capacity. Undoubtedly, there is a considerable field for a smaller vehicle of say 16 to 18-passenger capacity, and it should be given due consideration.

Discussion of Papers at the Chicago Service Meeting

THE discussion of the papers presented at the Chicago Service Meeting included written contributions submitted by members who were unable to be present and the remarks made at the meeting. In every case an effort has been made to have the authors of the several papers reply to the discussion, both oral and written, and these comments, where received, are included in the discussions. For the convenience of the members, a brief abstract of each paper precedes the discussion, with a reference to the issue of THE JOURNAL in which the paper appeared, so that members who desire to refer to the complete text as originally printed and the illustrations that appeared in connection therewith can do so with a minimum amount of effort.

In addition to discussion given below, the discussion of the paper by J. F. Page entitled The Car-Owner versus the Service-Man, is printed elsewhere in this issue of THE JOURNAL, following the paper itself.

A SERVICE-MAN'S CRITICAL ESTIMATE OF AUTOMOTIVE ENGINEERING

BY B. M. IKERT

AFTER generalizing on the need for greater consideration in automobile design for service and maintenance requirements, the author discusses the accessibility of car parts at some length with the idea of pointing out difficulties encountered by service-station mechanics when parts are inaccessible, this having a bearing also on the length of time required for repair work and the consequent increased cost to the car owner.

Specific instances are given and illustrated in which improvements in design could be made to obviate trouble. These are inclusive of cylinders, cylinder blocks, pistons, bolts, cap-screws, nuts, valves, dashboard instruments and general take-up adjustment. Special emphasis is placed upon certain inaccessible parts that necessitate excessive dismantling.—[Printed in the April 1922 issue of THE JOURNAL.]

THE DISCUSSION

PRESIDENT B. B. BACHMAN:—There has not been the proper contact between the man who designs, the man who builds, the man who sells, the man who maintains or services, and the man who uses the motor vehicle. Reasoning from ordinary human affairs, such contact as there is and such influence as is brought to bear will have their greatest weight dependent upon the aggressiveness of the contender. The man who is closest to the designer is the man who is building, usually, and the man that is almost as close, particularly in the present-day markets, is the man who is selling. The man who is servicing is not so close, and the man who is using is considerably farther away. The man who is building has fairly complete access to the designer, and the man who is selling gets contact with the man who is designing with a reasonable degree of facility. The service-man is farther away, but he does know the individual who is responsible for designing. Mr. Ikert's paper deals largely with passenger-car operation. I am interested more particularly in the transportation of merchandise, but the underlying fundamental principles are the same. The man who is using should remember that he ought to have direct access to the man who is designing, through the service-man and the salesman.

All design work is a matter of compromise. I believe the machine has never existed and I doubt whether it ever will exist that is perfect from any one standpoint. It will be effective from a theoretical viewpoint in some one particular or other; from a production viewpoint for some theoretical reason, from a selling standpoint for theoretical and production reasons, and from a service standpoint from some of the other reasons. The man who is designing must take all these various elements, bring them together and study each individual item. Because some designers have not had a proper conception of the relative values of these different items, there has been a predominant influence in their work, due to the fact that some of these factors have brought a greater influence to bear on the designer than some of the others. If the designer could spare time enough from his other duties to canvass the Country, come into contact with the men on the firing-line and get their viewpoints, I believe more progress would have been made along these other lines. We believe that we can effect a contact between the user and the manufacturer, grouping with the manufacturer the designer and the producer, so that these fundamental problems will be in evidence and the designer will get a picture of some of the practical things that have not had much consideration.

Mr. Ikert mentioned the aging of cylinder blocks. It seems to me that this is not so much a service as a production problem. It has a bearing on service matters, as he outlined, but fundamentally, it seems to me, there are many factors of design that enter into it which vary the requirements. At a recent meeting the aging of cylinder blocks for 12 to 18 months was discussed. There probably is a considerable division of opinion as to what is the best method but from the production standpoint we must recognize that it is not good business to put into an automobile a casting that has been expensively reduced to good form and then, due to changing temperature conditions, have it leave that form in a radical degree.

There is room for considerable improvement in the matter of wiring systems from the motor-truck standpoint, along the lines of electrical application. The suggestions that Mr. Ikert made with regard to the standardization of wire colors seem to me to be well worth considering.

Mr. Ikert indicated a thought with regard to the so-called service engineer, one phase of which I wish to emphasize, in connection with the relation of the service-

man and the engineer. I believe that it is not necessary for the man doing service work to tell the engineer the trouble and how to fix it. That is one of the things that has held back the development of design from a service standpoint. The man doing service work has difficulty enough to fix troubles, and I believe that he is not usually possessed of the necessary qualifications to enable him to analyze, and to suggest remedies. It seems to me that this is fundamentally a part of the engineer's job. If a man comes to him with an honest criticism of something that is wrong, the engineer ought to be able to find a remedy for it.

A. H. PACKER:—Several years ago I was with a large electrical company on the service end of ignition, starting and lighting equipment, and later I was in its engineering department. In the engineering department, although rated as an engineer, I had the standpoint of the service-man. In the discussions I took one side of the argument and the other engineers took the other side. To illustrate, in laying out an ignition outfit, it was my suggestion that the cams be made in a certain way, which was an easy one, so that the ignition unit could be changed from a right-hand to a left-hand unit, but I was overruled. I was sent to install the first sample equipment. The engineering department's instructions called for a right-hand unit, which was sent, but I found at the factory that a left-hand unit was necessary, showing that the more universal a piece of apparatus can be made, the more likely it is to suit the engine for which it is intended. That may not come up after getting into production but, on other hand, the service-station would need to carry two types of cam in stock instead of one, for each size of engine; four, six and eight-cylinder right-hand and left-hand rotation parts would be necessary. Such difficulties are due to a lack of understanding of the service-man's problem, and a lack of appreciation of how serious it is. Another recent illustration is that of a designer for a large electrical company. He designed a generator and the test was absolutely satisfactory; but when the generator was put out into service, complaints began to come in. A practical mechanical difficulty had developed which nullified all the fine electrical design. If the designer had had an open mind. he would have gone into the field and got pointers from the service-man. Not having the service-man's viewpoint, he had no constructive ideas to remedy the defect. The result was that the company's reputation suffered because of an incidental feature that entirely discounted the advantage of the good engineering.

R. B. MAY:—There are a few relatively simple ways to get such information as has been suggested. Any service manager of a motor-car company or a manufacturing plant knows what parts are used in repairs and maintenance work; but has any analysis ever been made to ascertain the proportion of expense on such repairs between the value of the repair parts and the labor put upon them? When new apparatus is designed, if a time study were made in each manufacturing plant of each repair operation, one would know what the expense of ordinary repairs would be, and be able also to instruct dealers and service-men as to the best way to accomplish those repairs. We have had to do that in our own work. We find that it is possible to cut down the time for repairs to about 25 per cent of the ordinary time, when an accurate and detailed study has been made. That information could be distributed by each manufacturer as a part of his service to his various dealers. In that way the time for repairs could be cut down materially.

It has been estimated that car dealers change every 2 to 2½ years, which means that the bulk of service work on cars over 2 years old is done in independent garages that are not informed by car and accessory manufacturers at present of the best methods of maintaining the cars and the apparatus. The demand for such information is shown by the manuals that are sold at exorbitant prices. For instance, along electrical-accessory lines, one manual that simply takes the place of a manual that could be prepared by manufacturers sells for about $25.

J. C. GOREY:—For the past 10 years I have been engaged directly in handling assembled clutches, transmissions, universal-joints and axles that are used in assembled cars and trucks. The engineering departments at the factories are overburdened with questions daily. If the man who is in the field to repair and remodel different units would stop and consider the wonderful growth of this industry, he would try to remedy some of the troubles and lessen the number of complaints with which he is annoying the engineering department.

I am often asked questions. For instance, a man asked me how to obtain two beveled bushings for a certain car. I told him to go to the company that handles the same model of car. He wondered why the man from whom he purchased the car did not tell him that. We are wrong when we have not furnished the owner of the car with such information. We are afraid to tell him direct that another make of car has the same unit in it. He will find it out eventually and will ask why he was not told it in the first place. All there is to this whole thing is contact with the owner of the vehicle. We should tell him that he can get this part or that part where he pleases. Let the engineer fathom the greatest trouble and let everybody else do his part.

J. WHYTE:—In regard to the use of colored wires for the electric wiring, I have been developing a component part of a car that includes the electrical equipment as a standardized unit. We were very anxious that all the wiring in that equipment should be easily traceable from one end to the other, and decided to use different colored insulation for the wires. We desire also that the coloring should correspond with that used by the chassis manufacturers and made a survey as to how many manufacturers were using wires having insulation of different colors. We find that there are no two makes of car in which the same colors of insulation are used for any given specific purpose. We, therefore, had to standardize on a group of colors for our own work. I think that we ought to have a recommended practice on the colors of insulation to be used for the various electrical circuits in motor vehicles.

There is not a chassis manufacturer that I can recall who does not protect a wire with either metallic conduit or circular loom, and it is a physical impossibility without making a test on every circuit to find out where the other end of the wire belongs. The only drawback that I can see to using insulation of different colors is the fact that there are some different circuits that use the same size of wire. For instance, on the average car, we probably have seven circuits using No. 14 or No. 12 wire and, if the total wiring requirements are divided into seven parts, some purchasing agent will find out that if he buys it all with one color of insulation he can save money and the engineering department will be immediately overruled. In the past, much of this kind of trouble has been caused by purchasing departments in saving small sums in such ways.

Another trouble with regard to service on any component part or on any car is that there is not, in my ex-

perience, full co-operation between service and engineering. In designing a new unit, after we had it completed we tried to get in touch with every service-man we could find, realizing that the ultimate consumer is our best advertiser. We invited those men to come in, go over our unit and give us their criticisms. It meant some radical design changes to conform with some of the ideas and criticisms that we obtained. We have been surprised to find that a unit that meets the ideal of a service-man is the cheapest to produce. In other words, a unit that is economical from the service standpoint is also economical from a production standpoint; any part that is easy to dismantle and replace is easy to build in the first place.

Many car engineers neglect to figure service into the cost of production. The average car-factory does not get the full price for all of its service parts. The manufacturer must make concessions to keep the large distributors, dealers and owners satisfied. How many car companies have ever calculated the annual cost of such concessions and added that to the production cost of the vehicle? In many cases this cost would offset some of the economies that they think they have made by cheeseparing in the initial stages.

In regard to the use of service records and the improvement of design, the service departments of a few car-companies send a daily digest of major troubles to the engineering department. I know of engineering departments where these are carefully filed, but that is all that is ever done with them. If these digests were analyzed by some competent assistant in the engineering department, they would form a fine volume of data for refinements and improvements that should go into car design from year to year. Such a record can be obtained in no other way than from the performance of the car in the hands of the actual owner. The average owner mistreats his car. Cars that will perform through their engineering and development periods with absolutely no trouble, show some of the most weird performances when they get into the hands of the average owner. A digest of those troubles is invaluable in the engineering department if it is properly used.

CHARLES M. MANLY:—In my address at the 1919 Annual Meeting, I made a prediction, that I think is working out all right, to the effect that inside of 10 years we would have as many engineers working on the operating end of automotive work as we at that time had working on the designing and constructing end. Mr. Ikert's paper is very timely and should create much interest. I think that we are rapidly approaching the condition where we will have many more engineers in the operating end than we have had heretofore, due to several causes. First, the demand for designers is not so great now as it was a short time ago, and some of the designing men naturally will work over toward the operating end. But I think the biggest factor in the demand for operating engineers is that, especially in connection with merchandise transportation, the operating companies will have their own transportation engineers and require that the design of the vehicles be such that most of these minor service troubles will be eliminated. There will be more and more demand for the transportation engineer, and therefore there will be more and more demand for this interconnecting link that Mr. Ikert has mentioned who will bring home to the designers the necessity for meeting the sales situation, especially with the companies maintaining transportation engineers.

The matter of using different colors for the insulation of the wiring is of considerable interest to me. During the war the use of different colored insulation for the wires of the electrical equipment and different colored pipes for the gasoline and oil and other systems on airplane work was considered, and we tried to do some standardizing in this regard. The Navy had a system in use on the ships in the grand fleet in which certain colors meant certain things, but it appeared to be impossible for them to consider anything that was not patterned after the plan that was used in connection with the grand fleet. There were some other interests that believed other considerations were equally important. I believe the British Navy had a different system from the one that was in use in the American Navy. We were just commencing to get the matter pretty well straightened out when the armistice came. I think nothing further has been done on the subject since that time.

A. D. T. LIBBY:—I attended a recent meeting of the executive committee of the Automotive Electrical Association. One of the points brought up for discussion was, what we are to put up to our engineers to work out in the way of standards. The association, which comprises substantially all of the starting, lighting and ignition-battery manufacturers, is trying to promote the use of the S.A.E. Standards to the greatest possible extent. I have heard discussion as to the value of S.A.E. Standards, and their value was set forth in the December 1921 issue of THE JOURNAL; but everybody questions how much these electrical standards are used. The Automotive Electrical Association men who sell the equipment propose to get standards that they can sell. If they cannot sell standards, they want to find out why and then they want to get the present standards changed so that they can sell them. Many members of the committee of the Automotive Electrical Association are also members of the Electrical Equipment Division of the Society's Standards Committee.

As to referring the matter of wiring to the Electrical Equipment Division, I am very much interested because of having been its chairman in 1921. It was mentioned that resistance would be offered by the purchasing departments. Probably some resistance will be encountered also from the factory, because the production departments no doubt will object to handling several different colors of wire. If they handle only one reel, they can pull that wire out, chop it off, make up a cable and install it in the car; but if they must have five reels and pull off five different colors, they are likely to object. However that may be, more attention must be paid to what is put into a car from the service standpoint, and to what takes place after the car gets out into the hands of the user.

This Automotive Electrical Association has a Service-Men's Committee. This committee is wrestling with the very problems that we are discussing. What is service? How are we to take care of it? How is service to be paid for? Will service be absorbed by the service-men or will it be turned back to the factory? All those questions are vital and difficult to answer. Many of the questions that arise in connection with service must be answered in each individual case. One cannot say where the charge will fall. In addition to the committee I have just mentioned, an association known as the National Electric Service-Men's Association has been formed. It is holding a meeting here in Chicago at present to discuss the same problems we have before us here today.

Service, as outlined in Mr. Ikert's paper, should be the subject of future papers during the coming year. Nothing is more important than the service given car equipment. The design of the engine, the carbureter, the magneto and the ignition unit are secondary to the manner of their installation in the car. The illustrations in

Mr. Ikert's paper are very apt and show the difficulties that are brought about by improper installation. After the engine and the other units are designed, when they are put into the chassis they may not fit and may be inaccessible. In one car in particular the magneto was connected with the generator so that the breaker-box of the magneto was within ½ in. of the rear leg of the engine and it was impossible to get at the breaker-box without dismounting the entire generator and magneto. Just such points as this can be overcome by proper cooperation between the designing engineer and the man on the outside. I have in mind another car that, to my mind, is ahead of any other car in regard to the accessibility of its units. The engineers of that car are to be congratulated on the thought that they have given to placing the car units so that they can be gotten at by the service-men when the car gets out into operation and is sent to them for repairs.

PRESIDENT BACHMAN:—We are certainly very much gratified to hear from Mr. Libby that the organization he mentions will promote the work of the Standards Committee. It is this kind of cooperation that we are working and hoping for from all the commercial organizations and, on our part, we want to deserve it.

C. D. HOLMES:—This problem of inaccessibility is one that I have been "getting at" for years, particularly in regard to the proper construction of engines for truck service; for in truck service delay for upkeep and repair runs into money faster than anywhere else. A truck should be kept moving constantly, as every hour's delay means a heavy loss. It may be of interest to know that the word "Get-at-able" was trade-marked more than 10 years ago because, at the time, I believed that sooner or later the public would tire of paying out so much of its good money for the taking down and setting up again of an engine to make a simple repair. Further, the taking down and setting up is often done carelessly, particularly as to replacing gaskets, with the result that new troubles are started altogether too soon. An engine should never be taken down or taken apart in any way if this can possibly be avoided, for it may be weeks before the engine will run properly again. All this is because the construction or housing of the engine is such that, in case of trouble on the road, the part to be repaired or adjusted is usually so inaccessible that a wrecker has to be called, the truck towed in, and the engine taken apart before the truck can be put on the road again. Had the construction been different, the repair or adjustment could have been made on the spot, and the truck could have proceeded under its own power with but short delay. Measured in money, this means about $50 in the one case as compared with $1 in the other. If this appears to be an exaggeration, figure a supposable case in which a heavily loaded truck that is only 5 miles out on the road has engine trouble and must be towed in by a wrecker. The repair cost is but one item, for the entire cost includes that of towing, loss of time on the road and the time of at least two men who wait around.

If any such saving as this can be made in upkeep and repairs through accessibility that permits getting directly at every part, why has this not been done before? This is a natural question, although one that I shall not undertake to answer here, but the facts are evident and the remedy can be applied. Instead, we have undertaken about everything else connected with the engine and have experimented with the carbureter, the manifold, the ignition and the like. Many papers have been written on these various subjects, but the sum total of all such improvements works out to a saving that is small compared to what might be effected if the engine were only made accessible and simpler. This might be done and is certain to be done soon, if public demands are heeded.

The construction that I have in mind would cost but little, if any, more than the type now in general use. The efficiency of the engine would in no way be impaired; rather, it would be improved. The weight would be less. In looks, it would be simplicity itself. On one side there would be no attachments whatever; on the other side which, for safety in working, would be the side away from the road, there would be only the necessary carbureter, generator, starter, magneto and pump, all built into and forming a part of the housing. Yet these parts would all be removable separately, all of the oiling parts also being reached easily from the side and not from underneath.

In an address[1] at the 1921 banquet of the Society in New York, we were told by R. E. M. Cowie, vice-president of the American Railway Express Co., that "the motor vehicle is a wonderful machine, but it is too complicated," and that "the cost of upkeep of the average motor-vehicle makes the average man stagger." His advice was to simplify everything. He said, "The simpler the machines are, the more efficient they are, the less liable to trouble, and the more economical in operation and upkeep." These statements go to prove my point, and to make clear that this is the cloud on the horizon that threatens the very life of the industry. Yet we have gone on as before, presumably because no one dares to pioneer a new style of engine construction that is solely for the benefit of the user. The engineers of one of the large commercial organizations had it figured out that, with engines all of the type referred to, there would result a very great saving in upkeep, because delay in this case ran into money very fast and the aim was to keep moving at any cost.

In the United States Coast Guard Service, where life is at stake, the only engine they would consider was built so get-at-able that even the crankshaft could be taken out through the side, and any piston or crank-box could be changed in a short time. In fact, it is a matter of record that rescue work was performed while a piston was out of the engine. Had it been impossible to remove this piston when the trouble occurred, the boat would have been helpless and life would have been sacrificed. Everything possible is done to make the engine perform under the most adverse conditions. With spare parts on hand, they can meet almost any emergency, owing to the exceptional accessibility of the engine. This only goes to show that, where necessity is the master, we can accomplish things that under other circumstances might seem impossible. The Coast Guard Service is certainly the hardest of any on the powerplant. We can take a lesson from what it has done and build our truck powerplants so that the user will not be obliged to pay out money unnecessarily for upkeep and repairs, owing to inaccessibility.

How often it is the case that the really big thing that we are looking for is right before our eyes, but we go traveling around and around unable to see it. I am convinced now, more than ever, that I can show greater economy of operation, with the average driver, with an engine that is get-at-able in the sense to which I refer, with no other particular qualification except oversized bearings, than can be shown with any inaccessible engine that has all the very latest improvements. We fail to realize how a driver can upset our theoretical economies

[1] See THE JOURNAL, March, 1921, p. 249.

by careless and poor regulation but, given spare parts and an easy way to use them, it would be a strange man indeed who would not help himself. At least, a man can be forced to do so, or to show good and sufficient reason why he does not. This can be controlled largely if the engine is sufficiently accessible, but no one has yet been able to devise ways for making a careless driver keep his regulator set to the best advantage. So long as regulations are necessary, this will always be true. Hence, I say that simplicity and "get-at-ability" mean greater economy of operation than anything else.

HOWARD CAMPBELL:—One of the peculiarities of the American people is their knack of isolating some one word of our fast-developing "American" language, and using it for every legitimate and illegitimate purpose until it is worn to a frazzle. The indiscriminate use of the words "efficiency," "camouflage" and "sell" shows what may happen to an otherwise perfectly good word when the public fastens upon it in this way. In connection with the servicing of automotive units the word "service," which has always covered a multitude of sins, is becoming more and more indefinite and, unless something is done to rescue it before it is too late, it will have lost its original meaning and some other word will need to be devised or provided to fill the place for which this word was intended.

The purchaser of an automotive unit buys his car, truck or tractor, as the case may be, with a vague understanding that his purchase entitles him to something that is known as "service." In an effort to make this service effective, the various companies have organized service departments. Most of them seem to look upon the service departments as necessary evils, and that is as far as their interest goes. It is true that some manufacturers have their own distributing stations, and customers are supposed to come there for the necessary repairs. Such manufacturers usually have at least one man in the repair-shop who has been through the factory service-school and is assumed to be capable of doing any job required in what is supposed to be a standardized manner. But, even then, the likelihood is that this man must do the job with tools that he or the service-manager has devised, with the result that the job may or may not be done correctly and in a reasonable length of time, so that it is approximately in the condition in which it was turned out of the factory. I understand that the manufacturer of what is probably the most popular car in the world requires only that the dealer or service-station manager purchase car parts from the factory, and keep his prices within a certain limit, to obtain the right to use a sign that reads "Authorized Service-Station." So far as I have been able to learn, all that the manufacturer authorizes is the use of the car parts. The service-station manager can use any kind of machinery that he likes or any kind of tools that he has or may devise, and can turn out the work in whatever length of time he requires. Of course, the price factor limits the time to a certain extent; but it has nothing to do with the quality and, so far as I can learn, this is the only automobile firm in existence that attempts to regulate the prices charged.

It should be obvious, even to the most casual observer, that a job done with poor tools will require a much longer time than the same job done with tools designed for the express purpose. Further, good work cannot be done with poor tools. So, under the present system, where each service-station manager or garage foreman buys only such tools as he must have and cannot make, and manufactures all of the others out of such material as he has at hand, it is obvious that "service," as exemplified by a dozen service-stations, is very likely to mean a dozen different things.

I think there is no question that the largest automotive service-stations in the world are on Long Island, New York. Long Island City is just across the East River, within a few minutes' travel from "Automobile Row" on upper Broadway, New York City, and it is easily accessible, yet out of the crowded section. The service plants there include those of the Packard Motor Car Co., Pierce-Arrow Co., the White Motor Co., the Rolls-Royce Co. and several others, including the Hellman Motor Co., which is an "authorized" station. The New York Telephone Co. has its own service plant there and it is by far the most complete. This company owns more than 1000 motor cars and trucks, including nearly 700 Fords.

In the summer of 1921 I spent some time in each of these service plants, gathering material for a series of articles that have since been published in the periodical with which I am connected. I had an opportunity to observe the manner in which different organizations went about the matter of repairing motor cars and trucks. I discovered, among other things, that not only did these different service-managers employ different tools and methods for performing approximately the same operation, but that different kinds of tools and methods were employed for performing the same repair operation on the same kind of car or truck. Five different service plants had five different methods of align-reaming the main bearings. They all had different ideas and tools for reaming and facing connecting-rods, and one plant had two different kinds of fixtures for performing this operation. Practically all these pieces of equipment were designed by the shop foreman or superintendent, and were made in their own shops. One plant uses an electric hoist on an overhead trolley to lift an engine from the chassis, another uses a chain hoist and a third uses a floor crane operated by a crank. One plant uses a certain kind of reamer for reboring the cylinder blocks, another uses a different kind, a third uses a patent cylinder-reboring tool and yet another grinds the bores. The chances are that four different car-owners would be charged four widely different prices for having approximately the same kind of job done.

The New York Telephone Co. does not hesitate to make any equipment necessary to keep its cars and trucks in operation. It services its own units. This is considerably different from servicing the machine of a customer who will pay whatever the bill amounts to with little, if any, protest. Consequently this company has tools for removing bearing-caps, pulling out camshafts, removing clutches from their shafts, removing drive-shaft sleeves from Ford engines, holding bearings while reaming, compressing the packing in a rear-bearing case, testing magnetos and doing all kinds of jobs that without special equipment would require much more time than the company thinks is necessary. It makes every piece of equipment that is required to expedite the repairs, whether it be a tool, jig, or templet, and the service-plant manager claims that they save enough in every instance to pay for the tool many times over. Those of our factory service-managers who are looking for ideas would undoubtedly find some good ones there.

Some years ago the designers of a certain well-known make of car included a steering-gear in which no provision was made for taking up the wear. In due time numbers of these cars began to appear in the service-station to have the steering-gear repaired. Some one there devised a method of repairing this part so that it was practically as good as new and so that any future

wear could be taken up without much trouble. But I doubt if any of the others of this company's service plants know how to do this, or whether the home plant ever heard of the idea. Suppose that the owner of one of these cars were to take it into this service-station to have the play in the steering-gear taken up. He would have his car back in a reasonably short time, pay a reasonable price for the job, and probably would feel perfectly satisfied. Suppose another man who owned the same kind of car took his car into another service plant for the same purpose. The chances are that they would either make him buy a new steering-gear, or give him a job that would wear loose again in a short time. If it should happen so that those two men met and compared their experience, one can imagine that the latter man would be very angry.

The foreman in the machine shop of the Long Island service plant of the New York distributors of another well-known make of car has devised a method of testing valve-springs so that he knows that every spring used is of the proper tension. The fact that he finds some weak springs shows that such an operation is necessary. But I doubt that all the others of this company's distributors have similar testing devices. The same foreman invented a gage for testing the grinding of valve-heads. The gage is bell-shaped, having a hole down through the handle for the valve-stem, and the inside of the bell is ground to the exact angle that the valve-seat was intended to have. The gage is made of tool steel, hardened and ground. The grinding operator grinds the angle of a valve until he has a seat, then he puts a bit of Prussian blue in the bell of the gage and rubs it around. Then he inserts the valve and turns it around in the gage while the valve-seat rubs against the Prussian blue. An examination of the seat will show whether the angle is perfect and whether it is seating all the way around or not. If the seat is not perfect, he changes the set-up of the machine until the desired results are obtained. Such accuracy as is obtained by this means would not be possible by the use of the ordinary bevel protractor such as the average machinist uses, and certainly the job would take much more time if the operator had to try the valves in a cylinder block. Why does this company not furnish each one of its service plants with one of these gages?

Contrast the methods in use in these service plants, which are supposed to be the best in the world, with the methods in use in the factories in which the cars were made. Practically all of the nine service-plants that I visited were operated by manufacturers who pride themselves that the methods in use in their factories are the best that can be obtained or devised. Their manufacturing methods, their tools, their gages and their inspection methods are standardized. Everything in their shops that can be standardized is standardized, so that all parts will be alike and every car turned out just exactly like every other one. But when the man who has bought one of these cars needs some repair work done, he finds that one shop gives him one kind of a job, that he gets the car back in 3 days, and that it costs him, say, $45. He finds also that another shop gives him another kind of a job, that he gets the car back in 6 hr., and that they charge him $12. The worst of it is that the shop where he got the poorest job and paid the most money may be the distributor's own service-station.

I was told recently of a new manager who was sent to a certain Western city to take charge of a sales and service organization for the makers of a very prominent high-grade car. After he had been there for some time he discovered that the amount of service work done by his establishment was comparatively small and, upon looking up the records of owners in his district, found that very few of them ever came near his place. He walked down the street until he located a car of the brand that he was dispensing, broached the subject of service to the owner and asked him where he had his repair work done. The car-owner replied to the effect that the local service-station did not have the proper equipment to do a decent job, that it took them about three times too long to do a job and that when the job was done, it was not satisfactory. This manager returned to his own service-station and looked it over. He found that the machine-shop equipment consisted of two vises, one of which was broken, a post drilling-machine and an arbor press. It is strange that the company was even able to sell any cars in that town, because it is a fact that the reputation a distributor has for giving service very often makes or loses sales for him, especially if he is handling a car of the better grade.

The time will come when a sign reading "Authorized Service-Station" will mean that the operators of the shop are using equipment that has been approved by the builder of the car whose name appears on the sign, and that the work will be done according to approved and standardized methods as far as possible. It is even possible that the price charged will be standard except for such variations as are necessitated by the difference in overhead expense and wage scales in the different localities. Then such a sign will mean that the factory is standing back of its service plants with a guarantee as to both quality and price. I believe that under those conditions the automobile owner would not turn his car over to an irresponsible mechanic who is operating a one-horse garage any sooner than one would take a typewriter or a dictaphone to a machine-shop to have it repaired. When that time arrives, the word "service" will have a definite meaning to the manufacturer and to the purchaser of the unit.

B. M. IKERT:—I have noticed in several instances that changes in cars that would facilitate certain operations could be made at a slight cost. In one case the flywheel is put on with cap-screws; the main-bearing cap cannot be dropped because it hits the cap-screws. It can be done by sliding the bolts out of the bearing but, to get those out one must reach up behind a plate with a little wire-hook and one cannot see to do this. That could be overcome, as one mechanic told me, by fitting flat-head bolts and providing some sort of key to prevent the bolt from turning on the other end. If that were flush with the flange, the whole bearing-plate would drop right down and all the trouble would be ended. To do that might require some retooling of the factory, but it would help very much. The same sort of thing is true of many other installations. These are only small items, as a rule, but they are just nasty enough to tie up the whole job.

Another instance is that of a fan-belt on a car. I can see no reason why the manufacturer could not have set the radiator about 1/4 in. farther forward; as it is, the distance between the radiator and the fan-pulley is just a trifle less than the width of the belt, which is a pretty heavy belt. To take the radiator off is such a difficult job that, rather than do so, the mechanics take a bar and pry the radiator far enough away so that they can jam the fan-belt between the pulley and the radiator. A thing of that kind is discouraging to the service-man. One man said that the parts are all designed well and are pretty clean-cut but that he ran into difficulties when one assembly is made out of them. One particular engine that I looked at a year ago was an excellent one but, when

it was installed in a chassis, it would puzzle a service-man to get at the valves and other similar parts because the steering-gear and the carbureter interfered, wires prevented taking the plates off, and the nuts could not be reached with a wrench.

One large accessory company that makes speedometers has records of the length of time required to take a speedometer out of a car, work on it and put it back. In some cases it runs as high as 1 hr. to get at the job after the shaft has been taken out. In such a method as Mr. Whyte mentioned the assembly of the speedometer is right in front of the driver; by the removal of two screws a plate is released and the speedometer can be pulled out. A service-man could do that job in about 5 min. All of the other units are grouped about this one and all the connections are made at one point on the steering-column; so, the chassis wiring can be hooked to one side to take the body off. Car painters have told me that frequently the greatest charge they have to make in refinishing a car is for rewiring it. I recall one instance of inaccessibility in which the body had to be pried up to take a storage-battery out of a car, because the body overlapped the battery so far that the battery could not be removed except by doing all this really unnecessary work.

COMMERCIAL-BODY SUPPLY AND SERVICE

BY C. M. MANLY AND C. B. VEAL

SPECIFYING the four general plans that have been followed by chassis builders in securing body equipment as being the building of bodies in their own shops; on contract by the body maker to plans and specifications of the chassis builder; by a local body maker to the order of the dealer or the owner; and the assembling from stock of standard sectional units recommended by the dealer or selected by the owner, the authors discuss each of these plans in detail.

With regard to the plan of using standardized sectional bodies, the different sizes of chassis used for commercial purposes are separated into four specified groups and the production of a complete standard line including a number of styles of body for each chassis is commented upon and illustrated, inclusive of detailed considerations of the all-metal body. The advantages to the dealer and to the user of the factors that are advocated in body building are enumerated, and the standardization of commercial cars and trucks is considered briefly.—[Printed in the March, 1922, issue of THE JOURNAL.]

THE DISCUSSION

PRESIDENT B. B. BACHMAN:—There are two fields in the study of what the characteristics of bodies should be; one is the engineering and the other is the salesmanship field, but, it seems to me that, as with practically everything else, the engineering field comes first. In the commercial-car field there is no doubt that the influence of the horse-drawn vehicle with regard to body types has exerted practically a predominating influence; the study of bodies, as such, to harmonize with motor-truck chassis has been more or less confined to the so-called mechanical body for handling specific types of material, in the form of a dump body for handling solids or of a tank with a pump for the handling of liquids. The other bodies are inherently horse-drawn types and they have not advanced very far from their prototypes. This may be justifiable and no advance is possible but, because the unit that is hauling the load is so entirely different in its characteristics, it would be very strange if the old type of body that was most effective should be the most satisfactory for mechanical propulsion.

The outline of unit construction that Messrs. Manly and Veal have made is most interesting. Undoubtedly there are certain difficulties that will suggest themselves. Probably the principal one is the differences in type of various chassis. Many of those can be overcome largely by adherence to the work that has already been done by the Standards Committee and what may be done further, but there are some limitations, of course. I can see readily that the lighter unit, on chassis that have been standardized in other ways and are more or less similar in general type, probably offers a larger field due to both the similarity and the larger volume than those units in the strictly heavy class, and I can understand also why the question of volume, even if everything else is taken into consideration, probably drops down into relatively small proportions to get the advantages of a real production program.

P. H. WILLIS:—As to the advisability of standardizing on heavier types of body, the company I represent has had considerable experience and more or less remarkable success in such standardization on a few strictly standard types, and most particularly in standardizing several types of platform for chassis and trucks. The engineering on the bodies can be handled more easily because we can eliminate excessive widths and build the sills on the proper widths with the strength more evenly distributed. This works out very well. We have branches in different cities to which we ship knocked-down machine-parts. We get a lower classification rate in this way and can ship the body parts as cheaply as we can ship the raw lumber, thus saving excess freight.

J. E. SCHIPPER:—Some time ago I investigated this matter of standardizing certain features of the chassis to eliminate the tying up of a considerable amount of money in partially fabricated bodies. Such a thing as the distance between the side-rails of the chassis on vehicles of definite capacities varies to a considerable extent, although there is not very much reason for it; in fact, I believe that it could be standardized readily and that the Society has the matter under consideration. It seems to me that one of the reasons for high prices is the tremendous stock of material on hand in a raw condition and also in a partially fabricated state awaiting orders from body users for some definite chassis. For example, a dump body, which might be applicable to a 2 or to a $3\frac{1}{2}$-ton chassis, can be made up just as far and no farther until the body-builder knows exactly what make of chassis it is to be installed upon. A little work in the direction of the standardization of parts of the chassis that have to do simply with the attachment of the body might result in great economy for the truck-body industry. When I looked into this subject a considerable time ago, there seemed to be a great amount of interest in it on the part of the body-builders.

C. M. MANLY:—Our reason for preparing this paper is that we feel that anything that can be done to cheapen transportation is certain to have a great effect on the increase in the demand for such transportation. Therefore, original or first cost and also the cost of maintenance, which involves a continued use of the truck and the avoidance of tieups as well as the saving in actual

cost of repairing, are very large items in this matter of commercial bodies. The portion of total cost of the complete vehicle that the body represents is a considerable item. Everything that can be done to decrease the cost arising in connection with it is certain to have an important effect on expanding the field of automotive transportation.

We are very much interested in getting the largest amount of available information, especially from the users, in connection with experience on bodies that have been made, and what their defects are. This applies to bodies that have been either custom-built or built in any other way. It is of interest to ascertain the viewpoint of transportation engineers in regard to what they feel is possible in the way of standardization of types; also to get all the data possible as to the things that limit standardization in connection with the chassis. The matter of having every width of chassis frame from 32 to 40 in. for the same sizes of truck is exactly in line, apparently, with many of the other variations that formerly existed in connection with what the position of the bolt-holes and engine legs and other engine-installation dimensions should be. Standardization must play an even more important part in all automotive work if we are to reap the benefit of the large foreign trade that we must have if we are to utilize the automotive producing facilities of this Country to their fullest extent. I believe there is nothing that will count more in obtaining that foreign trade than extension of our standardization work as affecting the question of service.

PRESIDENT BACHMAN:—I think that neither the body-builder nor the chassis designer can outline the limitation individually; it must be a combination of the two. Some things made certain characteristics of chassis design necessary; other things are not necessary. At times we try to reduce the chassis widths so as to get as short a turning-radius as possible and the steering-gear will be out beyond the crankcase; many things must be considered.

F. A. ANGLER:—In operating a fleet of 25 to 30 trucks I find that we can reduce our costs on bodies by standardization. After having tried to build bodies in our own shops, we find we cannot do this, even to our own specification, at anywhere near as small a cost as that of an assembled body.

E. W. TURLEY:—We have occasion during the year to buy a considerable number of bodies in different parts of the United States. If a body is built for a certain model of 2-ton truck, I doubt that this body will fit on any other 2-ton-truck chassis. If we could buy 40 bodies for 2-ton trucks, all of the same size, we probably could get them built more cheaply than they are being built at present. The bodies we buy cost from $350 to $400. Each time we buy a different make of truck, we must have a body made for that particular truck. What we have done is to use a tracing-cloth drawing. When the repair bills on bodies show that some certain feature is breaking down, we change the tracing so that the next blueprints that are sent out will be an improvement upon the old body. If the distance from the end of the frame to the driver's seat were measured on each make of 2-ton truck in use in the United States, I doubt whether any two would compare closely enough to prevent the necessity of having a body of different size built for each one of those chassis. If the chassis were standardized, even to this degree, for say 1½, 2 or 2½-ton truck sizes, we could buy all of the bodies we need for the entire year at one time and save considerable money. It would be desirable to ship these bodies knocked-down. We often have to buy bodies locally, on account of freight rates, and take the chance of getting a local body-builder to build them according to blueprints. He may not follow our blueprints, with the result that we do not get as good a body as we would like to have.

MR. WILLIS:—I think it would be asking too much of the automotive industry to require it to standardize on a certain length of chassis for a certain width of truck. For instance, a 2-ton truck hauling barrels has about a 40-ft. wheelbase. We could hardly standardize on certain sizes of body. A 14-ft. body might be required on a ½-ton truck to haul paper boxes. But we have never had so much trouble in mounting bodies on various widths and lengths of frame as we have had with cabs. The latter demand more standardization. There is a great difference even in the different models of the same make of truck in the dimension on which we must determine the cab body. The dashes on trucks vary in width from 28 to 72 in. No standard cab-body would accommodate itself to that variation in dashes. Some truck builders wish to save all the frame length they can for the body; they may put the driver right up on the dash and give him 27 in. to fold-up in. Another company may go to the other extreme, so that the driver must almost lie down in his seat to reach the clutch-pedal. We have those conditions to meet in fitting cab bodies.

Another condition is that some springs are low and some high. To put pneumatic tires on a truck that has a low frame, we must build-up the sills under the body and the body is not stable on the low frame. Sometimes we get a high frame and a low body is wanted on it; we cannot make the sills thin enough to accomplish that.

We have more trouble with the cab body than anything else. If there is to be any standardization, I think it ought to be on the front end rather than on the rear.

GEORGE S. CAWTHORNE:—The Society has sent out a questionnaire asking for data, with a view to standardizing truck cabs. The dimension from the dashboard to the center of the seat box is one of the main dimensions in the standardizing of cabs. I think it can be standardized because a driver requires just as much room in one truck as in another.

C. B. VEAL:—I think that the present recommended practice of the Society will go a long way toward clearing up the situation so far as mounting is concerned, for the larger sizes of truck. I do not know whether the body-builders generally are aware that there is any such recommended practice. The next move, as I see it, is on the part of the truck companies. The truck builder should be made to realize that standardization is to his advantage. There is no need of making trucks unless they can be sold; selling them is the trouble today. One cannot sell trucks or bodies, but only transportation and service.

What Mr. Willis said about the possibilities of building standardized heavy bodies is true, but the man who attempts it is hampered in many ways, one of which is that he cannot turn out just one thing continuously. Ford bodies can be made day after day and they will all fit Ford chassis because they are all alike, but that cannot be done for the large truck. Parts of them can be turned out and various combinations of parts can be distributed so that, by drilling enough holes, making enough changes and making them sufficiently special, a branch station can accumulate enough different things finally to bolt them together and get some different combinations of body. But the body-builder is handicapped; he cannot give the best service under such conditions.

More Car-Miles per Gallon of Fuel

By O. C. Berry[1]

Detroit Section Paper *Illustrated with* Charts

Economy tests carried out in France indicate that it is possible to obtain a larger number of miles per gallon from cars made there than from cars made in this country. The author states that it would be well to make a careful study of the factors influencing car economy and to assure that our future car models take full advantage of all possible means of increasing their economy.

Figures are presented showing the extent to which economy can be increased by changing such factors as the carbureter adjustments, time of the spark, rear-axle ratio and speed of driving. A car that normally will go 21 miles per gal. under favorable test conditions at 20 m.p.h. was increased to 43 miles per gal. at 20 m.p.h. The study is not completed, but it has gone far enough to demonstrate its value. This progress report is presented to stimulate thought.

The American automobile has been developed to a stage where it is a remarkably reliable vehicle. The hundreds of different makes of car do not vary in economy much more than their differences in weight would account for, and the offhand conclusion logically would be that there is probably not much more room for improvement in economy than in mechanical reliability. In this connection the recent economy tests in France are of interest. A Voisin limousine weighing 5300 lb. ran 28.3 miles per American gal. A Citroen car weighing 2500 lb. traveled 53 miles per gal., and a Petit-Peugeot car, weighing 1200 lb., 76.9 miles per gal. Contrasting these figures with the performance of the average American car makes it evident that the margin for improvement is indeed great. Probably no one feature of American cars has received less careful attention from automotive engineers than the capacity for economical running. No fuel-economy features are incorporated in the European cars mentioned that we do not know about and understand, and I do not concede for a moment that the foreign engineer is one whit more resourceful or well informed than the American. It is therefore incumbent upon us to look the situation squarely in the face and, recognizing the importance of fuel economy, to study carefully all of the factors influencing the number of car-miles per gallon of fuel, and to make certain that every possible improvement is included in the design of our future models.

Carburetion

One of the outstanding reasons for the poor mileage of the average American car is poor carburetion or, more accurately, poor carbureter adjustment. To make this clear, it will be necessary to show the effect of the richness of the fuel-mixture on the power and economy of an internal-combustion engine. In studying this point, use was made of a Willys-Knight engine mounted on an electric dynamometer. The results of these tests are shown by the curves in Fig. 1. In these curves the number of pounds of gasoline per pound of dry air in the fuel-mixture is plotted horizontally and brake horse-

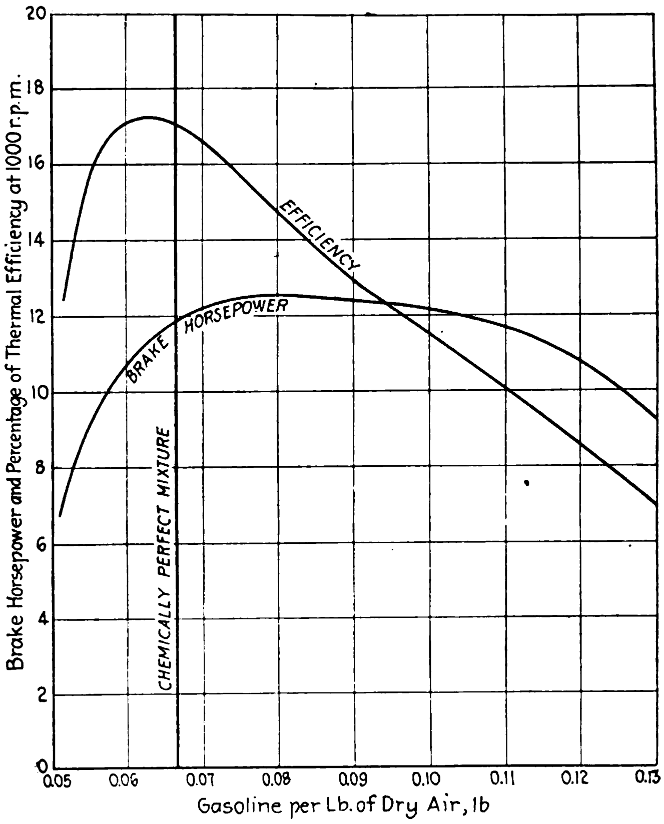


Fig. 1—Curves Showing the Results of Tests to Determine the Effect of the Richness of the Fuel-Mixture upon the Power and Economy of an Internal-Combustion Engine

power and percentage of thermal efficiency are plotted vertically. In carrying out the tests the throttle was arranged so that it could be securely fastened, and all of the tests were run at a constant engine speed. The gasoline was weighed to 0.01 oz., the air metered to 0.10 cu. ft. and the brake-load, speed and temperatures were carefully measured. Under these conditions and with the carbureter adjusted to give a rich and powerful mixture, the first test was run. The power, efficiency and mixture ratio were then computed, and a point established on both the power and the efficiency curves. The gasoline adjustment was then made slightly leaner, the brake-load adjusted to produce the correct speed, and a second test was run. This procedure was repeated until the mixture became too lean to allow proper engine performance. More gasoline was then introduced each time until the mixture was entirely too rich. The mixture was thus made alternately richer and leaner until a sufficient number of points had been located on each curve to indicate clearly the effect of the richness of the fuel-to-air mixture on both the power and the efficiency of the engine.

It will be noted that there is a wide range of mixtures through which the richness has little effect on the power of the engine. The lean and the rich mixtures give

[1] M.S.A.E.—Chief engineer, Wheeler-Schebler Carburetor Co., Indianapolis.

equally good power, and both result in what seems to be perfect engine performance. This accounts for the fact that the comparatively crude carbureters of earlier days gave mainly satisfactory performance.

The range of mixtures giving the highest efficiency is very narrow, and corresponds to about the leanest mixture with which the engine will run without missing. As more gasoline is added, the efficiency drops off very rapidly; the richest mixture producing full power will result in only about one-half of the maximum efficiency. This explains why it is that of two cars of the same make performing nicely, one may go nearly twice as many miles per gallon as the other.

Why Over-Rich Mixtures Are Used

There are several reasons why nearly all of the carbureters are adjusted so as to give too rich a mixture when the engine is hot. One is that gasoline flows much more slowly when cold than when warm, an opening large enough to deliver the required charge to a cold engine will inevitably deliver too much when the engine is warmed-up. Another is that cold air is more dense than warm air; the weight of air delivered through the air-opening in a carbureter is greater when the air is cold than when it is warm, thus causing the mixture to become richer as the temperature of the engine increases. Thus, the mixture furnished by a carbureter with a fixed adjustment will become richer as the temperature rises. To make matters worse, not nearly all of the gasoline in a cold mixture is vaporized so that it can be burned. This makes it necessary to supply a considerable excess of fuel in a cold mixture to get the engine to run at all.

The American public has been taught to demand a car that will start easily in any kind of weather and continue to run without requiring any attention or adjusting. They have been supplied with a large variety of non-adjustable, or what has aptly been termed "fool-proof," carbureters. These carbureters will deliver a mixture rich enough to start well in cold weather and, due to the great range of the explodable mixtures of gasoline and air, will seemingly keep the engine running perfectly with full power, in summer or in winter. The only fault to be found is that, when the engine is hot, the mixture is much richer than is necessary and the number of miles per gallon is correspondingly low. A non-adjustable carbureter cannot be made so that it will make starting easy and also give high efficiency in driving.

A carbureter adjusted for maximum economy on a hot engine will deliver too lean a mixture to start with when the engine is cold. There are two ways in common use for meeting this situation. One is supplying a choke to shut-off the main supply of air to the carbureter, thus enriching the mixture. The other is providing a means of adjusting the carbureter from the dash or the steering-post of the car.

The action of the choke is very severe. It causes raw gasoline to be sucked into the engine in large quantity. Its action becomes increasingly severe as the engine speeds up; if the mixture is rich enough to start the engine it will increase in excess fuel as higher speeds are reached. This makes it necessary to readjust the choke every time the speed is changed.

The dash adjustment can be made very satisfactory in action if it is used correctly, and convenient as well. Its use is easily understood, because the idea is to get it set as lean as possible; it is very easy to tell when it is too lean, for the engine will lose power and backfire through the carbureter. A proper dash adjustment set right for one speed and load will be correct at all speeds and loads at the same temperature, and will need to be changed only as the temperature changes. The mixture should be just right for a hot engine with the dash adjustment set at its leanest position.

Discussion of Curves

The curves in Fig. 1 show that a richer mixture is required for maximum power than for maximum efficiency. It is therefore obvious that a carbureter adjusted for maximum miles per gallon will deliver too lean a mixture to give the full power of the engine.

This difficulty can be obviated in the following manner. When driving on a level road at any speed the law allows, the engine is never called upon to deliver nearly its full power. Under these conditions the most important consideration is economy. When a steep hill is to be climbed, acceleration is desired or a neighbor wants to race, full power is the important thing. The constant-speed driving is done with the throttle pretty well closed, and the power work with the throttle wide-open. It is possible to design a carbureter that will furnish the engine with the most economical mixture at all ordinary speeds on a hard level road, and with the most powerful mixture when the throttle is wide-open.

The fuel is almost never entirely vaporized in the intake-manifold of the engine. A considerable portion flows along the manifold walls as a liquid, lagging behind the air that passed through the carbureter with it. When the engine is idling this layer of liquid is very thin, but under full load it usually forms a fairly large stream. When the engine is put under load quickly, the air rushes ahead and leaves at least a portion of liquid on the manifold walls. If the mixture is lean enough to be efficient under steady running conditions, this temporary impoverishment will often be sufficient to stall the engine. It is therefore necessary to incorporate in the design of the carbureter some special means of temporarily enriching the mixture during acceleration. There are many types of construction in use, some of which are very satisfactory. It is therefore possible to obtain perfect acceleration, even when using a mixture lean enough for maximum efficiency. The more perfect the manifold design is, the easier this becomes and, with the very best manifolds, very little special provision is necessary for acceleration.

The Ideal Carbureter

In my estimation the highest type of carbureter yet developed for American gasoline will

(1) Furnish the engine with the most efficient mixture when the car is driven at a constant speed on a hard level road
(2) Provide the engine with the most powerful mixture when the throttle is wide-open
(3) Make perfect acceleration possible, even when adjusted for maximum economy
(4) Have a dash adjustment that will make starting and warming-up an easy matter, even in the coldest weather

There are carbureters on the market that meet all of these requirements. They make a greatly increased number of miles per gallon possible, and are more satisfactory to handle than the cruder fool-proof varieties.

The Timing of the Spark

The second factor influencing the economy of the automobile is the timing of the spark. If the spark passes too early it will lessen the power of the engine and tend

to produce what is called a "spark knock." If it passes too late it will also reduce the power of the engine. The proper timing of the spark in any given engine will vary with two things, the load the engine is carrying and the speed at which it is running. Nearly everyone knows that the faster an engine runs the earlier the spark must pass to give best results. The load condition has received comparatively little attention, and I will therefore enlarge upon it.

In making a study of this point an engine was mounted on an electric dynamometer in such a way that the exact time of the passing of the spark could be read, as well as the speed of the engine and the brake-load carried. The engine was run through a series of different spark-settings at each of a series of different throttle-openings. The results of these tests are shown in Fig. 2. In the curves the lead of the spark in degrees on the flywheel is plotted horizontally, and the brake-load carried is plotted vertically. All of the tests were made at 1000 r.p.m. Each curve was made at a constant throttle-opening, the brake-load being changed along with the spark-timing to keep the speed constant.

It will be noted that a lead of 20 deg. is required for maximum power at full throttle, and that a lead of 25 deg. will result in a knock. As the throttle is closed gradually, the lead required for highest power is increased until, with the lightest load carried, a lead of 40 deg. is required. At this throttle-opening the engine will carry only 60 per cent as much load with a 20-deg. spark-lead as with a 40-deg. lead. It is so disagreeable to have an engine knocking during acceleration and hill-climbing that the spark is almost never advanced beyond the point where knocking occurs at full load. Since the engines in our American cars run at such a low percentage of their power capacity, the spark is timed much later than it should be for greatest economy, which results in a considerable loss in miles per gallon.

The vacuum in the intake-manifold of an engine varies in almost inverse ratio with the brake-load carried. It therefore follows that the lead required by the spark at a light load, beyond that required at full load, will vary

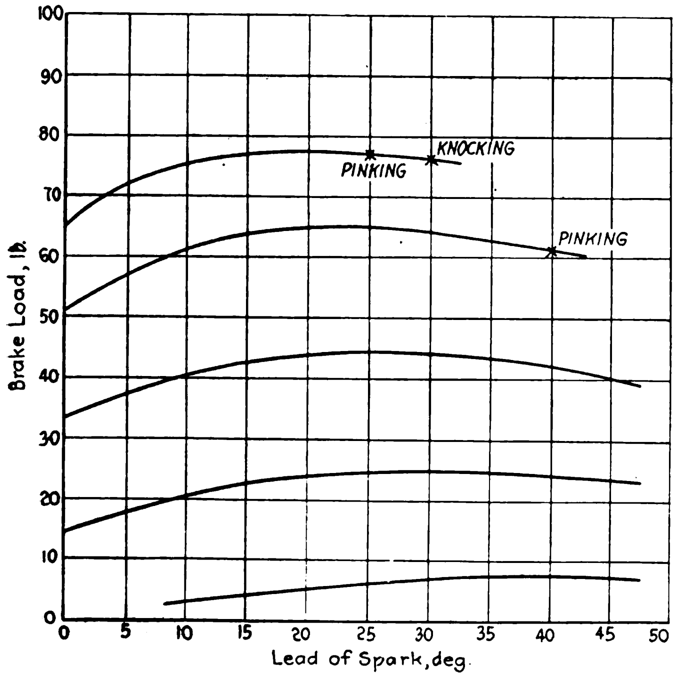


FIG. 2—RESULTS OF TESTS TO ESTABLISH THE RELATIONSHIP BETWEEN THE LEAD OF THE SPARK AND THE BRAKE-LOAD

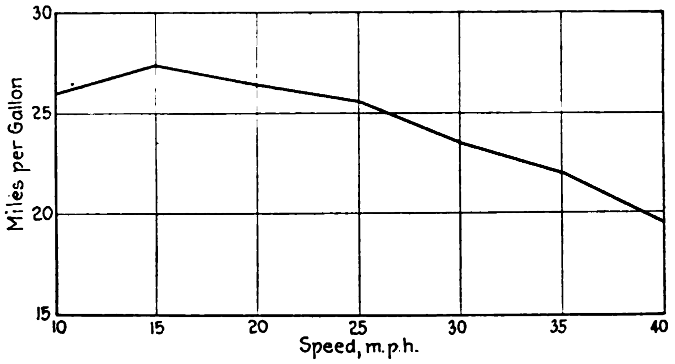


FIG. 3—CURVE SHOWING HOW CHANGES IN THE SPEED AT WHICH A CAR IS DRIVEN AFFECT THE MILEAGE OBTAINABLE FROM A GALLON OF GASOLINE

directly with the intake-manifold vacuum. This fact has been made use of in producing an automatic device for changing the time of the spark to correspond to the load carried. When properly installed, together with a standard controller compensating for speed, this device provides a completely automatic and very accurate means of timing the spark. This device is not on the market at present, but I have conducted a series of tests showing its merit and feel confident that it will be perfected and presented to the public in the near future. When properly installed, it should add considerably to the economy obtained by the average driver.

The speed at which a car is driven will cause a considerable variation in the economy obtained. This is shown in Fig. 3. The tests here reported were made on a hard level road, running in both directions over a carefully measured course. The car was tested with the top and windshield up and at a weight of 3100 lb., including the driver and the observer. Cars of this make probably will average about 21 miles per gal. under test when in the condition in which the average driver keeps his car. By freeing the brakes, having the spark accurately timed and the carbureter carefully adjusted, the record indicated by Fig. 3 was obtained.

THE LOAD ON THE ENGINE

Another opportunity to improve the economy of our automobiles has to do with the large reserve power in the engines. A good American car can climb a steep hill in high gear, and accelerate very rapidly. This delightful "activity" necessarily means that under ordinary driving conditions on a hard level road the engine is called upon to exert a very small percentage of its full torque capacity. To get accurate information on this point, a car was chosen of a make known to be carefully designed, well built and having a good average accelerating ability of about 3.2 ft. per sec. per sec. Tests were made showing the torque capacity of the engine and the power required to propel the car at varying speeds. The results of these tests can be taken as representing the average performance of good American cars.

The method of conducting the tests was to insert between the carbureter and the intake-manifold a plate having a hole drilled through it that would serve as a throttle-opening. A determination was then made of the maximum speed the car could attain on a long stretch of hard level road with this orifice in place. This test was repeated with a number of orifices of different sizes. With the engine removed from the car and connected to an electric dynamometer, tests were run to determine its torque capacity at varying speeds with each of the orifices in place. Knowing the size of the rear

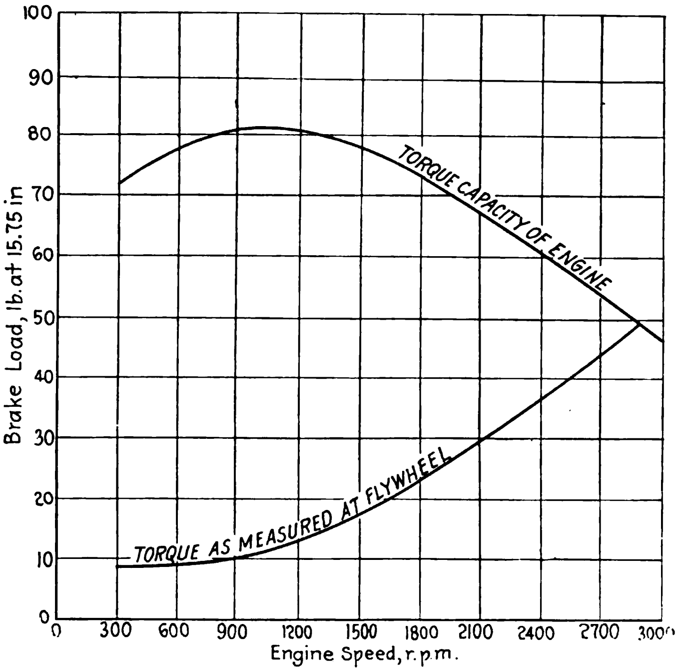


FIG. 4—RESULTS OF TESTS MADE TO OBTAIN THE TORQUE REQUIRED TO DRIVE THE CAR AT VARIOUS ENGINE SPEEDS AND THE TORQUE MEASURED AT THE FLYWHEEL OF THE ENGINE

wheels and the gear-ratio, the engine speed corresponding to any car speed is accurately known. The torque required to drive the car at any one of these maximum speeds for a given orifice is the torque capacity of the engine at that speed with that orifice, as shown by the dynamometer tests. The results of these tests are shown in Fig. 4. In these curves the engine speed is plotted horizontally and the brake-load at a radius of 15.75 in. is plotted vertically. The upper curve shows the torque capacity of the engine and the lower one shows the torque, measured at the flywheel of the engine, that is required to propel the car at corresponding speeds. The distance between the upper and lower curves is therefore proportional to the reserve power of the engine, and represents its ability to accelerate rapidly or climb a steep hill.

In this particular car an engine speed of 1000 r.p.m. corresponds to 20 m.p.h. It will be noted that at this speed only about 14.7 per cent of the torque capacity of the engine is used when driving on a good road. Anyone familiar with the performance characteristics of an engine knows that this fact alone will account for a large reduction in the thermal efficiency obtainable from the engine under ordinary driving conditions. This is such an important consideration that a special series of tests was carried out to show the exact situation.

EFFICIENCY AT DIFFERENT LOADS

It is well known that the brake thermal efficiency of an engine varies from zero at no load to a maximum that occurs at a little less than full load. Engines differ according to their design in ability to perform under different conditions, some doing comparatively better at light loads and others at full load. Tests were therefore carried out on the engine previously used to determine the effect on its thermal efficiency of changing the load.

The engine was installed on the dynamometer and run at 1000 r.p.m., at a series of different throttle-openings. At each throttle-opening a series of tests was run at different carbureter settings, similar to the tests plotted in Fig. 1, and showing the mixture corresponding to highest-efficiency at that throttle-opening, together with the corresponding power and efficiency. The brake-load was expressed in terms of brake mean effective pressure, and a curve drawn through the points showing the highest efficiency at each load. This curve, shown in Fig. 5, indicates the maximum efficiency obtainable with this engine at each brake-load, the engine running at 1000 r.p.m. The efficiencies will vary as the speed is changed, so that this curve should be interpreted as applying only to the one speed.

The carbureter was adjusted leanly enough to give maximum economy at all loads until the throttle was wide-open. After this point was reached, the increases in power had to be obtained by enriching the mixture, until the full power was obtained. These richer mixtures result in reduced economy, thus causing the curve in Fig. 5 to show a decided drop as full power is approached. This curve therefore offers one more illustration of the importance of using lean mixtures when high economy is desired.

Fig. 5 shows how important it is to run the engine under a comparatively large percentage of its full-load capacity. Since the miles per gallon of any car under given running conditions will vary directly with the thermal efficiency of the engine, a change in the load factor on the engine can result in greatly increased economy. This fact was checked in part by installing a different rear-axle ratio and noting the change in the car economy. The standard ratio was 4⅞ to 1, and resulted in a load factor of 14.7 per cent at a speed of 20 m.p.h. Under these conditions the car could go 28 miles per gal. when loaded so as to weigh 3100 lb. A gear ratio of 2½ to 1 would result in nearly twice the former load factor, and Fig. 5 shows that this should result in an increase in fuel economy of about 50 per cent, or to 42 miles per gal. Under test the car actually ran 43 miles per gal. These tests were all made with a

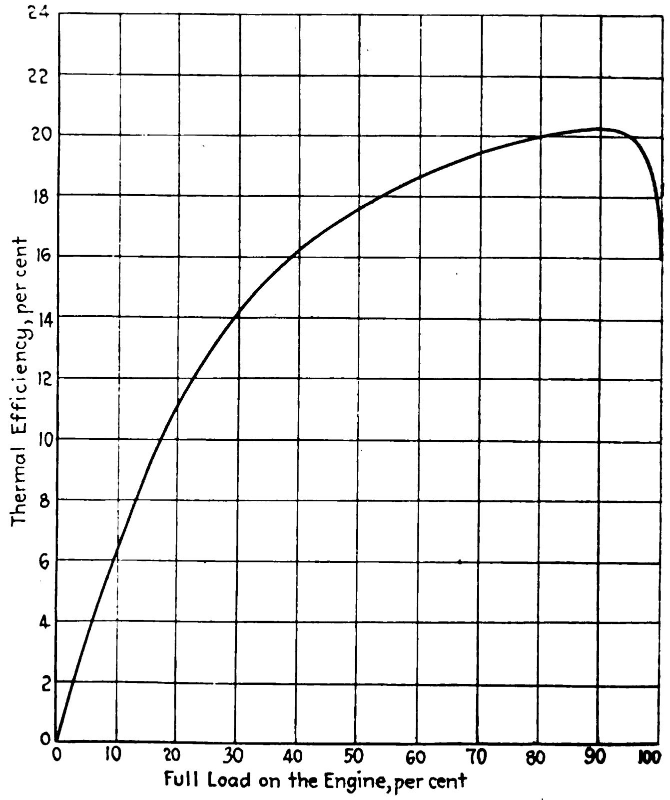


FIG. 5—VARIATION IN THE THERMAL EFFICIENCY OF THE ENGINE AT DIFFERENT LOADS

carbureter equipped with a dash adjustment to make starting easy in cold weather.

Efficiency and great ability to accelerate cannot be obtained at the same gear-ratio. The former requires that the engine run at a high percentage of its torque capacity and the latter the other extreme. We cannot afford to compromise the delightful activity of our cars. On the other hand, I feel confident that the first company to produce a car having the activity to which we are accustomed and capable also of going more miles per gallon than any other car of its weight in the Country, will become both rich and famous. We can accomplish this if we have four speeds forward. Third speed could be called the "power" or "speed" gear and the fourth the "economy" gear. Such an arrangement would be beneficial by

(1) Greatly increasing the miles per gallon
(2) Reducing the engine vibration at the common touring speeds
(3) Increasing the speed at which one could tour with comfort and satisfaction on good roads
(4) Enabling one to tour at high speed with the mental satisfaction that would result from knowing that one is not punishing the engine, while any other car at the same speed would be going entirely too fast for its own good
(5) Greatly reduce the wear on the engine, thus prolonging its life and reducing maintenance costs

The disadvantages are that (*a*) the four-speed transmission would be expensive when sufficiently well-made to run quietly on two gears and (*b*) the gears would need to be shifted more often.

The possible advantages are therefore fundamental and far-reaching and the disadvantages of a practical nature and, although difficult to overcome, not too difficult for automotive engineers when they really become interested in mastering them.

Education of the Buying Public

The automotive engineer gives more attention to designing a car that will sell well than to making it meet his ideas of perfection. Our fuel has been cheap and plentiful and economy has not been demanded by the average buyer. He has rather been taught to demand a car that will accelerate well and climb any hill on high gear. He seldom stops to think how much the activity of his car costs him in gasoline, oil and repairs. He has never been educated to see how much can be gained by accepting different standards of performance. The automotive industry should make it a point to get this information before the public and thus create a demand for a more economical car.

The driver is not aware of how much fuel his car is using as he rides along and is accordingly not apt to handle it economically. For this reason it will be well for us to give the flowmeter serious attention. Such an instrument has great possibilities as a means of educating the public to demand greater economy in their care and would logically result in cars being kept in better mechanical condition. If a flowmeter can be developed that will never cause trouble, it should be installed as standard equipment on all new cars. It has already been developed to a point where it can unquestionably be made of value to the service-man.

Economy Contests in America Desirable

It might be well to run a series of economy tests in this Country similar to those that have been carried out in France. Unfortunately, a number of American engineers have questioned the practical value of these tests. They state that the carbureters are adjusted so leanly in these tests that the cars are incapable of good general performance and would be unfit for general use without changing the carbureter adjustment. As a matter of fact, it is possible to build a carbureter that will perform well even when adjusted for maximum economy, and these tests would prove this. In planning such contests a careful set of standards of performance should be established. These standards should be high and every car required to meet them before taking part. Under these conditions the results would be of real practical value and lead to wonderful development in the way of a public demand for an economical car.

By applying the ideas brought out in this paper to a standard representative car, its mileage was increased from 21 to 43 miles per gal., under similar test conditions. Our knowledge of how to increase car economy is not yet complete. There are factors influencing miles per gallon that are not mentioned here. This paper is merely a progress report presented at an early stage. I feel however that it has gone far enough to demonstrate the value of the study and warrant a conviction that American engineers can be depended upon to produce cars showing economies that now seem almost impossible.

By lessening wastes in running and storing petroleum, by perfecting the processes of making gasoline and especially by developing ways of producing substitute fuels economically and in large quantities, the chemists of the Country can help make it possible for future generations to ride in automobiles. In the meantime, automotive engineers should be steadily increasing the economy of our cars and thus add to the value of each gallon of available fuel.

THE DISCUSSION

W. S. James:—Regarding the comparison Mr. Berry made between American and foreign-built cars, it should be borne in mind that the European tests he quoted were run under very special conditions. When driving around the curves, if a driver slowed-down very much his engine usually stopped. All the cars were adjusted for very economical running to obtain the maximum number of miles per gallon of fuel. The Petit Peugeot car weighed about 1000 lb. and gave a performance of 70 miles, or 35 ton-miles, per gal. Let us compare this with the Buick performance. The Buick weighed at least 3000 lb. and gave 20 miles per gal., or about 30 ton-miles per gal. There is little doubt that, under special conditions similar to those existing in the tests discussed, the Buick performance could be increased to 25 miles per gal., which would mean about 37½ ton-miles per gal. or a figure at least as good as that of the Petit Peugeot.

I had an opportunity in Europe to obtain manufacturers' statistics on the fuel mileage of European cars and compared them on the basis of ton-miles per gallon with similar statistics from American companies exhibiting cars at the New York Automobile Show. The figures can possibly be taken as equally optimistic in both cases. The results of the comparison showed that the number of ton-miles per gallon obtained from cars in France was no greater than the average number of ton-miles per gallon obtained in the United States. In England the number of ton-miles per gallon is possibly 10 per cent greater than in the United States. In both instances, however, the average number of miles per gallon is greater by possibly 20 per cent in England and from 10 to 15 per cent in France. This simply means that we are building heavier cars, with engines of at least the same efficiency.

Using the volume of air drawn into the engine per ton-mile as another basis of comparison, American engines draw in about 30 to 35 per cent more air per ton-mile. This means that our engines run at more nearly closed throttle, and, as Mr. Berry pointed out, our pumping losses are very much greater. In spite of the fact that our engines pump 30 per cent more air, American cars are still only 10 per cent low in ton-miles per gallon. This does not mean that we are any farther behind European performance in actual engine efficiency, but that we are using heavier cars. In fact, at the New York Automobile Show, I could find only about four cars made in the United States that weighed less than 2000 lb., loaded. In England there are about 65 makes of car that weigh over 2000 lb. and 67 that weigh under 2000 lb.

If the amount of gasoline burned in internal-combustion engines is 4,000,000,000 gal. per year, and it were to flow over Niagara Falls at the mean rate of flow of the Niagara River or 222,000 cu. ft. per sec.; it would take a little more than 40 min. to flow over the falls. The total volume of crude oil from which this gasoline was produced would require over 2½ hr. to flow over the falls. This helps us to understand what a gain it would be to save only 10 per cent of that volume of liquid. Even a 1-per cent saving in gasoline on the average amount used in this Country would mean a large volume of liquid.

I thought I knew how to drive a car economically, but since I have had a gallons-per-hour flowmeter on my car I have changed my mind. I think Mr. Berry's point about the use of flowmeters was stated very well. The greater part of the waste of gasoline is not through the ignorance or the lack of application of the knowledge of the engineers, but through a lack of knowledge and information on the part of drivers all over the Country. If a driver had some sort of tell-tale that would indicate whether he was using more gasoline at one time than at another it would help to save gasoline. The instrument could be graduated to indicate dollars and cents per hour, or miles per gallon or whatever seemed desirable.

O. C. BERRY:—I do not wish to convey any unfavorable impression regarding American engineers or car performance in this Country. Some tests, such as Mr. Nelson's at Indianapolis, show performances considerably better than those of the French. But the performance of our cars in the hands of the average driver can be improved. The problem we are facing is to persuade the average driver to increase the number of miles he obtains from a gallon of gasoline.

P. F. HOWELL: — The flowmeter would no doubt be better understood if it were called a gasoline speedometer. It is an instrument that will indicate the amount of gasoline that is being consumed per hour when used in connection with an automobile, and gives the miles per gallon delivered by the car when read jointly with the speedometer. The instrument was developed about a year ago and was found to be of value when used in the final adjustment of new cars before delivering them. The volume of the distributor's business comes from the small contracts for from 5 to 20 cars each. This business is often rather expensive in regard to service and replacements of parts; the only possible way the necessary evil of servicing can be overcome is to furnish the dealer with cars that are accurately adjusted, eliminating the necessity of service after sale and delivery are made.

With such an instrument not only can a standard of carbureter adjustment be set but it can be maintained in spite of the poor help that is often the only help available. Nearly everyone in the automobile business thinks that he can adjust a carbureter accurately. This is not a fact, from a standpoint of efficiency and economy. But even inexperienced help can adjust a carbureter accurately by using a flowmeter, as has been proved. It has proved that some old-time theories are wrong. For instance, the general opinion was that 20 to 25 m.p.h. was the best speed to drive any car economically, but this instrument proves that the speed may vary from 12 to 30 m.p.h., depending upon the make of the car and its condition.

The procedure to follow, when adjusting and servicing motor cars, is to first jack the rear wheels off the ground. Start the engine, shift into high gear, speed up to about 20 m.p.h. and then move or change the carbureter adjustment until the maximum speed is indicated on the speedometer for the least amount of gasoline that is passing through the flowmeter. Then enrich the mixture by carbureter adjustment until the speedometer shows a loss in speed. Then watching the flowmeter reading only, set the flow at one-half way between "too rich" and "too lean" by altering the adjustment. This will result in the maximum amount of economy and power, no guess-work having been employed to set the carbureter accurately. After the carbureter has been set and while the car is still up on jacks and in high gear, speed up the engine until the flowmeter indicates 1 gal. per hr. Then note the speedometer reading. Change the spark-timing until the greatest speed combined with a smooth-running engine is attained. This should be noted as being a certain number of miles per hour to a certain number of gallons per hour. If any other adjustments are necessary, they should be cared for at this time. The result will be noted in the gain or loss of speed.

To complete the final adjustment of the entire car, the work should be continued by checking up the alignment, removing the friction from the driving members and releasing the brakes so that they do not drag. The proper quantity and kind of grease should be provided for the universal-joints, the differential and the bearings and, upon completing and making the necessary adjustments, the car should again be shifted into high gear, the engine speeded up until the flowmeter registers 1 gal. per hr. and the speedometer reading noted to see if any improvement in efficiency has been achieved from the work that has been done. If all this work has been thoroughly completed, the engine and the driving members will have been adjusted to their highest degree of efficiency, and the result in miles per gallon can be considered the standard to which other cars of the same make could, and should, be adjusted when upon jacks. The car should then be removed from the jacks and driven on the road. The miles per gallon should be noted and the oil, bearings, tires and other sources of friction cared for, to eliminate as much friction as possible and the attempt should be made to obtain more speed for the same amount of gasoline. The result in the end will be the maximum number of miles per gallon to which any motor car of this make can be adjusted, and this will be the standard in miles per gallon at which all cars of this make should operate on the road. The difference between the results on the road and on jacks might be termed "road resistance." This loss in miles per gallon might be termed the "standard loss" to be expected in this particular make of automobile. We have found in this way that it costs no more to condition new automobiles up to a standard, and costs much less for service,

because of being able to locate the trouble in less time. Often one adjustment is enough to bring a car back to its standard number of miles per gallon.

MR. BERRY:—No matter what we think about the value of the flowmeter for the production car, it is unquestionably a fine instrument for the service manager. I have driven with a flowmeter for about a month. I find that it tells why we do not get better mileage in winter. It takes as much gasoline to start the car on cold mornings as to maintain it at boulevard speed after it is warmed up. I do not know whether it will be possible to increase very materially the number of miles per gallon in winter on cars driven in the city, because one cannot cut down the gasoline to a close adjustment in winter; however, most cars are used in summer and on long drives. For those conditions we ought to be able to get almost the theoretical maximum output. It is possible with the American carbureter to cut down so that the maximum mileage is obtained in touring with a warm engine, still having plenty of power when needed.

GEORGE BREEZE:—Were the performances in France rated in United States or British Imperial gallons?

MR. BERRY:—I was quoting figures that had been reduced to United States gallons. They used benzol, which gives something like a 14-per cent better mileage per gallon than gasoline.

MR. BREEZE:—We equipped a Ford car with a dash adjustment such as you mentioned. It takes as much gasoline to warm-up the engine and get the car out of the garage as it does to run 1 mile afterward. It requires 1¼ turns of opening of the fuel-adjusting valve for the first ½ mile and 1 turn for the next 1½ miles. Up to that point the average is about 16 miles per gal. Thereafter it is 25 miles per gal., with about a ⅞ turn. That is with standard equipment. Running with special equipment and a ⅞ to ¾ turn will give 31 to 32 miles per gal, and a ¾ to ⅝ turn gives 37 miles per gal. I have found that one can cut the warming-up time in half by heating the mixture. Heating the mixture has more to do with warming-up the engine quickly, than anything else on Ford cars.

MR. BERRY:—I know of one foreman at the Ford factory who drove from Detroit to Indianapolis and return with his family and averaged 33 miles per gal. during the trip.

A MEMBER:—Mr. Berry said that the pumping loss is one of the greatest losses at low speed. Why cannot the carbureter be developed to obviate that pumping loss by reducing the engine speed and leaning the mixture, rather than by increasing the manifold suction. We have made many tests in running engines on gas instead of gasoline. To get the best economy, instead of running up or throttling the engine down, we kept leaning the mixture. In running the engines idle on gas instead of gasoline, we found that we used only about one-third of the fuel that we used when running the engines on any carbureter I have ever tested.

MR. BERRY:—The explodable range of a mixture of gas and air seems to be considerably wider than with gasoline vapor and air. The ideal carbureter I have endeavored to describe would be arranged to give the richness of mixture corresponding to the highest efficiency during the touring range that is as far as one can go in that direction. If the amount of gasoline is decreased below that point, we not only do not increase the efficiency but we decrease it. It is impossible then to continue to lean the mixture and control the engine in that way more than the carbureters are now doing.

MR. BREEZE:—Have you ever tried stratification of mixtures?

MR. BERRY:—No.

MR. BREEZE:—I have seen some experiments in that regard in certain classes of engines.

MR. BERRY:—I know that some experiments have been made on a design that is intended to handle the mixture in that way. I believe one particular engine has two combustion-chambers. The upper one is used for idling and low-speed running, and the main combustion-chamber is for higher power. The small combustion-chamber has always been fired first, the resulting high temperature and pressure making it possible to burn very poor mixtures in the main cylinder. The compression can be kept more nearly constant in an engine of this type. This engine is only in the experimental stages and I cannot give very much information concerning it.

AERONAUTIC SAFETY FUEL-TANK AWARDS

A SUBJECT that received much attention during the war, fireproof fuel-tanks for aircraft, was the basis of a recent competition in England, the results of which are reviewed in *Flight* for May 4, 1922, p. 261, and in *Aeronautical Engineering* for May 10, 1922, p. 336. The winning safety tank is described and illustrated in *Aeronautical Engineering* for May 17, 1922, p. 354.

Prizes were awarded to three successful contestants after a preliminary series of tests followed by final tests on four tanks of each of three makes, the latter tests including a firing test of five rounds from a machine-gun in which armor-piercing and incendiary ammunition was used, and a crash test in which the tanks mounted in a fuselage structure were crashed to the ground at a speed of about 50 m.p.h., this speed being considered far above that at which a pilot would have any chance of survival.

Three types of tank seem to have met these tests successfully. Two of them consist of specially designed sheet-metal, enclosed in rubber casings. The third is made of two rubber-composition containers, one within the other, separated by a space filled with a non-inflammable gas under a slight excess pressure.

The basis of award for the competition was as follows:

(2) Remaining attributes, 100 points maximum
(1) Crash-proof qualities, 100 points maximum

The second group was subdivided as follows:

Durability under service conditions in the absence of accidents	25
Indifference to extremes of temperature	25
Adaptability of design to large capacities	10
Simplicity of construction	10
Adaptability of design to various shapes	10
Accessibility of fittings	10
Cost of production	10

A practicable crash and fireproof fuel-tank for commercial aircraft should eliminate one element of risk that the general public considers very important in securing safety in aviation.

Principles of Motorbus Design and Operation

By G. A. GREEN[1]

SEMI-ANNUAL MEETING PAPER

AFTER the presentation of this paper at the Summer meeting of the Society, it was discussed by some of the members present, and replies to various points raised were made at that time by the author. The stenographic report of this discussion has been submitted to those participating, for correction or amplification, and such changes, where received, have been incorporated in the report of the discussion printed below. For the convenience of the members, a brief abstract of the paper, which was printed in the July issue of THE JOURNAL, precedes the discussion.

ABSTRACT

IN the paper an attempt is made to answer the broader phases of the questions: What constitutes a bus? and In what respects does a bus differ from other classes of automotive equipment? by establishing the principles on which the design and operation of motorbuses should be based. The treatment of the subject is in the main impersonal, although specific references to the practice of the Fifth Avenue Coach Co. and illustrations of its equipment are made to emphasize the points brought out. The questions of the unwisdom of overloading, rates of fare and the service requirements are discussed briefly as a preface to the paper proper.

The factors controlling bus design are stated to be (*a*) safety, (*b*) comfort and convenience of the public and (*c*) minimum operating cost. The various subdivisions of each are commented on in some detail, and numerous illustrations and tabular data supplement the text. The conclusions reached are that trucks or automobiles, either modified or unmodified, are absolutely incapable of rendering satisfactory and economical service as buses; such failures of buses as have occurred were due to the combination of extemporized equipment, indiscriminate operation, overloading and lack of experience; and, if the Society would concentrate its standardization work on the motorbus, much good could be accomplished.

THE DISCUSSION

GEORGE W. CRAVENS:—Some years ago I had charge of locomotive design for the General Electric Co. Among the locomotives we designed were the first ones for the New York Central Railroad, which were made with the motors as the frames of the trucks. The center of gravity was very low; in fact, most of the weight was in the trucks and much of it was unsprung. The result was that the first time the New York Central locomotive tried to make high speed on some of the curves it tipped over, due to the fact that the rails tipped over because the center of gravity was so low and the weight mostly unsprung. I state that first because I want to ask Mr. Green if any of his experiments showed that the center of gravity might be too low for safety and for comfort, but especially from the standpoint of safety due to putting excessive side-strains on the wheels when striking obstructions in going around curves. I notice that he favors the steel wheel. I want to raise the question as to the effect of low center of gravity throwing excessive strains on the wheels, causing side-strains on the bearings and springs, and what effect that has upon the riding qualities as well as on the maintenance of the running-gear.

For interurban bus-transportation, has the double-deck bus ever been used successfully? Is it better on general principles outside of cities, where turning radius is not important, to use a short wheelbase with a double-deck or a longer wheelbase with a single-deck bus? About what does experience show as a fair average seating-capacity that should be satisfactory for the purpose?

HARRY A. TARANTOUS:—In Table 5 Mr. Green mentions a material charge of $759.81 for chassis repairs. Is there any cost for sleeve replacement during the year included in that? Is any special grade of gasoline used, or is a standard gasoline sold in New York City employed?

B. S. PFEIFFER:—Is there any difference in the gasoline consumption per ton-mile of the J-type and the L-type buses?

MERRILL C. HORINE:—There has been some talk of the danger of excessive overhang back of the real axle, because of the necessity of turning in congested thoroughfares. Is that really a factor and is there a possibility of side-swiping a neighboring vehicle on the turn?

It is a popular notion that the outside stairway is a danger, because of the possibilities of the passenger falling off. I myself have had to hold on pretty tightly in making stops on Fifth Avenue buses, but I have never fallen off. Did anyone ever fall off?

A very low frame and floor with ordinary-size wheels requires an exceedingly large wheelhouse which, in the case of some buses, entirely eliminates leg-room from at least two seats in the bus. Is there any cure for that, if it is found to be a real disadvantage in practical operation?

In connection with the flat floor Mr. Green advocates in the driver's compartment, that means that the driver has no toe-board. Does not that require the driver to ride the clutch continuously; if so, is there any objection to that if the clutch throw-out bearing is properly designed?

With further reference to the clutch, does Mr. Green favor the use of a clutch brake? I had not noticed much clutch-brake effect in shifting gears on Fifth Avenue buses, and I wondered if there is any reason for that.

I notice that the frames are straight in Mr. Green's designs. There must be a reason why he has not gone to the rather radical drop and gooseneck frames that we are beginning to see on some of the special bus designs.

T. V. BUCKWALTER:—On the question of height of center of gravity, Mr. Cravens has explained why they found it desirable to build a high-center-of-gravity New York Central locomotive, and asked whether that applies to our automotive practice. The first electric locomotives on the New York Central were built with a very

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low center of gravity and the motors were built on the axle. A rather small wheel for a locomotive, about 44 in. in diameter, was used, and it was found that the riding characteristics were very bad. The locomotive had an exceptionally severe surge on the rail; when the wheels struck a curve, they would hit the rail with a series of jerks. The rails would not stand up under this surging. I think it has been corrected by the redesign of the locomotive to use trailer trucks which ease the locomotive in going around curves.

Before the Pennsylvania Railroad Co. put its locomotives into service in 1910 at the time of the New York electrification, it built a series of Brinell testing machines, four of which were put on each tie, and these testing machines were spaced along about 1 mile of track. They tried out all the various kinds of electric locomotive they could obtain, in comparison with steam locomotives. After a high-speed run they would take the Brinell reading in horizontal and vertical directions on each one of these ties for possibly 1 mile of track, an expensive and laborious operation but well worth while in developing where the center of gravity should be. It developed from this test that the low-center-of-gravity locomotive had the most destructive effect on the roadbed and, as the other types of locomotive were tried out and the center of gravity brought up higher, as I recall it to between 70 and 80 in., the destructive effect on the roadbed and likewise on the locomotive was lessened, until they reached the old type of high-speed passenger-locomotive with 80-in. drive-wheels and the boiler set on top of the frame. One of the earliest of the high-up locomotives with a clear space under the boiler over the wheels was the easiest riding locomotive of them all; and it was the only one that attained a speed of 107 m.p.h.

To apply this to the automotive situation, I am inclined to believe the reason the low center of gravity is so destructive in railroad work is that the surging must be resisted by one set of flanges, the flanges on the outside of a curve; the inside set of flanges has no function in keeping the equipment on the track. The reverse is true in automotive practice. We have four flanges, flanges on both sides. The adhesion of the tires to the ground constitutes the flange action and, by keeping the center of gravity fairly low, we get the effect of double-flange action. In other words, the outside wheels possibly have increased adhesion due to the increased weight that is due to centrifugal force; their effect in keeping the vehicle on its road is reinforced by the weight remaining on the inside wheels. Therefore, I think it is desirable that the center of gravity be kept fairly low.

In regard to the future of the bus as compared with the electric railway, the railway men saw the electrical manufacturing companies practically develop an electric locomotive. One of the reasons for that is that the railway men are so set in their ways that they cannot get into the habit of thinking originally on a new line of work. Mr. Green has given us a fine example of original thought on a new problem. He has thrown into the discard the practices that he has seen on other lines of automotive apparatus and has attacked this problem with a clean slate, working out each of the various details as a problem to be attacked from the ground up. That his solution has been wonderfully successful, as it has, indicates that this is the proper procedure by which to handle a problem of that nature.

The railway men have not been inclined to do that. When the first rail-car bus came up for discussion, some of the earlier specifications were to the effect that it must have Master Car Builders' trucks, a buffing resistance through the center sills of 550,000 lb. and the like. All of that applied 10 to 15 years ago to the gasoline rail-car and, at that time, they would not permit us to disregard those features. The cars built at that time could not be anything but failures; a problem of that kind must be attacked as a new problem and we must forget all our ancient theory.

I feel that there is a great future for the motorbus because the bus can receive and discharge its passengers at the curb and leave the center of the street free for other traffic. I believe that this one feature alone will increase the capacity of our streets in larger cities at least 100 per cent; that is, double the traffic over a street that has electric railway equipment, which in a good many of our crowded communities, where the streets are not wide, requires the stopping and holding-up of all automotive traffic because of the presence of the track-confined vehicles.

The second great advantage of the bus consists of the facility with which additional buses can run by one that is stopping to receive or discharge passengers; whereas, in comparison, the track-confined vehicle holds-up all following traffic when stopping for any cause. In like manner, when any trouble does develop in a bus, it can be run up to the curb or into a side street without creating a general hold-up of traffic; whereas it is a common sight to see electric rail-equipment bunched, often due to some minor defect or adjustment that, until corrected, holds-up traffic on that particular track. To sum up, the track-confined vehicle has a general tendency to run in bunches; whereas the tendency of the self-contained bus-system is to distribute traffic over the entire route.

We have endeavored to approach the builders of street-railway equipment with the thought of modernizing some of their equipment. Their general line of defense is that, we aim to give our customers what they want. The electric-railway people, on the other hand, say, We take what the street-railway-equipment company has to offer, and that, because it does not offer certain things, we do not consider them. That vicious circle is one of the things that prevents the established companies from developing and it is one of the reasons why the gasoline motorbus will succeed ultimately and displace a considerable amount of our electric-railway equipment.

Herbert Chase:—To what condition does Mr. Green attribute the unusually low friction that he says is characteristic of the Knight engine? I am aware that his company is using or is planning to use an oil of lower viscosity than is used customarily for that type of engine, but I fail to see why the friction of the Knight engine should be lower than that of an engine of the poppet-valve type.

I am sure that Mr. Green will agree that the bus as it is developed today, even by the Fifth Avenue Coach Co., is not an ideal vehicle in all respects and that there are many possibilities of improvement. It is my belief that these possibilities should be considered; that they should not be overlooked, as they are apt to be, through an effort to apply our present-day type of engine, gearset and the like, to the problem rather than design a vehicle primarily from the standpoint of the service to which it must be applied. The steam-operated bus, for example, has certain inherent advantages, such as high starting-torque, smooth and rapid accelerating ability and the elimination of a gearset. These advantages are realized in the steam buses which have been used in London to some extent for many years. What are Mr. Green's views about the possibilities in this direction?

E. W. WEAVER:—In speaking of the necessity for economy, Mr. Green brought out the fact that a low speed of revolution of the engine per minute and slight opening of the throttle tends to give economy. With the throttle barely cracked the engine has a very low volumetric-efficiency. If it were possible with a different gear-ratio to cut the number of revolutions per minute in two and increase the efficiency, I think the gasoline consumption could be cut down. I am working on a proposition tending in that direction.

A. A. BULL:—In Table 5 chassis repair is all grouped under one heading. Would it be possible for Mr. Green to separate this item so as to indicate in a general way which part of the chassis or component constitutes the greatest item of repair expense?

O. C. BERRY:—I wish to comment on the advantage of the Knight engine as compared with the poppet-valve engine. I am not sure that the Knight engine is superior. The idea that a higher compression can be used in a Knight engine than in a poppet-valve engine is contrary to my experience. Has Mr. Green any special data to bear that out? Secondly, Mr. Green says that the spark-plug is better placed in a Knight engine than in a poppet-valve engine. From my way of looking at it, the spark-plug should be placed at the point where the mixture is richest. In the Knight engine the spark-plug is placed at the point of leanest mixture. This is a matter of considerable importance and seems to offset entirely the other obvious points of advantage. The best figures I have been able to obtain indicate just about equally good performance in the Knight engine and the poppet-valve engine.

HAROLD W. SLAUSON:—Were any tilting tests made to determine the relative stability of the pneumatic and solid tires of the same capacity?

CORNELIUS T. MYERS:—As people gain experience in any industry they accumulate not only a number of positive facts that are very valuable, but certain negative ones. In between the lines of Mr. Green's paper, in which appears the accumulated experience of many years, a lot of "don'ts" will be found. Mr. Green could serve us to a still greater extent if he would put down some of the big "don'ts" that are written down in his book of experience. There are many things that one should not do; these have been tried and found wanting. In that connection, they would be particularly valuable to executives, because executives of street-railway companies and those considering going into bus transportation of any kind must depend upon opinions that at times are not very reliable. The information may be given with all honesty and to the best of the ability of those concerned, but it does not have that reliability that is absolutely essential to the proper development of the motorbus business. The editor of *Railway Age* was inclined recently to be a trifle cynical in viewing the aspects of the rail bus. He said this rail bus appears in cycles of seven years' average and seems to be now in the fourth cycle. I think the discussion of his views showed very clearly that in the previous cycles the railroad men had to depend entirely on their own viewpoints and experience, simply taking the general broad idea of applying a gasoline engine to operating a rail car. If steam-railroad executives and street-railway executives who are considering this matter will call on the experience of the automotive industry, work with it and pay particular attention to the "don'ts" that are written in the books of experience of those who have been in this business, I think we shall have a comparatively rapid and a safe development.

ARTHUR J. SCAIFE:—Mr. Myers' remarks are similar to remarks made during the discussion on the subject of the rail motor-car which was entered into at the meeting of the Metropolitan Section of the Society held in New York City recently. With reference to Mr. Buckwalter's remarks, there was considerable prejudice against the motorbus, and especially on the part of the electric-railway companies. This, however, has disappeared within the last 2 years. The street railways have evidently come to the conclusion that it is much better to get into the bus business than to fight it.

I visited California recently and had opportunity to study the bus business. We have 400 to 500 buses or stages operating in that territory, a number of which are on long-distance work. They are all operating under franchises at present. This business started several years ago in a manner similar to that of a great many of the bus lines that operate today in the Eastern States. On long-distance work the Westerners are, I think, about 3 years ahead of us in the East. A few of the operators have bought up all the small franchises, so there are only about three large operators in California at present.

The buses operate at about 35 m.p.h., maximum. The average wheelbase is 200 in. Each bus carries from 18 to 25 passengers. The later models are equipped with 36 x 6-in. pneumatic front tires and 36 x 6-in. dual rear tires. The rear seat usually is located directly over the rear wheels, similar to passenger-car design. They have a seating space from 32 to 25 in. It is necessary to have ample room for comfort when carrying passengers from 126 to 500 miles. For instance, you can leave San Francisco at 8.00 a. m. and arrive in Los Angeles at 11.00 p. m., on the same stage, using three drivers.

I took a trip from Los Angeles to Bakersfield, a distance of 126 miles, the trip being made in 5 hr., which is an average speed of 25 m.p.h. This stage carries 18 passengers, has a four-speed transmission, with a maximum vehicle speed of 35 m.p.h. In going to Bakersfield we passed over three mountain ranges, having 1130 curves on the 29 miles of mountain; the maximum grade is 9 per cent and the average grade 6 per cent. They maintain a very good average speed over these ranges and around the curves, but if you are subject to sea-sickness do not ride in the back seat, as they say that nausea is very common when sitting there.

There is no doubt that motorbuses are coming to stay, but just what the ultimate bus will be for suburban, city or interurban work is a question that will have to be settled within the next 10 years.

R. E. PLIMPTON:—Mr. Green mentions an acceleration of 2 m.p.h. per sec. I think that applies mainly to buses of the cross-seat type and is too high for those with the longitudinal seats. For the latter 1½ m.p.h. per sec. is probably the maximum that should be used. At 2 or 2½ m.p.h. per sec. passengers are thrown toward the driver or to the rear of the vehicle when it is being started or stopped.

Mr. Cravens brought up the matter of double-deck buses for inter-city operation. Even in smaller towns the bus operators often find that the standard single-deck bus gives trouble because of trees or other obstructions. On long-distance work there are likely to be many obstacles of that sort, even though the traffic is sufficiently dense to warrant the double-deck vehicle.

G. A. GREEN:—Mr. Cravens asks if we have ascertained, as a result of our experiments, whether it is possible to reach a point where the center of gravity is too low; since, under such circumstances, excessive stresses would be imposed on wheels, bearings and springs, thus adversely affecting the riding properties and the main-

tenance costs generally. With the bus, from a low-level standpoint, ground-clearance is the limiting factor. With our L-type bus this has been cut down to 6 in., which we consider closely approaches the minimum. We do not believe a lower center of gravity is either necessary or desirable. With our design the riding properties can be considered as good, and, compared with our high-level equipment, there is no appreciable increase in maintenance costs. But vehicles with a low center of gravity do present many serious problems in regard to the side stresses mentioned. These problems are not at all easy of solution.

It is rather interesting to note that the appearance of a vehicle is subject to marked change as its height is decreased. A vehicle lower than our L-type bus gives an impression of a box formation and, consequently, the artistic value of the finished product is greatly lessened.

Answering Mr. Tarantous' questions, the $759.81 covering material used in connection with repairs to the chassis as per Table 5, naturally includes repairs and renewals to sleeves. The renewal of sleeves cannot be considered as particularly serious. They are not very expensive. They should last at least a year and their replacement is a comparatively simple matter.

With regard to the matter of fuel, standard gasoline as sold in New York City is employed exclusively by us. This gasoline meets the Navy Department's specifications.

Referring to Mr. Pfeiffer's query relative to gasoline ton-mileage with our L-type and J-type buses, since the same powerplant is employed in both cases, a marked difference is not to be expected. We do, however, obtain a somewhat better average with the J-type bus. Primarily, this is due to minor refinements in connection with the engine design. With regard to the matter of fuel-consumption expressed in terms of miles per gallon, in this respect the J-type bus shows a very marked improvement over the L-type bus. This, of course, is natural because of the lighter vehicle-weight and less number of passengers carried. As a rule, a single-deck bus is expected to accelerate more rapidly than a double-deck bus, and acceleration is very costly from the standpoint of fuel. In this connection it seems likely that the average builder and user of single-deck buses will be somewhat disappointed with his fuel-consumption figures.

In reply to Mr. Horine, a rear overhang of an abnormal length is certainly a hazard. It is clear that such construction necessitates particular skill on the part of the driver to clear obstacles when rounding corners. Another very serious matter is that of riding properties. Naturally, if the wheels are too far from the rear end, the passengers on the rear seats will be subject to considerable movement.

The danger of the outside stairway is more apparent than real. So far as my recollection goes, we have had only one or two accidents due to people falling from the stairway. In view of the fact that we have carried in excess of 300,000,000 passengers, the risk cannot be considered as particularly serious.

Large wheel-housings are unquestionably most unsatisfactory from the passengers' standpoint. Of course, the depth of the housings is automatically decreased due to the wide track and frame, the necessity for which is emphasized in my paper. In short, the outer edge of the tires should closely correspond to the extreme overall width of the body and the shape of the housings should conform as nearly as possible to the adjacent members such as brake-drums and the like. The best proposition appears to be a cast or pressed housing with rounded corners and an integral panel.

Regarding the necessity of a flat floor at the front end, this is desirable primarily to avoid accidents, since obstructions of any kind are liable to interfere with ingoing and outgoing passengers. In any event, the natural position for the driver's feet is in the horizontal plane.

Concerning comments on the clutch, a clutch brake is not employed in our service due to the fact that up to the present we have not been able to develop a satisfactory design. The conventional arrangement, while helpful when changing into higher speeds, is decidedly a deterrent when changing down. It must be admitted that many of our men are not expert in gearshifting; but it will be readily understood that we are extremely anxious to keep to our schedules; in short, to employ the highest possible safe speed. In considering that 1 per cent of lost time represents in round figures $16,000 in the form of wages and $50,000 as receipts annually, it will be recognized at once that some damage to the gearing is not extraordinarily serious. Lastly, it is our aim from the standpoint of design to provide everything that is essential but nothing that can be dispensed with, and the clutch brake naturally falls under this heading.

As to straight versus drop frames, our L-type bus has a kick-up at both the front and the rear; but the J-type bus has this at the rear only. We believe a double kick-up is necessary only for double-deck buses where the frame must be kept extraordinarily low. The double-deck bus is intended for operation on city pavements that are usually in fair condition. The single-deck bus is a utility vehicle; this class of vehicle will have a much wider use and therefore the matter of road-clearance must be taken into account. Of course, in many cases single-deck vehicles will be operated over very bad roads. In view of these facts it would seem bad policy to construct vehicles the frame design of which positively restricts their employment to certain localities.

Replying to Mr. Chase, as a result of many tests carried out to determine friction-horsepower losses in connection with various types of engine, we are satisfied that our Knight-type engine possesses remarkable properties in this respect. Table 1 gives figures representing an average of many tests which bring out this point clearly.

TABLE 1—FRICTION HORSEPOWER

Engine Speed, r.p.m.	Friction Horsepower
600	2.2
700	2.6
800	3.1
900	3.8
1,000	4.5

The fact of the matter is that with the poppet-valve engine a surprisingly large amount of power is consumed in connection with the operation of the valves and valve mechanism. In this connection it is suggested that builders of poppet-valve engines might gain something by studying this question even more closely in the future than they have studied it in the past. The best proposition is to carry out what we term progressive friction-horsepower tests. The thought is to check up electrically the power required to turn the engine over, starting first with the crankshaft in its bearings and then gradually adding the reciprocating parts, such as the valve mechanism, valves and the like.

Regarding the matter of design, insofar as our product is concerned, we can only say that we are building vehicles that have embodied in them many of the principles that we believe are essential to permit of safe, economical and

satisfactory operation. Of course, the design cannot be considered as perfect. From a motorbus standpoint the automotive industry is in its very early stages, and any design must of necessity be in some respects out-of-date even before its production could be commenced, assuming of course that there was no delay.

We realize that there are inherent advantages in other types of propulsion. I thoroughly agree with Mr. Chase that steam has wonderful possibilities. An entirely new field is opened in connection with the application of steam along automotive lines for the hauling of light loads on steel roadbeds. This point is drawn particularly to the attention of those who are enthusiastic in connection with the use of gasoline-propelled railroad-units. We recognize the merits of steam, but our entire organization has been trained along gasoline-usage lines. We therefore feel, at least at present, that we should concentrate our efforts on the development of the product with which we are most familiar.

Mr. Weaver has criticized my statement to the effect that the entire theory of design should be based on the highest safe vehicle-speed for the smallest throttle-opening and consequent minimum number of engine revolutions. His contention is that, under such conditions, the engine has a low volumetric-efficiency. Substantially Mr. Weaver is correct and possibly my remarks are somewhat misleading, but they are not intended to be taken too literally. As a result, in our service the throttles are sufficiently open to avoid the condition mentioned. About the only time they are almost closed is when the vehicles are at rest and the engines idling.

Mr. Bull has requested that a further analysis be made of Table 5 of my paper. He desires that the item of $759.81 for material used in repairs to chassis be subdivided, preferably under the various units. Obviously this will demand a considerable amount of study. In view of this, perhaps it will be considered sufficient to indicate the units in order of their importance from the standpoint of renewal and repair cost, as follows:

(1) Engine
(2) Transmission
(3) Brakes
(4) Clutch
(5) Rear Axle
(6) Front Axle
(7) Steering
(8) Radiator

Mr. Berry's queries are answered as follows: Regarding our contention that a higher compression can be used with the sleeve-type than with the poppet-valve engine, we feel this is due primarily to the absence of valve-pockets and the spherically shaped combustion-chamber. Our statements are based on numerous comparative tests that have been carried out in connection with our own service.

Regarding the center spark-plug location this may be incorrect theoretically, but we doubt it. As a matter of fact, we have accumulated an almost unique record from the standpoint of fuel economy. Our figures show consistent economy and are superior to anything we have been able to obtain with poppet-valve engines, regardless of the spark-plug location.

As to the matter of fuel economy, we average in excess of 7 miles per gal. with our double-deck buses throughout the year. With the single-deck vehicles we average from 10 to 11 miles per gal. Recently we sent a J-type bus to Philadelphia on a test run. The fuel average was $14\frac{1}{2}$ miles per gal.; the speed average $22\frac{1}{2}$ m.p.h. It was not necessary to add any water, and but 2 pints of oil was consumed.

If I understand Mr. Slauson's question correctly, he has asked whether we have made any tilting tests to compare the relative stability of vehicles equipped with pneumatic, semi-pneumatic or solid tires. All our tilting tests have been made with solid tires, but, of course, assuming wheels having the same diameter, we would not expect to find any difference in regard to stability.

The wisdom of preparing a series of "don'ts" such as was outlined by Mr. Myers appears somewhat debatable. Those interested in the construction or operation of buses unquestionably will be able to read between the lines of my paper, and sufficient information has been furnished to enable the acceptance or rejection of the various theories propounded. The power of suggestion is the thing that really counts and, according to my viewpoint, no man, regardless of his standing in the industry, should have the temerity to definitely state before the Society what it should or should not do. He should simply give his views, or, better still, the views of his organization and the reasons underlying them.

As to the utility of various forms of equipment, it would seem that any operation requires two types. In the majority of instances initial operation can profitably be commenced with single-deck vehicles; then, after the service is built up, double-deck buses should be added. Even after the service has reached a point where double-deck vehicles are required, the single-deck type will be found extremely valuable to aid in the natural process of development and for operation during cold and wet weather, or for all kinds of special service.

The Saurer Bevel-Gear Testing Machine

THE extraordinarily high standard of workmanship needed in gearing if it is to be silent, efficient and durable under the conditions of speed and loading that it is required to stand in modern design has made it necessary to devise machines capable of detecting the minutest want of accuracy. Only by such means can the necessary corrections to the finished gear be determined, and errors in the gear-cutting machines rendered noticeable in time to prevent a continuance of unsatisfactory work. In the case of bevel gears the possible errors are very numerous. The wheels may be distorted in shape, the tooth-forms may be incorrect, the teeth of irregular pitch, the pitch cones not in uniform contact and so on. The Société Anonyme Adolphe Saurer, of Arbon, Switzerland, which was manufacturing gears of very high class for use in its heavy motor vehicles, realized the necessity for some accurate and satisfactory method of testing the finished gears, and consequently devised for its own use machines for both spur and bevel gears.

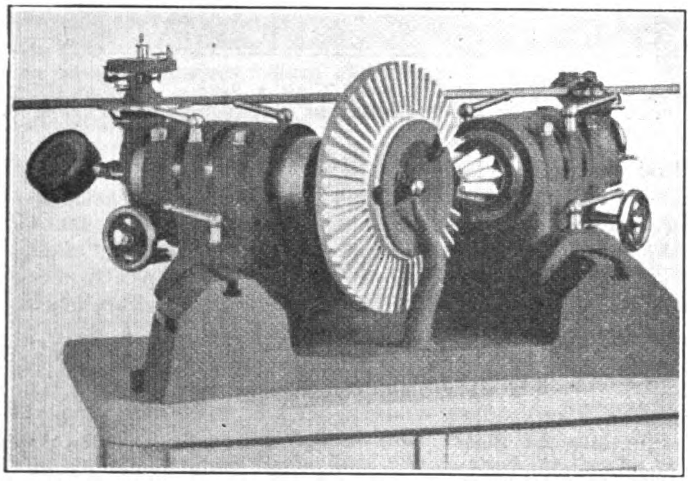

FIG. 1—GENERAL VIEW OF A MACHINE THAT HAS BEEN DEVELOPED FOR TESTING THE ACCURACY OF BEVEL GEARS

The bed of the latter machine, as can be seen from Fig. 1, is a heavy, circular casting, upon which a pair of sliding heads can be locked in any position, so that the angle between the axes of the heads corresponds to that of the bevels to be tested, a range from 52 to 150 deg. being obtainable. A scale on the bed, provided with a vernier, allows this angle to be set with great accuracy. The wheels to be tested are mounted as shown. To check for the existence of bodily distortion of a wheel, the procedure is as follows. The large milled nut *a*, Fig. 2, at the back of the wheel is loosened, the wheel being then pressed forward by a spring so that its teeth go right home between the teeth of the bevel, which may be either a standard accurate pinion, or the mate of the wheel, as desired. The pinion is then rotated by the small handwheel *b* at the right end of Fig. 3. Any distortion of the wheel will cause the spindle of the latter to oscillate endwise. The motion of the end of the spindle is magnified by the pointer of the gage *c* shown at the left-hand of the machine in Figs. 2 and 3 and any irregularity is clearly shown by the motion of the pointer on the scale, one division of which represents 0.01 mm. (0.03937 in.) of movement of the spindle.

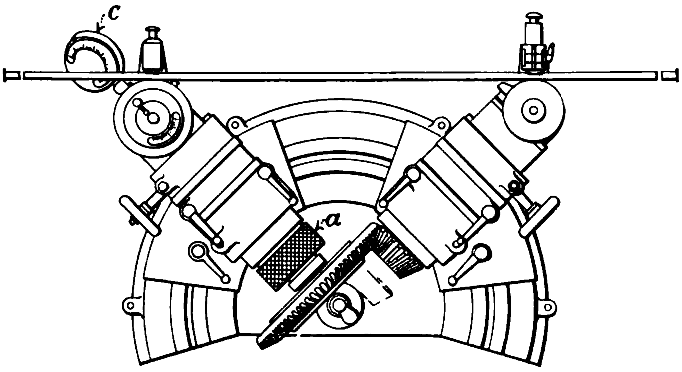

FIG. 2—VIEW LOOKING DOWN ON A PORTION OF THE MACHINE SHOWING THE RELATIVE POSITION OF THE VARIOUS PARTS

To test for regularity of tooth-form and evenness of

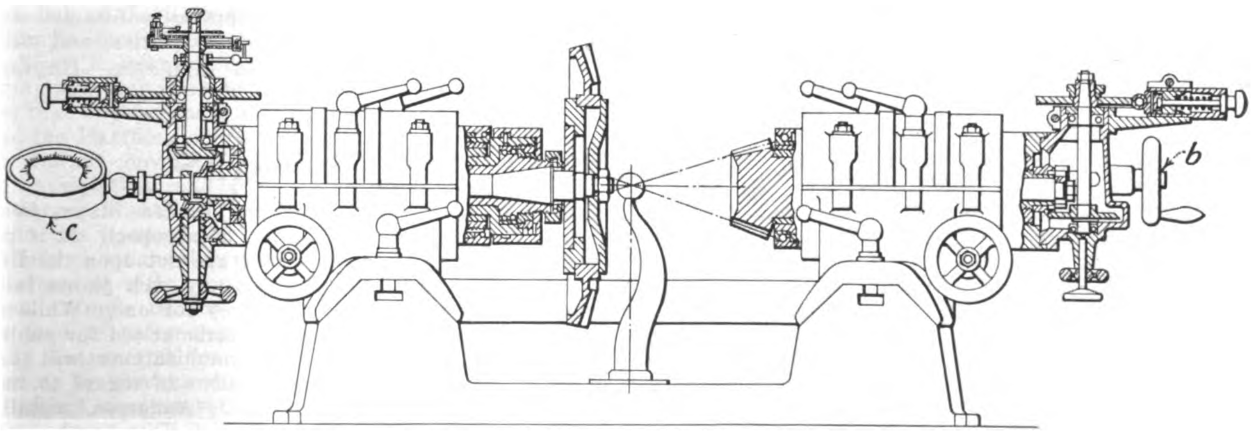

FIG. 3—ELEVATION PARTLY IN SECTION OF THE MACHINE

pitch, the wheels are set to mesh correctly. This is done by end-gages that determine the position of the spindles in the heads. Each spindle drives a vertical shaft at its outer end by a friction gear. The vertical shaft at the right-hand end of the machine carries a plain friction-disc outside the casing. The shaft at the other end of the machine is hollow and carries an indicating and recording instrument which rotates, of course, with it. Up the center of this hollow shaft is a fixed pin on the top of which is a circular table upon which the graphic records are drawn on charts. With this arrangement rotating either of the main gear spindles drives one friction-disc idly and rotates the indicating and recording instrument at the other end of the machine round the fixed chart. Under these conditions the pointer would remain stationary relative to its scale, and the pen would draw a circle on the chart. At the end of the machine that carries the instrument, there is, however, another friction-disc that can be rotated about the vertical spindle. These two friction-discs are coupled together by a rod at the back of the machine, the rod being kept pressed hard against their edges by spring-controlled rollers. The diameters of these friction-discs are so adjusted, in conjunction with the diameters of the friction bevels already mentioned, that the disc underneath the indicating and recording instrument would be driven at precisely the same speed as the instrument itself, if the speed-ratio of the bevels under test was absolutely constant.

A finger rotating with the latter disc is brought into contact with a stop on the instrument and is adjusted so

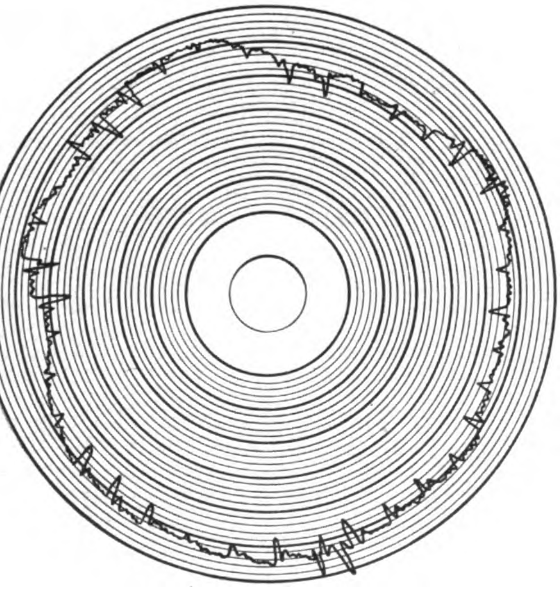

Fig. 4—A Chart Made by the Automatic Recording Mechanism of the Machine Showing Variations in the Speed-Ratio of a Pair of Bevel Gears Undergoing a Test

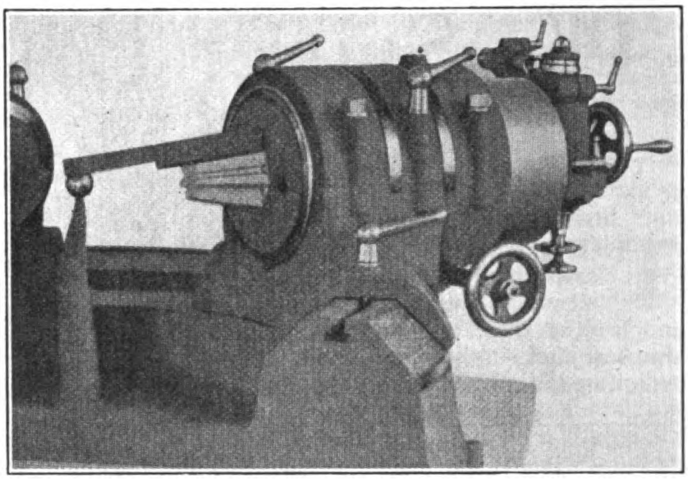

Fig. 5—The Machine Arranged With a Special Gage to Test the Angularity and Straightness of the Individual Teeth

that the pointer is moved to the center of its scale, and the pen also takes up its mean position on the chart. Any irregularity in the speed-ratio of the gears being tested, will cause the finger to move the pointer slightly as the two rotate together and will also cause the pen to move radially over the chart. Fig. 4 illustrates a chart drawn automatically in the manner described, by a pair of hardened bevel wheels made by the builder. The original chart was 3¼ in. in diameter, and the circles, 1 mm. (0.03937 in.) apart, corresponded to a magnification of 100 times the relative motion of the friction-discs. The wheels under test were a seven-tooth pinion gearing into a 55-tooth wheel, the pitch being 25 mm. (0.98425 in.) at the outer diameter. Such wheels are used for the rear-axle drive of the Saurer 5-ton truck with a 40-b.hp. engine, and in spite of the boldness of the design, it is stated that they have given excellent service.

For testing the straightness and angularity of the individual teeth, the gage shown in Fig. 5 is employed. A ball-ended column fitting into a conical seat in the bed is permanently located so that the center of the ball coincides exactly with the intersection of the axes of any bevel gears that may be tested. A hardened steel gage has a conical cup near one end that fits over the ball, and a true straight-edge is formed along part of the side of the gage. The straight-edge points exactly to the center of the ball, so that by using the gage as indicated in the illustration any defect in the points or flanks of the teeth can be observed.

The operation of the machine is very simple and needs no more care or knowledge than a good tool-room mechanic would possess. It is specially intended for the testing of gear wheels for motor vehicles, and will take gears up to about 18 in. in diameter.—*Engineering* (London).

SANDS USED IN ALUMINUM FOUNDRIES

THE Bureau of Mines, as one of the Governmental agencies directly cooperating in a general research of all foundry sands, organized by the American Foundrymen's Association and the National Research Council, is collecting data on the sands employed for the production of molds and cores in aluminum-alloy foundry practice. There is pressing need of collated information on the sands used in aluminum-alloy founding, and it is hoped, as a result of the investigation, to aid in disseminating knowledge on the subject.

A questionnaire is being sent by the Bureau of Mines to aluminum-alloy foundries in the United States to obtain data for use in the preparation of a report. It is possible that some foundries of this kind are not upon the Bureau's mailing list, and it is hoped that any such plants failing to receive the questionnaire will apply for one. While the replies to the questionnaire will be summarized for publication, the contents of individual communications will be held strictly confidential. Communications in regard to the matter should be addressed to R. J. Anderson, metallurgist, Bureau of Mines Experiment Station, Pittsburgh.

Research Topics and Suggestions

THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.

GEARS[1]

NOISELESS gears are desirable chiefly from considerations of comfort. Nevertheless, since noise in gears indicates improper performance, improvement in quietness may represent a real gain in economy, especially in the change-gear sets and the rear-axle gears, where larger amounts of power are being transmitted. In the case of the accessory drives, which absorb relatively small quantities of power, this economy is of less moment, but here noise is even less tolerable.

Causes of Noise

Among the causes of noise in gears, the most obvious is the actual blow produced when one tooth becomes disengaged before the next tooth takes the load, a condition often found in the past when pinions with too few teeth were used. Inaccuracy in cutting is recognized at present as one of the most prolific sources of noise, but at times even accurately cut gears will become noisy for various reasons, such as, for instance, the misalignment that results from slight wear at the axle bearings.

Noise-producing factors, however, are to be found not only in mechanical effects, but are inherent, more or less, in the various geometrical gear layouts. Some of the variables affecting the amount of noise in theoretically correct gears are: (*a*) uniformity of pressure on the tooth surface and of load distribution between contiguous teeth; (*b*) ratio of sliding to rolling; and (*c*) relative sensitiveness to slight disturbances. For example, it is well known that helical bevel gears make for a quieter drive than do straight bevels, partly on account of the more uniform manner in which the former takes the load, the stub-tooth form is better than the ordinary involute as regards opportunity for interference, and cycloidal gears will hammer on very slight separation of the pitch circles.

It has been stated that resonance effects contribute not only to noisiness but also to excessive tooth pressure in gears. According to this idea a periodic torque impressed on the transmission system by any means, such as by periodic fluctuations of the engine torque, or by periodic errors in the cutting of the gears, will reinforce the natural torsional vibration of a gearshaft whenever the times of oscillation become equal to, or multiples of, each other. At those critical speeds of revolution where the resultant fluctuating tooth-pressure torque is equal to the transmitted torque, there will be a parting of the teeth and hence chattering.

No simple formula can be given for the elimination of noise. That the gear manufacturer is doing his share toward producing smoothly running gears is indicated by the increasing use that is being made of highly sensitive testing devices such as the Hartness optical comparator, recently described at the Buffalo convention of the American Gear Manufacturers' Association, and the Saurer gear-testing machine. The latter, an illustrated description of which appears elsewhere in this issue of The Journal, magnifies and records any difference in angular velocity shown by a pair of gears when compared with a pure rolling of their pitch circles.

As an example of how correct tooth layout may nevertheless lead to noisy gears, the treatment of interference given by Vessey and Seager may be cited, but the general question as to what produces noise in perfect gear teeth is still a subject of controversy, as is illustrated in the illuminating discussion brought out by S. Bramley-Moore's paper on Developments in Transmission read before the Institution of Automobile Engineers in 1920.

The resonance problem had no serious treatment until Dr. J. H. Smith attempted a mathematical analysis of the conditions under which forced vibrations are produced in gearsets. As a result of his final equations, Doctor Smith proposes, among others, a system of gear design called the "nodal drive," which possesses fewer of the critical speeds that give rise to noise and excessive stresses than does a haphazard assembly. A nodal drive, according to Doctor Smith's definition, is one in which the free periodicities of all the shafts, when considered as fixed at their gearbox extremities, are the same. A practical application of this system to the redesign of a set of double-reduction gears in the turbine-propelled ship Melmore Head, is reported by J. Wilkie as having been thoroughly successful. It would be interesting to extend Doctor Smith's analysis to the smaller but more complex gear assemblages used in the automotive industry, as it is probable that his method is capable of indicating what resonance effects are to be expected in this case.

Finally, attention may be directed to Neil MacCoull's attempt to determine the intensity of the sound emanating from a gearbox, and to correlate it with gear efficiency.

Mechanical Efficiency

The power lost in the several sets of gearing that constitute an essential part of automotive transmission and driving systems, is naturally a matter of much concern in this era of intensive study of the separate factors that might contribute to better performance. It is important to know, therefore, how much the mechanical efficiency of gears in general depends upon the choice of the lubricant, the method of lubrication, the speed and the load, and how the numerous varieties of design compare in regard to power loss. Other less pressing questions deal with the possible dependence of efficiency upon various other factors, such as, for example, the elastic and physical constants involved in Doctor Smith's description of the effect of torsional oscillations.

We have answers to very few of these questions; in fact, only the National Physical Laboratory is publishing regularly accurate results obtained by a systematic study of gears, but that work is at present limited to the question of the choice of lubricants.

One of the major difficulties to be overcome before satisfactory experimental data can be gathered, is the design of a testing machine that will prove adequate for indicating accurately the small efficiency variations involved. It seems that this would rule out any method that depends on subtracting the output from the input, even though this subtraction is done mechanically, as in the machine developed by C. M. Allen and F. W. Roys, and used by Heindlhofer for the testing of rear-axle worm drives. The electrical "load-back" scheme as a precision method, proposed by P. F. Walker for testing gear efficiencies, appears to suffer from the same defect. On the other hand, MacCoull's arrangement of double gearboxes represents an ingenious attempt to reduce the problem to the direct determination of the power loss and has promising possibilities.

Another proposed method for the direct-torque-difference determination of the mechanical efficiencies of gears is embodied in Wilfred Lewis's machine designed primarily for

[1] Concluded from The Journal, July 1922, p. 90.

testing the strength of gear teeth. However, only the preliminary work has been done with it, and it is not certain that it will be available for the important investigations on gear efficiencies that must be made in the near future.

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THE OSCILLOSCOPE IN THE STUDY OF MOVING MACHINE PARTS

A DISCUSSION is given in a recent number of *The Autocar*[1] of the possibilities of the portable Oscilloscope as an aid for the designer of automotive machinery, since it permits the exact study and measurement of the movements of machinery parts functioning at high speed. The principle of the apparatus consists of illuminating the moving object to be examined with a series of flashes from an electric lamp, arranged to take place at regularly spaced intervals in the movement of the part. In other words, it enables the eye to take a series of flashlight photographs in regular sequence and in such a way that the part appears to be moving at one-hundredth of the actual speed. By special adaptations of the instrument, it is claimed to be possible to analyze mechanical noises, to get the true periodicity of vibrations with relation to any given shaft without calculation, and to diagnose many of the ills from which engines and other chassis parts are apt to suffer.

[1] See *The Autocar*, March 4, 1922, p. 349.

Special Notice

TRANSACTIONS, PART I OF VOL. 16 (1921).

To Those Who Paid Dues for the First Half of the Fiscal Year Beginning Oct. 1, 1920:

The Council has directed that members shall be charged two dollars ($2) per Part of Transactions of the Society, beginning with Part I of Vol. 18 (1923). For further information on this matter see p. 261 of THE JOURNAL for April.

The two Parts of Vol. 16 (1921) and the two Parts of Vol. 17 (1922) will be sent, without charge in addition to payment of respective dues, to members on request, if request is received by the Society within 60 days from date of notice to order.

This is a notice to order Part I of 1921 Transactions.

The printing order will be based on the number of copies ordered by the members.

Copies cannot be guaranteed to members who do not order them on or before Sept. 30, 1922

If you want a copy of Part I of the 1921 Transactions please tear out and mail the coupon printed below promptly.

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The Standards Committee Meeting

THE action taken at the Standards Committee meeting at White Sulphur Springs, W. Va., on June 20, on the subjects reported by the several Divisions of the Standards Committee, is recorded below. After brief introductory remarks in opening the meeting and the declaration of a quorum by President B. B. Bachman, Standards Committee Chairman E. A. Johnston was introduced and called for the reports of the Divisions as printed in the June issue of THE JOURNAL, together with an additional report on Head-Lamp Illumination submitted by the Lighting Division. The action taken by the Standards Committee on the reports was submitted to the Council and the general business session of the Society on the same day and approved without changes.

The reports as given herein are for the information of the Members of the Society in casting their letter-ballots on the adoption of recommendations that were approved at White Sulphur Springs. This letter-ballot will close and be counted Saturday, Aug. 19, and should be returned by all Members prior to that date.

In every case the complete recommendation or a portion of it is printed below. For subjects that were approved as printed in the June issue of THE JOURNAL, the page reference to that issue only is given. For reports that were corrected, changed or amended, such changes are also given in this issue and should be noted in the reports published in the June JOURNAL. Section C44 of the Constitution of the Society reads as follows:

VOTE REQUIRED FOR ACTION

Every question which shall come before a meeting of the Society or of the Council or of a committee, shall be decided by a majority of the votes cast, unless otherwise provided in the Constitution or By-Laws, or the Laws of the State of New York. The Council may order the submission of any question to the membership by letter-ballot. Any meeting of the Society at which a quorum is present may order the submission of any question to the membership by letter-ballot.

AGRICULTURAL POWER EQUIPMENT DIVISION REPORT

(1) TRACTOR DRAWBAR ADJUSTMENTS

The Agricultural Power Equipment Division recommends that the present S.A.E. Standard for Tractor Drawbar Adjustments, p. K40 of the S.A.E. HANDBOOK, be revised to conform with the limits specified in the table on p. 478

AXLE AND WHEELS DIVISION REPORT

(2) MOTOR-TRUCK FRONT-AXLE HUBS

The Axle and Wheels Division recommends that the tables on p. 476 be approved by the Standards Committee for adoption as an extension of the present S.A.E. Recommended Practice for Motor-Truck Front-Axle Hubs, p. F1b of the S.A.E. HANDBOOK

[This report was approved as printed on page 476 except that in the table under the caption General Information for Motor-Truck Front-Axle Hubs, the title under Spindle No. which read "Spindle Load Rating in Lb. on Solid Tire at Ground" was changed to read "Assumed Values on Which Calculations for Spindle Sizes Were Based."]

THE DISCUSSION

W. G. WALL:—With regard to the groove at the back end of the spindle thread, the width is given but not the depth. Is that assumed to be the same depth as the thread?

CORNELIUS T. MYERS:—No depth of groove is specified in the table. The groove is cut to an ordinary diameter so that the full thread can be run up to the shoulder. If it is thought advisable to specify the diameter, it can be included, but the usual practice is to have the bottom of the groove slightly below the bottom of the thread.

MR. WALL:—This is a detail. It is necessary that this groove be not too deep, but still deep enough that the thread can be cut without trouble. If the groove is made too deep, it simply weakens the end of the spindle.

CHAIRMAN E. A. JOHNSTON:—I think that is a very good point, as it is always desirable to specify dimensions in such cases; otherwise in purchasing material made outside, the undercut may be too deep.

MR. MYERS:—The stresses encountered at that point are so small that even if the usual depth of cut were doubled, it would not be of material difference. In a subsequent report the Division would be glad to submit a recommendation for practice, but we would like to have the present report approved.

PRESIDENT BACHMAN:—There is one point I would like to bring up for the information of the Standards Committee. It was the subject of some correspondence I have had with the Division with regard to the second part of the table dealing with General Information for Motor-Truck Front-Axle Hubs. It relates to the spindle load-rating. I have two thoughts on this particular proposition. One is personal, based on my feeling that the figures as they are given are not complete enough, and that if we want to consider giving such figures for general information, they ought to be extended.

I checked over one or two of them from a static load point of view only. As I recall, the result was a static-load stress of about 6500 lb. per sq. in., which seemed to be rather low. I did not attempt to take into mathematical consideration all of the stresses that would be imposed on the spindle from side-skids and other forces, as I have a certain amount of hesitancy in accepting mathematical results that are based on such premises.

The other matter is more of an official one. The Council has had called to its attention the desirability of formulating what may be called a motor-truck rating-formula, which in its essence would partake of the nature of the engine-rating formula which we know as the N. A. C. C. Horsepower Formula. I am inclined to feel that if anything can be done on that subject, we ought to know it within the next few months, certainly by the Annual Meeting next January. I am also inclined to feel that this particular recommendation as presented may have some complicating influence upon that work. Therefore, I would like to have considered the remarks I have made toward eliminating this portion of the proposed recommendation, including the dimensions, until we can

give it further consideration in whatever Division or Committee may be called upon to consider this other matter in conjunction with the Axle and Wheels Division.

MR. MYERS:—The point President Bachman has made is important. In connection with that, and his fear that anything printed in this report might be used in adverse legislation, I suggest, with reference to the second table, that instead of the title, "Spindle Load-Rating in Lb. on Solid Tire at Ground," the title "Assumptions on Which Calculations for Spindle Sizes Were Made" be used, so that there will be no basis for the figures being taken as definite load ratings. These values are to be published, not as standards or recommended practices; but as collateral data in connection with the work of this Division, which had a new problem to solve and wishes the basis on which it was worked out to be known.

In connection with the other matter that President Bachman mentioned, the Division thought it inadvisable to publish too many data at this time. The static load-ratings are low but, in the opinion of those who passed on this matter, they are not the controlling feature, whereas the dynamic side-skid ratings or stresses are. This was considered very carefully and the sizes were not based entirely upon calculations. The calculations were made so as to be able to compare one size with another and each with the practice as exemplified in the motor-truck industry.

When we had selected a series of sizes that seemed to meet with the approval of practically every one with whom we discussed it, the matter was taken up with the bearing manufacturers and slight modifications were made to use bearings of the most economical sizes.

No adverse criticism, to my knowledge, has come to the attention of the Division with reference to the sizes recommended, and the result of the work shows a considerable possible saving in the cost of bearings, on the basis of the loads selected.

The Division will, I believe, gladly present in detail at the next Annual Meeting the calculations and diagrams on which it based its recommendations. But approval of the work done to date is desired, if it can be put in such form as will meet Mr. Bachman's objections and any others that may be raised in the Council.

PRESIDENT BACHMAN:—The change that Mr. Myers suggests in nomenclature would meet the objection I raised. Possibly the description of the formulation of the report might be given fully at some time in the form of a paper or contributed discussion. I think some valuable work was done that will be of interest to many of the members.

ELECTRICAL EQUIPMENT DIVISION REPORT

(3) BREAKER-CONTACTS

The Electrical Equipment Division recommends for S.A.E. Recommended Practice that the hexagon-head of breaker-contacts and check-nuts shall be ¼ in. across flats and that the threads shall be No. 10-40

(4) IGNITION-DISTRIBUTOR MOUNTINGS

The Division recommends that the present S.A.E. Standard for Ignition-Distributor Mountings be revised by

(1) Specifying a dimension of 27/32 in. for the distance from the bottom of the distributor mounting barrel to the bottom of the tongue on the distributor half of the driving coupling of the Type-B ignition-distributor

(2) Changing the limits for the bore in the coupling from 0.4930 in. maximum and 0.4920 in. minimum to 0.4915 in. maximum and 0.4905 in. minimum

(5) MAGNETO MOUNTINGS

The Division recommends that the magneto dimensions given in the illustration on p. 477 be adopted as S.A.E. Recommended Practice, as an addition to the present S.A.E. Standard for Magneto Mountings, p. B14 of the S.A.E. HANDBOOK

THE DISCUSSION

F. W. ANDREW:—After collecting the data regarding the number of different magnetos installed in the service, covered by the recommendation, it was found that there were more than 50 different sets of mounting dimensions in use. Of the 50, the one that is recommended largely predominates. Rather than make an entirely new standard, we thought it was best to use this one.

(6) STARTING-MOTOR FLANGE MOUNTINGS

The Electrical Equipment Division recommends that dimensions "C," the diameter of the bolt-holes for sizes Nos. 1 and 2 of the S.A.E. Standard for Starting-Motor Flange Mountings, p. B19 of the S.A.E. HANDBOOK, be changed from 7/16 to 13/32 in.

ENGINE DIVISION REPORT

(7) FLYWHEEL HOUSINGS

The Engine Division recommends that the present S.A.E. Standard for Flywheel Housings, p. A1 of the S.A.E. HANDBOOK, be revised to specify that the clearance space for crankshaft flywheel bolts shall be 6½-in. minimum diameter and ¾-in. minimum depth

IRON AND STEEL DIVISION REPORT

(8) LEAF-SPRING STEEL

The Iron and Steel Division recommends the adoption of the specification on p. 480 for leaf-spring steel tolerances as S.A.E. Recommended Practice

(9) STEEL SPRING-WIRE

The Division recommends that the chemical compositions given in the table on p. 480 be adopted as S.A.E. Standard

[This report was approved as printed on page 480 of THE JOURNAL after deleting "for round, cold-drawn wire up to 3/16-in. diameter, except for some types of springs used in clutches, which are hot-rolled," in the paragraph immediately preceding the table, and the word "helical" in the caption of the table.]

THE DISCUSSION

MR. MYERS:—It seems to me that this specification might readily be given in the standard steel compositions with a note to the effect that it is suitable for helical springs, but I think it should not be called "Steel Wire for Helical Springs."

G. L. NORRIS:—I think that when this report is printed in the S. A. E. HANDBOOK it will appear among the standard steel compositions with a suggestion that it is suitable for helical springs. We do not recommend, as I understand it, steels for any purpose; but leave the application to the user.

G. S. CASE:—I have one suggestion as to the wording of the recommendation. The words ". . . except for some types of springs used in clutches, which are hot-rolled" are inappropriate because hot-rolled springs are not made of wire.

MR. NORRIS:—It may be drawn wire that is hot-coiled. Whether the report means "hot-coiled" is not clear. I believe it should read, "The Division recommends that

the chemical compositions given in the accompanying table be adopted as S. A. E. Standard for spring-wire."

MR. CASE:—That is all right.

MR. MYERS:—Not "for spring-wire." That should be in an accompanying note.

PRESIDENT BACHMAN:—I think that the sense of the meeting is clear. If the recommendation of the Division for Nos. 1350 and 1360 Steels be approved and the Division requested to frame a proper explanatory note, that will appear in the Iron and Steel Division's report but not as a part of the recommended practice, the purposes of the discussion will be fully served.

CHAIRMAN JOHNSTON:—I think what Mr. Bachman has in mind is that these steels can be used for many purposes.

MR. MYERS:—I will move the adoption of those two specifications to include only the composition of the steels.

LIGHTING DIVISION REPORT

(10) ELECTRIC INCANDESCENT LAMPS

The Lighting Division recommends that the G10, G12 and G16½ lamps be omitted from the S.A.E. Standard for Electric Incandescent Lamps, p. B3 of the S.A.E. HANDBOOK

(11) ELECTRIC INCANDESCENT LAMP VOLTAGE

The Lighting Division recommends that the present S.A.E. Standard for Electric Incandescent Lamp Voltages be revised by omitting the 8 to 10 and the 18 to 24 voltage-ranges and the reference to four and nine battery-cell arrangements

(12) HEAD-LAMP ILLUMINATION

Part 1—Laboratory Tests for Regulatory Purposes

Due to a rather complex situation that has arisen in connection with this subject, the Lighting Division was unable to make any definite report until after the June issue of THE JOURNAL was published, and the Division's recommendations on this subject are therefore printed in this issue.

The members of the Division feel that it is imperative that action by the Society by way of revision of the present S.A.E. Recommended Practice, p. B6 of the S.A.E. HANDBOOK, for Laboratory Test for Operating Requirements, be taken now inasmuch as such revision will lose its practical value if deferred until the January 1923 meeting of the Standards Committee.

At the meeting of the Lighting Division in November 1921 it was felt that the existing S.A.E. Recommended Practice should be revised in view of the improvements which had been made in motor-vehicle head-lamp equipment and lighting characteristics. A Subdivision was therefore appointed to cooperate in conducting tests which were then being made at the National Lamp Works of the General Electric Co. in Cleveland, and to report to the Lighting Division. While this work was progressing the Committee on Motor-Vehicle Lighting of the Illuminating Engineering Society prepared a report which was approved and published by that Society in February 1922. The Society's Subdivision, upon receiving this report, reviewed it carefully and subsequently reported its recommendations to the Lighting Division at its meeting on May 3, 1922. The Subdivision's report was the same as the published report of the Illuminating Engineering Society except that it included recommendations for two major changes and a few minor changes as a result of the tests that had been run.

The major changes that it was suggested be made in the Illuminating Engineering Society report were:

(1) The addition of a sentence under the paragraph "Requirements for Approval," that would specically state that in the case of different States having the same regulations, approval of a device by a specified laboratory test in one State may be accepted without further test by another State

(2) Under the paragraph "Approval," it was felt that the sentence "No device involving the use of a tilting reflector shall be approved unless it conforms with these specifications in its 'tilted-up' position and has an extreme tilting range of no more than 3 deg." would place an unwarrantable limitation on the development of controllable head-lighting devices intended to furnish superior road illumination for operation under conditions where they can be safely used

The minor changes recommended were

(1) The Subdivision recommended that the italicized portion of the following sentence appearing under Test 1 for a pair of headlamps, should be omitted:

A pair of testing reflectors, mounted similarly to the head-lamps on a car, shall be set up in a dark-room at a distance of not less than 60 *nor more than* 100 *ft.* from a vertical white screen

(2) The substitution of *BL* and *BR* points for the *B* point on the diagram for test positions of the head-lamp beams in the Illuminating Engineering Society's report

(3) 2000 minimum cp. was recommended at point *A* in place of 1800 cp. in the Illuminating Engineering Society's report

(4) 7500 minimum cp. was substituted at points *BL* and *BR* in place of 7200 cp. at points 1 deg. to the left and to the right of *B* in the Illuminating Engineering Society's report. Points *BL* and *BR* are respectively 1 deg. each side of point *B* and are the same as the locations in the Illuminating Engineering Society's report

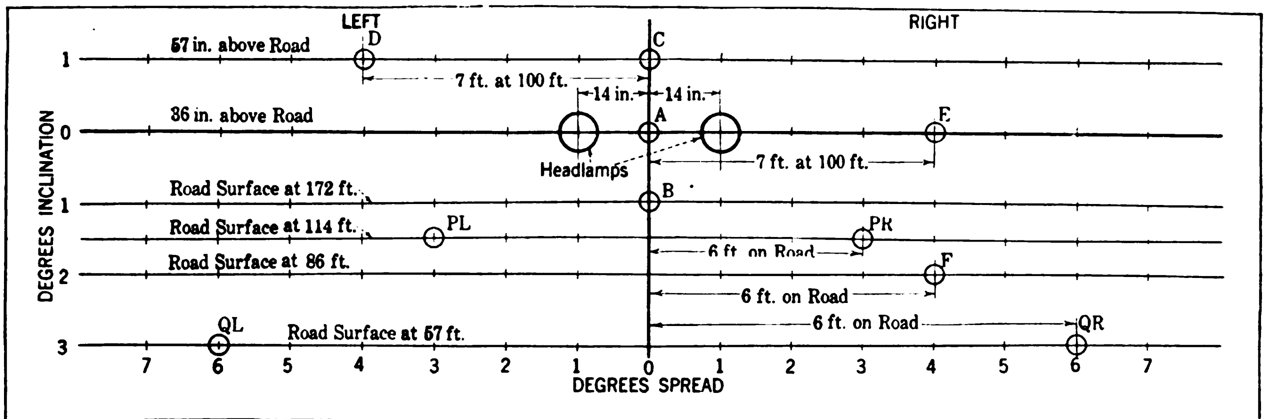
(5) Under the Test 2 for one head-lamp 2000 minimum cp. was proposed instead of 1800 at point *A* in the Illuminating Engineering Society's report

(6) A minimum of 3500 cp. was substituted for 3600 cp. at points *BL* and *BR* as described above

The Subdivision's report was discussed at length, the Division finally voting that since the Society and the Illuminating Engineering Society had cooperated in this work previously and as no opportunity had been had to consider the Illuminating Engineering Society's revised specification with the Illuminating Engineering Society Committee prior to its approval and publication by the Illuminating Engineering Society, the Illuminating Engineering Society Committee should be requested to hold a joint meeting with the Lighting Division for this purpose. The Division felt that but one specification should be approved by both Societies but that the changes noted above, particularly the major ones, were very desirable.

In the meantime a problem in connection with tail-lamp illumination was referred to the Illuminating Engineering Society by the Massachusetts State authorities and a joint meeting of the two Committees was called for and held on June 2. The situation with regard to headlighting specifications was also discussed at this meeting and it was voted that a letter ballot of the Illuminating Engineering Society Committee should be taken on the question of that committee reconsidering its report in view of the recommendation of the Lighting Division. Immediately thereafter the Lighting Division members held a meeting at which the following action was taken:

The Lighting Division appreciates the necessity for uniform laboratory testing specifications for motor-



1920 LIMITS.	
A-B	4800 cp Min.
C	2400 cp Max.
D	800 cp Min.
E-F	1200 cp Min.

1922 LIMITS.		
A	1800 cp Min.	6000 Max.
B	1° Right & Left	7200 Min.
C	800 cp Min.	2400 Max.
D		800 Max.
PL to PR		5000 Min.
QL to QR		2000 Min.

FIG. 1—DIAGRAM OF TEST POSITIONS

vehicle head-lamps as the basis for uniform State legislation but is strongly of the opinion that the modifications referred to above pertaining to "tilting devices," and the acceptance by one State of tests conducted in another, are of prime importance and should be agreed to by the two Societies

It was further voted to waive the suggested changes in measurements and candlepower values in the tests and to recommend for adoption as S.A.E. Recommended Practice the Illuminating Engineering Society revised specifications of February, 1922, provided the major modifications mentioned above should be approved by the Illuminating Engineering Society prior to June 20, 1922

It was also voted that if the Illuminating Engineering Society does not favorably consider the modifications advocated for the specifications, the Division recommend for adoption as S.A.E. Recommended Practice, to supersede the present S.A.E. Recommended Practice, p. B7, S.A.E. HANDBOOK the laboratory test procedure and numerical values set forth in the Illuminating Engineering Society revised specifications, and place itself on record as being opposed to the paragraph on "tilting devices" included in the said Illuminating Engineering Society specifications, and in favor of adding the sentence referring to acceptance by one State of tests made in another State

It was also decided that a definite report should be submitted for approval at the Standards Committee meeting on June 20 in accordance with the above action by the Division.

The Lighting Division thereupon recommended for adoption in revision of the present S.A.E. Recommended Practice on pp. B6 and B7 of the S.A.E. HANDBOOK, the accompanying specifications for the laboratory testing of motor-vehicle electric head-lamps. These specifications are taken from the Rules Governing the Approval of Headlighting Devices for Motor Vehicles, issued by the Illuminating Engineering Society and dated Feb. 9, 1922, and are intended to serve as the basis for legal requirements as to minimum road-illumination and maximum glare.

Part 2—Laboratory Tests for Desirable Illumination

At the same time that the Subdivision was studying the matter of regulatory laboratory-test specifications for motor-vehicle electric head-lamp illumination, it prepared a report on the laboratory determination of candlepower values to give the desirable amount of road illumination under normal operating conditions that would serve as the objective in designing and building head-lamps. Head-lamps conforming to the laboratory test under Part 2 of this report will conform to the regulatory laboratory-tests for minimum illumination and maximum glare specified in Part 1 of this report.

A report was submitted at the Lighting Division meeting of May 3 and approved as a specification separate and apart from that of Part 1 on the proposed laboratory tests for motor-vehicle electric head-lamps.

The points on the lighting chart referred to in the report are the same as those in Fig. 1 in the report for laboratory test specifications given in Part 1, with the exception that the *SL* and *SR* points are additional, as explained in the recommendation in Part 2.

The Lighting Division therefore recommends that the following proposal be adopted in revision of the present S.A.E. Recommended Practice for New Head-Lamp Equipment, p. B7 of the S.A.E. HANDBOOK:

	Candlepower Min.	Candlepower Max.
A	2,000	6,000
B and intermediate points 1 deg. to right or left	25,000	....
C	800	2,400
D	.....	800
PL	10,000	....
PR	10,000	....
QL	4,000	8,000
QR	4,000	8,000
SL	500	....
SR	500	....

SL and *SR*—Four deg. of arc below the level of the lamps and 12 deg. of arc to the left and to the right respectively of the median plane

LABORATORY TESTS FOR MINIMUM ILLUMINATION AND MAXIMUM GLARE VALUES

(A) TEST OF DEVICES USED IN PAIRS

Samples for Test.—The samples submitted to the testing authority shall be representative of the device

as manufactured and as marketed. They shall be accompanied by printed instructions for their use as issued by the manufacturer of the device. The samples submitted shall include as much of the accessory equipment peculiar to the device, except batteries, as is necessary to operate the device in its normal manner. In the case of front glasses the samples shall be one pair each of 8 5/32 (or Ford size) 8½, 9 and 9½-in. diameter when practicable.

Reflectors and Incandescent Lamps.—In the case of devices to be used in connection with standard parabolic reflectors, the reflectors used in making the laboratory tests shall be of standard high-grade manufacture of 1.25-in. focal length with clean and highly polished surfaces, and as nearly truly paraboloidal in form as practicable, and as approved for this purpose by the Bureau of Standards.

The incandescent lamps used in connection with the laboratory test shall be of standard manufacture and as approved for this purpose by the Bureau of Standards. In the case of devices involving the use of special incandescent lamps, such lamps, together with any necessary accessories, shall be submitted.

Marks of Identification.—Each device submitted must bear a distinctive designation prominently and permanently indicating the name and type of the device. Special incandescent lamps submitted in connection with devices shall bear the manufacturer's normal, clear-bulb rating.

Adjustment of Devices.—The testing authority shall adjust the device in accordance with the printed instructions issued by the manufacturer, which instructions must be adequate for practical purposes. An exact description of the adjustment made for test shall be given in the report.

Focal Adjustments of Incandescent Lamps.—The following designations of the focal adjustments of the incandescent lamp in the parabolic reflector are adopted:

- Principal Focus. The beam, with bare reflector or plain front glass, is nearly parallel and of the smallest possible diameter
- Rear Focus. The beam, with bare reflector or plain front glass, diverges as much as possible without having a dark center
- Front Focus. The beam, with bare reflector or plain front glass, converges and crosses near the lamp and then diverges as much as possible without having a dark center
- Special Focus. A special focal adjustment is allowed only when it can be clearly defined and described

Photometric Tests.—The tests shall be as follows:

TEST 1

A pair of testing reflectors, mounted similarly to the head-lamps on a car, shall be set up in a dark-room at a distance of not less than 60 nor more than 100 ft. from a vertical white screen. If a testing distance of 100 ft. is taken, the reflectors shall be set 28 in. apart from center to center, and if a shorter testing distance is taken, the distance between reflectors shall be proportionately reduced. The axes of the lamps shall be parallel and horizontal, or tilted in a vertical plane in accordance with manufacturer's adjustment. The intensity of the combined light shall then be measured with each pair of samples in turn, with the reflectors fitted with a pair of incandescent lamps of the gas-filled type, 6-8 volts, 21-cp. rating. The lamps shall be such as will give their rated candlepower when operated at their rated efficiency. They shall be operated at their rated candlepower.

Measurements shall be made at the following points at the surface of the screen:

- *A.*—In the median vertical plane parallel to the lamp axes, on a level with the lamps
- *B.*—In the median plane 1 deg. of arc below the level of the lamps
- *C.*—In the median plane 1 deg. of arc above the level of the lamps
- *D.*—4 deg. of arc to the left of the median plane and 1 deg. of arc above the level of the lamps
- *PL* and *PR.*—1½ deg. of arc below the level of the lamps and 3 deg. of arc to the left and to the right respectively of the median plane
- *QL* and *QR.*—3 deg. of arc below the level of the lamps and 6 deg. of arc to the left and to the right respectively of the median plane

A diagram of test positions is shown in Fig. 1.

All pairs of samples tested under the conditions prescribed above shall conform to the following specifications for observed apparent candlepower:

- Point *A*, not less than 1800 cp. nor more than 6000 cp.
- Point *B*, not less than 7200 cp., and there shall not be less than 7200 cp. at any point on the horizontal line through *B*, 1 deg to the left and to the right of *B*
- Point *C*, not over 2400 cp., and not less than 800 cp.
- Point *D*, not over 800 cp.
- Points *PL* and *PR*, at each of these points and at every point on the line between them, not less than 5000 cp.
- Points *QL* and *QR*, at each of these points and at every point on the line between them, not less than 2000 cp.

Note.—The above testing directions are drawn specifically to cover the case of devices accessory to parabolic reflectors of 1¼-in. focal length. In the case of other classes of [electric] devices where these directions evidently cannot be applied literally, their intent must be adhered to and the testing positions and candlepower limitations shall govern in all cases.

TEST 2

A single pair of samples taken as an average representative of the device as manufactured, shall be submitted to a complete test with the same testing equipment as specified for Test 1. This test shall show the light-distribution characteristics by actual measurements made in accordance with the best laboratory practice.

Distribution of Samples.—One pair of the samples submitted shall be retained at the testing laboratory for the purpose of future reference and as samples of construction.

(B) TESTS OF DEVICES USED SINGLY

Motorcycle head-lamps are used singly and not in pairs, and have commonly a reflector of smaller diameter and shorter focal length. Hence devices for use in connection with them are not included in the same classification as those for other motor-vehicles. For the laboratory tests of such devices two samples of representative sizes shall be submitted. They shall be tested with representative motorcycle head-lamp reflectors. The numerical limitations of apparent candlepower for Test 1 with one lamp only shall be as follows

- Point *A*, not less than 1800 cp.
- Point *B*, not less than 3600 cp., and there shall not be less than 3600 cp. at any point on the horizontal line through *B*, 1 deg. to the left and to the right of *B*
- Point *C*, not more than 2400 cp.
- Point *D*, not more than 800 cp.
- Points *PL* and *PR*, at each of these points and at every point on the line between them, not less than 2500 cp.
- Points *QL* and *QR*, at each of these points and at every point on the line between them, not less than 1000 cp.

Test 2 shall be made with one lamp and not with two.

Other deviations from the details of procedure are

obviously made necessary because of the fact that single devices instead of pairs are subjects of test.

THE DISCUSSION

L. C. PORTER:—With regard to the Lighting Division's report on Head-Lamp Illumination, I would say that the chief purpose of head-light regulation is, first, to prevent accidents, due to glare, and, second, to provide adequate lighting for night driving. To accomplish these things it is necessary to have specifications for lamps and reflectors that will regulate the projected light so as to prevent glare and to give adequate illumination where it is needed. Such specifications are worthless however if they are not enforced after they are drawn up. Therefore, to accomplish the ultimate object of both this Society and the Illuminating Engineering Society, the specifications as drawn up must be enforced by those who are in position to do so. These are the motor-vehicle administrators in the several States.

The Illuminating Engineering Society's headlight work, which has been under way for about 6 years, has been carried on primarily to assist the various State administrators in bringing about the conditions that I mentioned, safe and adequate driving illumination. Over 50 per cent of the cars in the United States and Canada have been operating under the old Illuminating Engineering Society specifications. The Headlight Committee has gradually won the confidence of the various administrators in the several States and has their backing in stiffening up the requirements. Naturally, it was necessary when this work was started, to begin with something that equipment on the market at that time could meet, and then gradually stiffen up the requirements.

A number of States have modeled their laws after the specifications that have been prepared. An association has been organized of motor-vehicle administrators representing the New England States, New York, New Jersey, Pennsylvania and Maryland.

In addition to the mechanical specifications that are printed in this report, the Illuminating Engineering Society has made certain other recommendations, largely at the request of the State administrators. I refer to the provisions given in the I.E.S. report under the captions "Reports" and "Approval," as follows:

> The report of the test shall be rendered in duplicate to the State authority and shall be signed or initialed not only by the expert making the test, but also by an executive officer of the institution making the test
>
> No device shall be approved which does not pass the laboratory tests
>
> No device involving the use of a tilting reflector shall be approved unless it conforms with these specifications in its "tilted up" position and has an extreme tilting range of no more than 3 deg.

The Division takes exception to the provision last given above. It seems to me that it involves largely legal administration. In the Eastern States where the roads are crowded and not very mountainous, and there are not many curves, probably a headlight that can be tilted up is a dangerous thing. The administrators in those States certainly feel that it embarrasses them in enforcing their regulations. On the other hand, in the Middle West and in the Western States, in the mountainous districts, headlights that can be tilted up are certainly a good thing. It seems to me, therefore, that the subject-matter of the clause in question should be left to the different State administrators, as provided for in the next clause which reads:

> The State authority reserves the right to refuse approval of any device which in his opinion is liable for any reason to prove unsafe or unsatisfactory in practice. Among the defects that may cause rejection under the above rules are unnecessary loss of light due to absorption or diffusion; abnormal or unduly complicated adjustment; unstable or bad mechanical construction; unduly bright or dark areas or excessive contrast in the illuminated field; irregular or badly defined cut-off line

There is another clause, entitled Alterations in Design, that has been found in practice to assist very materially laboratories in making tests and administrators in recognizing equipment on the road as equipment that has or has not been approved by the State:

> In case the design or construction of any device is changed in any way that alters the characteristics by which it is ordinarily identified, a new name or type designation must be given and the device may then be submitted for approval on the same basis as a new device
>
> An alteration in the design of an approved device that does not affect its distinctive appearance, but is made for the purpose of improving its performance, or to correct for alterations made in the standards of construction of incandescent lamps or reflectors, may be allowed under the original approval of such device, provided due notice of such alteration is given to the State authority and verification tests satisfactory to the State authority are made, which show that the device as altered complies with these specifications.

I think that is a very good provision.

There is one more paragraph on Verification Tests, which reads:

> As a safeguard against deviations of the design and construction of an approved device from that on which approval was originally based, the State authority reserves the right to submit from time to time samples of approved devices to the testing authority for a verification of their performance, or to require the submission at suitable intervals of certified copies of reports of such verification tests satisfactory to the State authority made by a person or an organization fulfilling the requirements of a testing authority. In case copies of verification tests are not submitted as required, or the verification test shows failure of any device to conform to the specifications, the State authority may suspend or withdraw approval of such device

That seems to be a proper provision because in making lenses, for example, the molds wear and the lenses may change so much that they do not give the distribution of light that the original lenses that were submitted for approval gave.

I would like to move that that portion of the Illuminating Engineering Society's report which is presented in the Lighting Division's report be approved as S.A.E. Recommended Practice, and that the Standards Committee endorse that portion of the Illuminating Engineering Society's report entitled "Reports" and "Approval," with the exception of Par. 2 under "Approval," which reads: "No device involving the use of a tilting reflector shall be approved unless it conforms with these specifications in its tilted up position and has an extreme tilting range of no more than 3 deg."; and also adding a clause giving one State authority to accept tests conducted in another State.

C. A. MICHEL:—With regard to stating specifically that one State may accept another State's tests for the approval of a device, I would say that at the present time an outlay of approximately $1,500 is required to get the approval of a device throughout the Country; involving a number of hand-made samples in some instances, because a manufacturer must have his device approved before he can go into regular production. The testing

fees mount up so rapidly that it is imperative to have a clause of that kind to encourage making improvements in devices.

PRESIDENT BACHMAN:—I think that the motion is too complex. I would prefer to vote on parts into which it might be divided. We have two things under consideration: the adoption of specifications of numerical values, etc., that properly can be interpreted as S.A.E. Standards of a technical nature; and the endorsement by the Society of acceptance tests and other matters which in my estimation partake somewhat of politics and on which the Society would do well not to commit itself. I am not at all sure that the constitutional restrictions of the Society are effective in this particular case. I am inclined to believe it is dangerously close to the borderline. While I appreciate thoroughly what Mr. Porter and Mr. Michel have said and some of the State administrators have taken up this question with me, I am very strongly of the opinion that we would go outside of our legitimate field if we endeavored to draft laws.

I do not underestimate the conditions under which the States are operating in their endeavor to equalize the use of the highways among all people and to safeguard the rights of the citizens who use the highways. I think that in approving technical values upon which tests may be based we will go as far as we can. To go beyond that, attempting to outline methods of legal enforcement or of administration, is outside of our province.

MR. CRANE:—I had more or less to do with this headlight illumination question about 2 years ago, when the attempt to assign numerical values over a considerable field of illumination was first made. This was evidently a move in the right direction. On the other hand, it has become plainer every day that we must compromise. We know that there is no really good driving illumination that is not glaring in some positions. We have taken the best averages that we could, illuminating as strongly as possible where it can be done safely, and toning the illumination down where it is apt to reach the eye of an approaching car driver.

I think that the Society should stop at what it considers to be the best compromise covering these conditions. The Society should not take action in the Standards Committee on the question of adjustable or non-adjustable headlights. That is a question partly of engineering practice and partly of state administration. We all know that the standard head-lamp in the standard adjusted and fixed position does not give a thoroughly good driving light; and I am sure that we do not want to take the position that the driver should be permitted to vary his head-lamps from a normal condition when he wants to. I have seen too much in my own driving of the misuse of apparatus permitting this, although much of it is unintentional. The driver has nothing to show him when he is within the law and when he is not, and he puts his device into a pretty good position as far as he is concerned from a driving point of view, and lets it go at that.

With regard to the Division's report, I would like to move, Mr. Porter's motion not having been seconded, the adoption of the specifications as outlined covering only safe illumination, and eliminating all extraneous matters pertaining to application, methods of administration or approval of adjustable head-lamps as against non-adjustable head-lamps.

CHAIRMAN JOHNSTON:—Mr. Crane's motion is in effect to adopt Part 1 of the Lighting Division's Report on Head-Lamp Illumination.

[The motion was seconded by H. W. Alden.]

OSCAR F. OSTBY:—The acetylene interests were drawn into this headlighting matter because the specifications did not include all types of head-lamp. The tentative rules and regulations, as laid down by the Illuminating Engineering Society, as you probably know, were sent to the various States and used as a basis for State legislation. All of the States in which we have had the privilege of presenting our side of the story have made exceptions to the rules and regulations as laid down by the Illuminating Engineering Society.

I believe that the test of any specification is its practical application under operating conditions, and that rules and regulations that do not work out under practical operating conditions should not be endorsed by such a prominent organization as the Society of Automotive Engineers. I will refer to the State of Massachusetts, the last State that passed a headlight law under the Illuminating Engineering Society's specifications. We did not know that the proposition had gone as far as it had. I went to Boston and found that a temporary list of approved devices was about to be issued under the Massachusetts law, which was based absolutely on the Illuminating Engineering Society's specifications. I called on Frank Goodwin, the registrar of motor vehicles, told him that I represented the acetylene interests, and presented my case. Among other things I called his attention to the fact that there are as many acetylene head-lamps used on motor trucks today as there were on passenger cars in 1910 and 1911, and that most of the motor trucks operating in Massachusetts could not possibly comply with the proposed regulations.

As the result of a pretty thorough investigation, in which the motor-truck people in Massachusetts were just as much interested as we were, Mr. Goodwin's office made an exception for acetylene head-lamps and issued a temporary approval, which has since been made permanent; for acetylene head-lamps equipped with a 5/8-ft. burner, 6-in. spherical mirror, and a clear glass front, which produces the necessary 2100 cp. called for by the law. In other words, so-called non-glare lenses are not required for acetylene head-lamps because it was found under practical conditions that they are entirely unnecessary.

Any specification or law that does not contemplate practical operating conditions does not belong in statute books. We have had to contend with this proposition everywhere that it has appeared because it has been misconstrued by State legislators to include acetylene head-lamp illumination. State legislators are not well posted on light, electric lighting and road illumination, and for that reason these specifications have been accepted as head-lamp regulations, with no classification as to the nature of the source of light.

MR. R. S. BURNETT:—This was one of the points considered by the Lighting Division in preparing its report, which is presented for approval as applying only to motor-vehicle electric head-lamps, and, if adopted by the Society, will be so published.

CHAIRMAN JOHNSTON:—We have before us a motion, duly seconded, to approve Part 1 of the Lighting Division's Report on Head-Lamp Illumination as submitted.

[The motion was carried.]

MR. MICHEL:—Part 2 of the Division's report is not a part of the Illuminating Engineering Society's report. Part 1 gives values for legal minimum illumination and maximum glare, while Part 2 gives values for desirable driving light. Head-lamps which comply with the specifications given in Part 2 will comply with the tests specified in Part 1.

CHAIRMAN JOHNSTON:—Is there any further discussion with reference to Part 2 of this report?

MR. PORTER:—I move that it be approved.

[The motion was duly seconded and carried.]

(13) HEAD-LAMPS

The Lighting Division recommends that the present S.A.E. Recommended Practice for Head-lamps, page B1 of the S.A.E. HANDBOOK, be revised as indicated on p. 480.

(14) MOTORBOAT LIGHTING VOLTAGES

The Lighting Division recommends that the accompanying report submitted by the Subdivision be adopted as S. A. E. Recommended Practice.

(1) For motorboats and small cruisers having combined starting and lighting equipment, it is recommended that nominal 6-volt (6 to 8 volts) or nominal 12-volt (12 to 16 volts) systems be used

(2) For larger cruisers having separate lighting equipment, it is recommended that the 32 or 110-volt system be used

NON-FERROUS METALS DIVISION REPORT

(15) ALUMINUM ALLOYS

The Non-Ferrous Metals Division recommends that Specification No. 34 given on p. 481 be adopted as S.A.E. Standard.

THE DISCUSSION

CHARLES PACK:—This specification was written at the request particularly of D. L. Gallup, formerly of the Nordyke & Marmon Co. Dr. Zay Jeffries, who submitted the specification, stated that about 95 per cent of the aluminum pistons used in automotive construction are made to it. A number of other alloys were submitted but I believe that the specification given surely represents major practice in the automotive industry.

(16) WHITE BEARING METALS

The Division recommends the adoption of the extended specifications for White Bearing Metals Nos. 11, 12, 13 and 14 given on p. 484.

[Specification No. 10 on p. 484 was approved as printed except that the tin-content was changed to 90.75 per cent minimum for ingots and 90.00 per cent minimum for cast products.

Specification No. 11 on p. 484 was approved as printed, except that the tin-content was changed to 86.00 per cent minimum for cast products and 87.25 per cent minimum for ingots.

Specification No. 12 was approved after adding a tin-content of 59.50 per cent minimum for cast products and 60.00 per cent minimum for ingots and changing the lead-content for cast products to 26.00 per cent maximum and for ingots to 25.25 per cent maximum.

Specification No. 13 was a misprint in THE JOURNAL on p. 484 and was approved as follows:

	Cast Products	Ingots
Tin	4.50 to 5.50	4.75 to 5.25
Antimony	9.25 to 10.75	9.75 to 10.25
Lead, Max.	86.00	85.50
Copper, Max.	0.50	0.50

Specification No. 14 was approved as printed on p. 484, except that the lead-content was changed to 76.00 per cent maximum for cast products and 75.25 per cent maximum for ingots.]

THE DISCUSSION

MR. PACK:—I would like to point out that the recommendations here given are somewhat of a departure from the usual methods of specifying alloys. I would also like to mention the fact that we have only extended and not changed the present S.A.E. Specifications Nos. 10, 11, 12, 13 and 14, for babbitts. We found in practice that automotive plants generally purchase ingots or finished bearings. The present standards cover finished products very well, but in purchasing ingot metals for the purpose of fabricating bearings it is not possible to maintain the S.A.E. Standard in the finished product. For example, in the case of S.A.E. Babbitt No. 11 the producer of the ingot has a tolerance of plus or minus ½ per cent in the tin-content. It is customary for the ingot manufacturer to take all the leeway he can on the tin-content, it being the most expensive element in the alloy. The possibilities are that he will produce an ingot containing 87.1 per cent of tin, with the result that the automotive manufacturer who uses that ingot for producing bearings will have only 0.1-per cent leeway before he gets below the specified tin-content.

It was therefore thought advisable, and it is in accord with the practice in some of the larger automotive plants, to establish another set of specifications for ingots, allowing the present specifications for babbitts to stand for finished products or, as we call them, cast products. It will be noticed that in every case we have kept within the mean of the tolerances permitted in the present specifications, but holding the limits much closer on the ingot, which is also in accordance with good practice.

After these specifications were originally adopted, the Division felt that the tin should not be specified with maximum and minimum limits, but that a minimum only should be established.

(17) WROUGHT NON-FERROUS ALLOYS

The Non-Ferrous Metals Division recommends that Specifications Nos. 77, 78, 82 and 83 given on pp. 482 and 483 be adopted as S.A.E. Standard.

[The caption for Specification No. 77 (see p. 482) was changed from "Phosphor Brass Strip for Flat Springs" to "Phosphor Brass Strip." Specification No. 78 was approved as printed on p. 482 and Specification No. 82 was approved as printed on p. 483 after changing the caption from "Brass Wire for Brazing" to "Brass Wire" and adding the words "brazing and" before "torch welding" under General Information White Bearing Metals.]

THE DISCUSSION

MR. PACK:—In drawing up Specification No. 77 for phosphor-bronze strip for flat springs the Division tried to adhere as closely as possible to the American Society for Testing Materials specifications, which has been the practice in the past. It was felt however that the American Society for Testing Materials specifications should be abridged somewhat so as to bring them more into conformity with the general practice of this Society. Specification No. 78 also was drafted by Dr. Jeffries and carefully discussed at the meeting held in May. Here also American Society for Testing Materials practice was followed, but omitting some unnecessary features from the Society of Automotive Engineers standpoint. I believe that these specifications represent the best practice in the aluminum-sheet industry.

The Society has adopted a brazing wire, I believe, by previous action. The Division now submits a specification for wire for brazing, given in Specification No. 82. Under General Information, it is stated that wire in accordance with this specification is suitable for torch welding.

MR. MYERS:—I offer the same objection in this case as in that of the steel wire specifications; the purpose for which this wire is intended is specified.

MR. PACK:—The purpose for which this wire is intended also appears under General Information. If you will refer back to the piston-alloy report, you will find a note that

This alloy cast in permanent molds is used principally for pistons, camshaft bearings, valve-tappet guides and other parts where high hardness and good bearing qualities are essential.

The Division felt that this is a similar type of specification and I believe that this particular alloy is well recognized as a brazing metal only.

MR. MYERS:—I have no objection to that, but I think that the caption referring to brazing ought to be removed.

MR. PACK:—I think this is largely a matter of policy and if it is the policy of the Standards Committee to omit such references in captions, I see no objection to removing the words "for brazing" and having the heading read "Brass Wire."

C. M. MANLY:—It seems to me that we are discussing a number of technicalities. If we print a lot of specifications and almost conceal what they are meant for, it looks as though we are afraid of what we are doing. If this brass wire is only good for brazing, why classify it as a general composition, and state somewhere in a footnote what it is really intended for, and make the getting of information from the S.A.E. HANDBOOK all the more difficult? We ought to stand on our feet in these matters and state what we mean.

In connection with "Phosphor Strip for Flat Springs," why flat springs? But if that is what the specification is intended for, let us say so. I did not raise any objection in that case because it seemed to me the Division has said what it meant. It seems to me the Division has said what it means regarding wire for brazing. I do not believe this is in any way breaking down our system in the Society's standardization work. We are, of course, trying to keep away from standardizing design and placing too close limitations through standard specifications; but on the other hand, if we are going to render the greatest benefit in this work, let us say what we mean.

MR. PACK:—These specifications, particularly the one for brass wire for brazing, were the direct result of requests from automotive manufacturers, including one for a wire suitable for brazing that is considered standard practice in the brass mills. Mr. Bassett, Mr. Price, and Mr. Webster, members of the Sub-Division, recommended this wire as especially suited for brazing. It may possibly have other uses, but the object of the Division in preparing this specification was to recommend a good brazing wire.

MR. CRANE:—The fact that under General Information it is stated that "This is also suitable for torch welding," indicates that it is not exclusively a brazing wire. I like to see the S.A.E. HANDBOOK of uniform style. There ought to be uniformity in presenting the information. In this case I would like to see the specification presented as for "Brass Wire," and in the General Information a note that "Wire in accordance with these specifications is suitable for brazing or torch welding."

MR. PACK:—Before this change is acted upon, would it not be well to reconsider the approval of Specification No. 77? We seem to have passed over that entirely without having had the same criticism as has been had on the Brass Wire for Brazing.

MR. CRANE:—We should use good form in presenting all such information in the S.A.E. HANDBOOK, purely as a matter of style, and for ease of consultation. We have the same thing with regard to White Bearing Metals. In each case there is a specification number, a general name for the metal and general information as to what it is useful for. I think that is an excellent form to follow.

PRESIDENT BACHMAN:—My thought in the matter is that this is largely a matter of captions. The value of the specifications is not lost by omitting reference to their purposes from the titles and putting it in notes under General Information. It is only an indication of how we want the information presented.

MR. PACK:—I believe President Bachman's point is a good one. It is a matter of general policy rather than of particular specifications.

CHAIRMAN JOHNSTON:—It seems to me that the Council could rule on some phraseology for all such cases, if it is permissible to pass these standards here and revise the phraseology later.

PRESIDENT BACHMAN:—I think that a general motion indicating the sense of the meeting as an instruction to the Standards Department would settle the whole matter. I move that the Standards Department be instructed to withhold from the caption and the body of these proposed standards any direct reference to the usage of the material, and that all such information be embodied in a footnote, in the form of General Information.

[The motion was seconded by Mr. Myers and unanimously carried.]

PARTS AND FITTINGS DIVISION REPORT

(18) BALL STUDS

The Parts and Fittings Division recommends that the dimensions for ball studs specified on p. 487 be adopted as S.A.E. Recommended Practice.

THE DISCUSSION

F. G. WHITTINGTON:—After a considerable amount of work this subject was presented at the Summer Meeting in 1921, at which time criticisms were made indicating that further work should be done. It was therefore referred back to the Division for further consideration. These criticisms proved to be very constructive and by serious consideration of them the Division has formulated what is believed to be a satisfactory recommendation. In augmenting the report of 1 year ago, we have extended the sizes to include smaller and larger ones than the five in general commercial use.

One criticism at the meeting a year ago regarding the ball size indicated the advisability of making the ball smaller than the normal size for two reasons, lubrication and manufacture. A tangential condition between the seat and the ball adds very materially to the lubrication factor, and obtaining this condition by both the ball and the seat being spherical, but of different sizes, avoids a bad machining job to provide a tangential effect by other means. Another good reason for undersize balls is that they can be made from standard instead of oversize stock, and standard drills and reamers can be used for the sockets.

Accordingly the Division has incorporated the undersize balls in the recommendation, varying the amount of undersize with the size of ball in three stages:

1/64 in. undersize on ½ to ⅞-in. sizes inclusive
1/32 in. undersize on 1 to 1¾-in. sizes inclusive
1/16 in. undersize on 2 to 3-in. sizes inclusive

Another criticism in 1921 was as to the amount of angular motion due to the shape of the neck. The dimensions recommended are such that the theoretical strengths of all the studs are proportional to their ball surfaces and they allow angular movement of the drag-links of 10 deg. in either direction.

The taper shank is of such length as to give a grip on the forging about equal to the large diameter of the shank and has a taper of 1½ in. per ft.

The thread is arranged to provide space for the lead of the chasers to run out. This space, equal to three

threads, is one regarding which there has been much misunderstanding. It is not an allowance for drawing the stud tight in its seat, which must be additional and sufficient to cover unavoidable manufacturing variations in the diameter of the shank and reamed hole, which will total from plus 0.003 to plus 0.005 in. The required allowance for draw is from 3/64 to 5/64 in. This deals with studs with good screw-machine finish, hardened and brushed to remove scale. Studs finished by grinding would, of course, have allowances much less than the figures given.

The nut to be used is a slotted nut of S.A.E. Standard dimensions, not a castled nut.

There is also shown on this stud a straight diameter extending from the large end of the taper toward the ball, to provide better fitting surface for dust-boots.

(19) Lock-Washers

The Parts and Fittings Division recommends that the present S.A.E. Standard, p. C5 of the S.A.E. Handbook, be extended to include the temper and toughness tests given on p. 524.

(20) Passenger-Car Front Bumpers

The Division recommends that the passenger-car front-bumper mounting shown in the illustration on p. 485 be adopted as an extension of the present S.A.E. Recommended Practice for Passenger-Car Front Bumpers, p. C55 of the S.A.E. Handbook.

THE DISCUSSION

Mr. Whittington:—I wish to augment the printed report by saying that the Division fully realizes there are some cars on which the mountings cannot be made as shown in the recommendations, but such a condition cannot be avoided. The fact that 90 per cent of the car builders approved such a recommendation clearly shows there is no reason why this recommendation cannot be approved. In connection with the present standard, the Division decided in working on the bumper mountings at this time to deal strictly with the mounting and avoid any discussion of the existing standard as to bumper dimensions. We realize that general bumper practice now makes the present standard of little value and we intend later, after the mounting feature is finished, to consider revision of the present standard, and not to confuse the work at this time.

(21) Plain Steel Washers

The Parts and Fittings Division recommends for adoption as S.A.E. Standard the table of plain steel washers given on p. 485.

(22) Serrated Shaft Fittings

The Parts and Fittings Division recommends that the taper and straight shaft-serrations shown in the illustrations and tables on pp. 488 and 489 be adopted as S.A.E. Recommended Practice for Serrated Shaft Fittings.

THE DISCUSSION

Mr. Whittington:—The work on serrated-shaft fittings has met with many delays, but has finally been brought to a form that the Division feels is sufficiently definite and complete to allow of its presentation to the Standards Committee for approval.

Considering first the straight serrated-shaft fittings, we found less tendency toward any standard among manufacturers than in the tapered fittings, but in drawing up our recommendations we have avoided conflicting with present practice as much as possible. The straight fittings are intended primarily for carrying strains in one direction. The selection of the number of serrations is important, as there must be enough to provide for ample adjustability but not so many as to cause manufacturing difficulties. Finding little uniformity in practice, the Division has selected 36 serrations on sizes 1/8 to 5/8 in. inclusive and 48 on 3/4 to 3 in. inclusive. This gives ample adjustability and affords easy indexing for machining.

With regard to tapered serrated-shaft fittings, we find a greater tendency toward uniformity and have accepted current practices as far as possible in our recommendations. Tapered-shaft fittings are intended for reversible strain loads, and are recommended in sizes from 1/2 to 3 in. inclusive; no sizes being specified below 1/2 in., as these are not used. The teeth on the taper fitting have practically uniform width of flat on top and bottom throughout their length. The number of serrations selected corresponds closely to present practice; 36 serrations for 1/2 to 1 3/4 in. inclusive, and 48 serrations on 2 to 3 in. inclusive. The taper is 3/4 in. per ft. on the outside diameter. The cutting angle is 1 deg. 37 min.

(23) Rod-Ends

The Parts and Fittings Division recommends that the present S.A.E. Standard for Rod-Ends, p. C8 of the S.A.E. Handbook, be extended by the addition of a 1/2-in. rod-end with a length of 3 in.

THE DISCUSSION

President Bachman:—It has always seemed to me that truck requirements have not been properly taken care of in considering these standards for rod-ends, clevis-pins and yoke-ends. Trucks equipped with solid tires and in which there is much vibration, offer a big field for a yoke-end having a larger pin than is required by any consideration of strength, both in diameter and in length of bearing, which will not wear so rapidly. That is one of the reasons I have never felt that this was a standard that we could use in our own product.

Mr. Whittington:—Is it your idea, Mr. Bachman, that we should have even heavier sizes than given in the heavy series?

President Bachman:—I think that the heavy series as listed here would probably have small usage on passenger cars, and for the reason I have indicated it is not satisfactory for trucks.

Mr. Burnett:—The Standards Department referred this subject to all the companies that we found were making these forgings and I believe they were all satisfied with the standards as they are, with this one change that is now recommended. There was a little difference of opinion regarding the 1/2-inch size, but apparently from the information we got from the forging manufacturers they have been furnishing and stocking these standard fittings for some time to the trade's satisfaction.

(24) Tank and Radiator Caps

The Parts and Fittings Division recommends that the present S.A.E. Recommended Practice for Tank and Radiator Caps, p. C58 of the S.A.E. Handbook, be extended by adding the note, "It is recommended that on passenger-cars, motor-trucks and tractors gasoline-tank filler-pipes have a minimum clear opening of 2 in. in diameter."

PASSENGER-CAR BODY DIVISION REPORT

(25) Top-Irons

The Passenger-Car Body Division recommends for S.A.E. Recommended Practice that the threaded-end of top-irons shall be 7/16 in.—14 U. S. S. with a 5/8-in. length of usable thread and a 1-in. length of stud.

SCREW THREADS DIVISION REPORT

(26) Screw Threads

The Screw Threads Division recommends that the

present S.A.E. Standard for Screw-Threads, p. C1 of the S.A.E. HANDBOOK, be extended to include the tables and definitions given on pp. 525 to 530 inclusive.

THE DISCUSSION

MR. ALDEN:—What success has been had in the manufacture of screws, bolts and nuts adhering to these tables?

G. S. CASE:—Very little work in this direction has been done because the tables as a whole have been adopted as standard only by the Army and the Navy. They have never been published in a form permitting the average manufacturer to use them. The intention is to present them so that manufacturers can use them. It is probable however that after they have been used somewhat, some changes will have to be made in them, but we do not know of any way to try them out except by putting them out. As a result of study, it is believed that they are in good condition to use at the present time.

The work of the Screw-Threads Division is different from that of other Divisions in that the automobile industry does not use much over 5 per cent of the bolts, nuts, screws and similar articles that are made. What we want to do is to arrive as nearly as possible at a standardization that can be used by all the industries. The great trouble has been that each industry has been standardizing from its own point of view. It has been the purpose of the National Screw Thread Commission, the Sectional Committee and the Screw-Threads Division to bring about uniform screw-thread standardization.

(27) SCREWS, BOLTS AND NUTS

The Screw Threads Division recommends that the explanatory text in the S.A.E. Standard for Screws, Bolts and Nuts, p. C2 of the S.A.E. HANDBOOK, be revised and extended to read as recommended on p. 530.

SPRINGS DIVISION REPORT

(28) FRAME BRACKETS FOR SPRINGS

The Springs Division recommends that the present S.A.E. Recommended Practice for Frame Brackets for Springs, p. H6 of the S.A.E. HANDBOOK, be cancelled.

(29) SPRING-EYE BUSHINGS

The Springs Division recommends that the present S.A.E. Recommended Practice for Spring-Eye Bushing and Bolt Tolerances be revised by the elimination of the bolt tolerances, the title of the Recommended Practice to be changed to "Spring-Eye Bushings."

STATIONARY-ENGINE DIVISION REPORT

(30) FLYWHEEL PULLEY LUGS

The Division recommends that the flywheel pulley lug dimensions given in the table on p. 484 be adopted as S.A.E. Recommended Practice.

UNACCEPTED RECOMMENDATIONS

ENGINE DIVISION

CRANKCASE DRAIN-PLUGS

The Engine Division recommends for S.A.E. Recommended Practice that crankcase drain-plugs shall have a minimum opening of ¾ in., be located at the lowest point of the oil-pan and be operable from under the engine hood.

[This subject, which was printed on p. 478, was referred back to the Division with the suggestion that it be discontinued inasmuch as it involves primarily a matter of design.]

THE DISCUSSION

MR. CRANE:—I do not want to be misunderstood in raising a question about this proposed recommended practice, but I think that the members ought to realize that in adopting this report we would be starting an entirely new kind of standardization, not of dimensions for the purpose of interchangeability, but a compendium of proper engineering practice.

No doubt this recommendation is excellent engineering practice. It might be that the Society would like to go much farther and eventually produce a standards book that would contain a lot of excellent advice for the designer, but I think we want to realize now what we are starting to do and decide whether we want to go through with it to the end.

MR. CHASE:—Is it proposed to use a ¾-in. or a ½-in. pipe plug? If not, how is this opening to be closed?

MR. CRANE:—I think that what Mr. Chase has said emphasizes the point that we are not attempting to standardize a dimension here. This recommendation is simply that any hole smaller than one of ¾-in. diameter is too small to drain the case in the shortest time, and that it ought to be as big as that or, as I have found, preferably it should be much bigger. The Division is not in a position to standardize the fittings or any other feature of this at the present time and the question is whether in view of that we should do anything about it at all. I only want to bring out the underlying principle of this kind of standardization.

CHAIRMAN JOHNSTON:—I think most engineers would agree with you on that subject.

A. L. CLAYDEN:—I agree entirely with Mr. Crane. At the same time, some of us might like to have a recommended standard size for oil-plugs, where these are used, specifying a diameter of ¾ in. and, if you like, a particular thread. If anybody improves on that, in the way of a device, well and good, but I think there might be a certain amount of value in having a standard oil-plug that at least would not be smaller than ¾ in.

R. J. BROEGE:—The Division found that the present oil-plugs are too small, and wanted to formulate a recommendation that would help to improve this part of the engines.

T. Y. OLSEN:—I think that the Division in studying this subject considered it more from the user's viewpoint. Possibly the members who use cars have had some difficulty in draining the oil out of the crankcase. We should provide some means for bettering this situation.

CHAIRMAN JOHNSTON:—I think we all will agree that anything that will induce designers to make these drainholes large enough and put them in a convenient place would be desirable.

MR. ANDREW:—There was a matter somewhat similar to this before the Electrical Equipment Division last year in connection with wires and cables. We felt that considerable could be done to indicate to manufacturers how large a cable should be used, but we did not think it was a fit subject for standardization. So we arranged for one of our members to present a paper on the subject. This largely accomplished the purpose.

PRESIDENT BACHMAN:—I am inclined to feel that Mr. Crane has raised a point that is worthy of our serious consideration, not by any means depreciating the work of the Division. The Division has tried to make a good recommendation, but the subject relates, I think, to invention and not standardization. Many of us would like to find a way of getting a hole big enough to drain the crankcase properly and have a device to this end operable from above the car, so that we would not have to put on a pair of gloves or a suit of overalls to do the job.

W. G. WALL:—I think Mr. Bachman has hit the idea exactly. Peculiar as it may sound, the crankcase drainplug is in the embryonic stage for the present. For a good many years we have used an ordinary plug. It is

not the function of the Standards Committee to do the designing for the entire automobile industry, but no doubt the industry will be very glad to improve the designs.

Therefore, I think that this matter should be left in abeyance and not even the size of the opening specified at present, as that would handicap some of the work that I know different engineers are doing, from which they will probably get some results during the next year.

Motorcycle Carbureter Flanges

[This subject was withdrawn as a report to be approved at the Standards Committee meeting because of criticisms that had been received, and was referred back to the Division for further study.]

NON-FERROUS METALS DIVISION REPORT

Specification No. 83, Soft or Annealed Copper Wire

[This specification, which was printed on p. 483, was referred back to the Non-Ferrous Metals Division for joint consideration with the Electrical Equipment Division inasmuch as the latter is now working on the preparation of specifications for magnet wire and it was considered desirable to have but one specification for this class of material.]

THE DISCUSSION

Mr. Andrew:—Is this specification intended to include magnet-wire?

Mr. Pack:—As stated in the General Information, it is intended mainly for electrical purposes.

Mr. Andrew:—The Society has a Sub-Division that is working on magnet-wire, in connection with the American Society for Testing Materials and also the American Engineering Standards Committee, and we hope to get a magnet-wire specification that will be unusually good. I feel that this specification goes only half-way and that it should be referred to the Electrical Equipment Division for joint consideration with the Non-Ferrous Metals Division later on.

Mr. Burnett:—Is this specification intended to cover annealed wire for both mechanical and electrical purposes, or will it be necessary to have separate specifications?

Mr. Pack:—The object was to cover both, but we eliminated some of the physical properties given in the American Society for Testing Materials handbook because we thought them burdensome. The main object of the specification was for all mechanical purposes in automotive practice, but I think one set of specifications covering both electrical and mechanical uses would be more desirable.

Mr. Manly:—I move that this matter be referred to the Council with the request that it be investigated as to whether it should be referred only to the Electrical Equipment Division or jointly to the two Divisions, to determine upon a suitable specification.

[The motion was seconded by Ernest Wooler, and carried.]

SCREW THREADS DIVISION

Gages and Gaging

The Screw Threads Division recommends that the statement submitted by Mr. Buckingham and printed on pp. 530-532 be approved by the Standards Committee for publication as general information only.

[This subject was referred back to the Division for the purpose of including more extensive consideration of gaging for lead error in screw-threads.]

THE DISCUSSION

Mr. Crane:—This report on Gages and Gaging is very creditable but it does not deal sufficiently with the importance of gaging for lead error. Manufacturers of nuts, bolts and screws have not talked much about these errors because they have great difficulty in controlling them. Yet the matter is very important. I would like to see this report modified by putting more emphasis on the question of gaging for correctness of lead.

SPRINGS DIVISION

Definitions

[This subject, which was printed on p. 532, was referred back to the Springs Division for further consideration in line with the discussion at the Standards Committee meeting.]

THE DISCUSSION

Mr. Crane:—In this report the definition of Free Height does not appeal very much to me, as a definition. It is given as

> the distance from a line through the center of the spring-eyes to the face of the axle-pad, when the spring is in a free, unloaded position.

About half of the springs used are underslung, the others being overhung. Under this definition the same spring could have two free heights.

The definition for Load Height as

> the distance from a line through the center of the spring-eyes to the face of the axle-pad, when the spring is deflected to rated load

is likewise ambiguous, referring to the axle-pad that may be above or below the spring.

W. C. Keys:—A line drawn through the center of the spring-eyes will not in many cases be parallel to the face of the spring-seat, and the distance of free height should be measured perpendicularly to the line through the eyes to the center of the spring-seat, preferably at the outside of the spring. This definition, as Mr. Crane states, leaves it optional to measure to either the inside or the outside of the spring, depending on whether it is overslung or underslung.

In connection with the definitions of Free Height and Load Height the internal or interleaf friction in the spring is not taken into account. I have laid leaf-springs on the floor on their side and tapped them to take out all inherent tendency to flex one way or the other, and get one height. Then they were put on a spring testing-machine, compressed and the load taken off slowly. Upon again measuring the free height there would be perhaps 3/8-in. difference from the first measurement. I think this point should be considered in the definition.

Walter W. Wells:—The definition of flexibility disagrees with my ideas of what it should be. According to this definition, a spring that requires 600 lb. to deflect it 1 in. would be expressed as having a flexibility of 600 lb. I would expect the number to indicate its stiffness rather than its flexibility.

PROGRESS REPORTS

In addition to the various recommendations that were discussed and voted on at the meeting of the Standards Committee, three progress reports were presented. These reports with such discussion as followed their submission are printed below.

The progress report of the Parts and Fittings Division on Brake Linings was not presented on account of lack of time. The progress that is being made on this subject will be published in future issues of The Journal.

BALL AND ROLLER BEARINGS DIVISION

Metric Thrust Ball-Bearings

This report is submitted as one of progress on the

standardization of the Metric Series of Ball Thrust-Bearings. The Subdivision has been working toward the complete standardization of light, medium and heavy series in each of the following types:

Single-Direction Thrust—Flat Type
Single-Direction Thrust—Self-Aligning Type
Double-Direction Thrust—Plain Type
Double-Direction Thrust—Self-Aligning Type

making a total of 12 series of metric sizes.

In beginning this work difficulty was encountered in that dimensions of thickness and outside diameter of the bearings, as established by the several manufacturers, were not the same for corresponding bore diameters. These discrepancies have been reconciled by adding to or taking from dimensions of the individual manufacturers. For the most part, dimensions have been increased to meet the existing individual standards, thus working toward the strengthening of thrust-washer sections and eliminating the weaker parts in the several series.

In the self-aligning series there was considerable variation in the dimension of the corner radii, and in the thickness of the self-aligning seat. These differences have been eliminated and, in addition, specific dimensions giving the location for the center of the corner radii have been specified. This is important, particularly to the users of self-aligning ball-bearings, as frequently they desire to incorporate the self-aligning seat as an integral part of their mechanisms. In the past they have been at a loss to determine this center-point accurately.

In the proposed standards the present established dimensions of both foreign and American ball-bearing manufacturers have been included as a basis for undertaking this work, but in order to obtain consistent series in the double-direction type it has been found necessary to develop a new light-series in both the plain and the self-aligning constructions. These will correspond with the light-series single-direction type.

The proposed bearings as developed so far have the general approval of the Subdivision and it is hoped to complete the work and submit the recommendation to the Ball and Roller Bearings Division at an early date for its approval.

ENGINE AND PASSENGER CAR DIVISIONS

Engine and Frame Numbers

At the meeting of the Society's Special Committee on Automobile Insurance, the Insurance Committee of the National Automobile Chamber of Commerce and representatives of the Underwriters' Laboratories in September 1920, the problem of devising methods of numbering automobile engines and frames was discussed in connection with a schedule of automobile insurance hazards that was under consideration at that time. Several methods were suggested which were circularized among automobile and engine manufacturers.

It soon became evident that the problem would require special treatment and it was therefore assigned jointly to the Engine and Passenger Car Divisions, which have discussed it at their meetings and made samples of a number of methods that will be submitted to the Underwriters Laboratories for test to determine their effectiveness. The work has been given considerable publicity throughout the country for the purpose of obtaining ideas from as many sources as possible.

The underwriters have stipulated that whatever methods are recommended must be approved not later than Jan. 1, 1923, to have the reduced theft-insurance premiums effective.

THE DISCUSSION

R. S. Begg:—Perpetual motion has nothing on the subject of non-changeable numbers for frames and engines. The problem is to obtain a system of numbering such that the numbers cannot be changed or obliterated without any layman or policeman being able to detect this. For instance, if the numbers were on detachable plates and when a new plate was put on bearing a different number, the change could be detected easily; it would be considered that we had a successful system.

The location of the numbers is not of direct interest at the present time as we are interested primarily in getting some system that will be satisfactory to the Society and pass the Underwriters Laboratories test. If we can get such a system it will not be hard to find a suitable location for the numbers on the frames and the engines.

The object in studying this matter is to lessen the ease with which the means for identifying stolen cars can be obliterated, and to secure lower premiums on car theft-insurance. The possible amounts to be saved in theft insurance has not been investigated thoroughly. It is very difficult to determine any specific saving because theft-insurance premiums vary, depending on both the make of car and the locality. It will be feasible of course to get only a general average that will be a reasonable estimate. We have been informed, however, that for cars having either an engine or frame approved numbering method, there will be allowed a $7\frac{1}{2}$-per cent reduction in premium, whereas the total reduction will be 20-per cent if both are numbered according to methods approved by the Underwriters Laboratories.

We have received many letters suggesting all kinds of schemes and offering patented devices. Among the suggestions received are

(1) Casting numbers on the crankcase. Such a method would not, of course, apply to frames
(2) Concentric expanded discs
(3) A special alloy cast in the crankcase
(4) A checker-board system
(5) A small thin fin cast on the crankcase on which numbers could be stamped. The numbered fin would be broken off by the dealer and held for the purchaser
(6) Raised number-plates welded to the frame
(7) Armor-plate number-plates riveted on, using two or more rivets which enter into the frame construction
(8) Fairly large numbers embossed deeply into the frame
(9) Small holes punched or drilled into the frames at places where the frames would not be weakened
(10) Numbers inset under a glass cover in a spring-bracket or other accessible casting or forging in the frame assembly. This system is applicable to engines also
(11) A thin plate, with the number embossed on it, held in a frame or riveted or blind-screwed to the car frame
(12) A thumb-print system
(13) An armored plate riveted into position, with special seals placed over the rivets. [See page 211 for an illustration of this method]
(14) Case-hardened plugs or plates bearing the numbers fastened into the frame with blind screws or some other such device. This method is applicable also to engines
(15) A plate of bakelite or similar material over the numbers, fastened behind the frame channel as in the preceding suggestion. This is applicable to engines also

(16) A series of embossed nickel plates set in the frame or engine in such a way that they could not be removed without tearing down the engine or frame. The reason for advocating nickel is that it is not easily affected by the acetylene torch. [See this page for an illustration of this method]

Several suggestions have been embodied in samples submitted for inspection today. These will be sent to the Underwriters' Laboratories for their experimentation. Among the samples are

(1) A front spring-hanger having a forged depression into which is set a series of thin nickel-plates with perforated numbers. The plates are held in position by the spring-hanger rivets, the numbers showing through a slot cut in the channel of the side-rail. If necessary, these can be welded-in. [See this page for an illustration of this method]

(2) A heat-treated nickel-steel plug with numbers stamped on circular serrations, held in the side-rail by a heat-treated nut, with a set-screw locking it and the head broken off. [See this page for an illustration of this method]

(3) Same as No. 2, except that it is made of tool-steel

(4) Numbers in a bakelite cap molded on a heat-treated nickel-steel plug held in the side-rail by a heat-treated nut locked on by a set-screw with the head broken off. [See this page for an illustration of this method]

(5) Circular serrations cut in the side-rail by the frame manufacturer; on which the car builder would stamp the serial number. [See this page for an illustration of this method]

(6) Sine-waves pressed into the side-rail by the frame manufacturer; on which the car builder would stamp the serial number. [See this page for an illustration of this method]

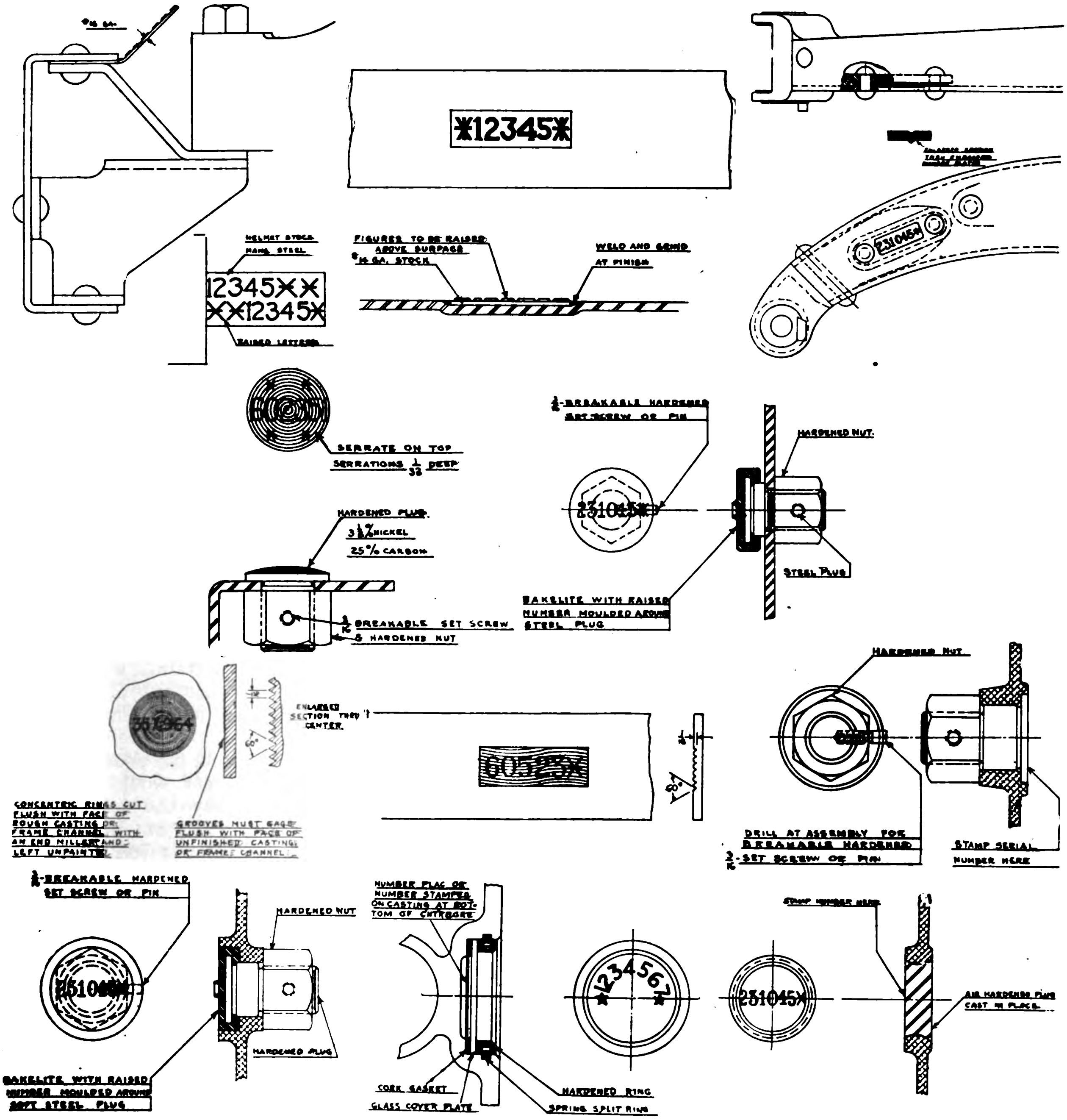


FIG. 2—SOME OF THE METHODS PROPOSED FOR NUMBERING ENGINES AND FRAMES

(7) A tool-steel plug held in the wall of the crankcase by a heat-treated nut locked with a set-screw with the head broken off. [See page 211 for an illustration of this method]

(8) The same as No. 7 except that it is made of nickel-steel heat-treated

(9) Numbers in a bakelite cap molded onto a heat-treated plug held in the crankcase by a heat-treated nut, locked on by a set-screw with the head broken off. [See page 211 for an illustration of this method]

(10) Numbers stamped on the crankcase wall and a glass placed over the numbers. The glass rests on a cork gasket and is held in place by a snap-ring of liberal size that locks in a groove cut in the case, a steel ring pressed flush into place concealing the snap-ring. [See page 211 for an illustration of this method]

(11) A heat-treated tool-steel plug cast in the crankcase. A chill is put across the plug so the heat in casting will not soften it. [See page 211 for an illustration of this method]

(12) Nickel numbers cast in an aluminum crankcase, the numbers being high enough to go through the wall of the casting and project on the outside about ⅛ in.

(13) This is the same as No. 12 except that it is an iron casting. It would be very difficult to destroy these numbers without cutting out a section of the crankcase

J. B. FISHER:—Last winter we made up some sample plates representing a crankcase casting, on which were cast two parallel ribs about 3 in. apart with the numbers stamped deeply in between with a 5-lb. hammer. The plates were sent to the Underwriters Laboratories for test, but in about 2 hr. they returned them, with the original numbers obliterated and new numbers stamped in. A description of these plates and the method of numbering was printed in THE JOURNAL last December and discussed at the Standards Committee meeting in January.

I believe that we cannot get up a system that cannot be beaten, but we can devise one that will make it a difficult and lengthy process for a thief to change the numbers. One such system, which may be about as good as any, is to stamp a number in 10, 20 or even 30 places on the frame and the engine. While this may seem foolish, I think it is about as practical a suggestion as we have received. Another point in this connection, which was mentioned by the Underwriters Laboratories, is that automobile thieves will seldom if ever attempt to change all such numbers. If one number which has not been tampered with can be located upon inspection of the suspected car, it is a simple matter to get a complete record of the car from the manufacturer, and a positive identification.

I prepared a few samples some of which we succeeded in changing so easily that it was hardly worth while to consider them. One consisted of a small piece of brass cast in the crankcase, with an impressed number somewhat similar to the raised safety-marking on checks. This numbering is really difficult to change but there is nothing to prevent the thief from soldering a new piece on quickly and easily.

There is a side to the question that has not been brought out. There are so many thousands of automobiles in use that it will be possible for a long time to come to remove a safety number entirely from a car so equipped and stamp in any other number in the simple methods now used. For instance, one of the methods used with a hardened-steel plug could be changed by knocking the plug off entirely, covering up the spot with a plate and stamping any other number somewhere else on the frame or the crankcase. A man suspected as the thief can claim that that is his number and a check on it would necessitate considerable time and effort. This is one of the phases of the subject we hope to overcome.

We should try to get the insurance underwriters to modify their requirements to the extent that they will approve a numbering system that is difficult to change and not require one that it is "impossible to change." We have experimented enough to know that almost anything can be altered, and if they will meet us half-way and accept a method or methods that will make it difficult to change the number, even though it can be changed easily in the laboratory, I think we will have accomplished something worth-while.

A. D. T. LIBBY:—Which one of the devices shown do the Divisions think the most of?

MR. FISHER:—That is an embarrassing question. It is largely a matter of relative value, as a few minutes of study will bring out the weakness of any one of the suggested methods. We do not know yet whether the Underwriters Laboratories can succeed in all these systems. It has been pretty successful so far in this respect in the case of the samples we have submitted.

W. C. KEYS:—Cannot the solution of this problem be left to the Underwriters?

MR. FISHER:—I do not know. I have not talked to them in that connection.

R. S. BURNETT:—This problem was first discussed at a meeting of the Society Special Committee on Automobile Insurance Schedules, the Insurance Committee of the National Automobile Chamber of Commerce and representatives of the Underwriters Laboratories in September 1920. We are relying on the Underwriters Laboratories to prove out what we devise. The Underwriters Conference became interested and intimated that if a numbering system for either engines or frames that would be proof against changing were devised, the reduction in theft-insurance premiums that has been specified here today would be allowed. The Underwriters Conference will not act in this connection upon anything that has not had the approval of the Underwriters Laboratories.

C. M. MANLY:—I would like to ask if the underwriters agreed that they would not raise the rates just before they make that reduction?

MR. KEYS:—All these different samples we have seen are very clever and interesting, but who is going to pay the bill? The manufacturer certainly will object, especially if he has a large production. If a small additional cost will be applied against the manufacturer, it will simply mean that the thing will not be adopted by him. But, as has been suggested, the underwriters might recommend something the owner would gladly comply with, and at his own expense have the numbering system applied to his car, and thus reduce his insurance.

MR. LIBBY:—If a car has an approved numbering system, the salesmen will use as a talking-point the fact that the buyer will get 7½ or 20-per cent reduction of his theft-insurance premium, as the case may be.

MR. BURNETT:—The first problem of the Divisions is to find some method that looks possible and feasible. The next problem is to reduce its cost to the lowest possible amount. We may not be able to get that combination but it is the aim of the Divisions to do so.

W. W. WELLS:—Has the method been tried of using secret numbers throughout the chassis, covered up so they could not be seen and their location would be known to only the manufacturer?

ENGINE CRANKCASE OILS

Tentative specifications, June 20, 1922

General.—These specifications cover grades of petroleum oil for the lubrication of internal-combustion engines, except aircraft, and are not recommended for the lubrication of turbines.

Only refined petroleum oils without admixture of fatty oils, resins, soaps or other compounds not derived from crude petroleum will be considered.

Specification No.*	Flash-Point, Min.	Fire-Point, Min.	VISCOSITY, SAYBOLT SEC. 100 Deg. Fahr.		VISCOSITY, SAYBOLT SEC. 210 Deg. Fahr.		Color (NPA) Darkest color allowed on mixture of oil and 50 per cent kerosene	Pour Test, Max.	Acidity, Mg. KOH per Gram, Max.	Conradson Carbon Residue, Per Cent, Max.	Corrosion Test
			Min.	Max.	Min.	Max.					
20	325	365	180	220	42	....	5	35	0.15	0.20	Required for all grades
020	325	365	180	220	42	....	5	0	0.15	0.20	
30	335	380	270	330	44	....	5	40	0.15	0.30	
030	335	380	270	330	44	....	5	0	0.15	0.30	
40	345	390	360	440	46	....	5	45	0.15	0.40	
50	355	400	450	575	50	....	6	50	0.15	0.60	
60	360	.....	.....	.....	55	65	....	55	0.15	0.80	
80	380	.....	.....	.....	75	85	....	55	0.15	1.50	
95	390	.....	.....	.....	90	100	....	55	0.15	1.75	
115	400	.....	.....	.....	110	120	....	60	0.15	2.00	

*For Specifications Nos. 20 to 50 inclusive, the numbers indicate the first two figures of the average Saybolt viscosity in seconds at 100 deg. fahr. of the grades indicated. The first cipher in Specifications Nos. 020 and 030 indicates that the pour-test value of these two grades is zero. Nos. 60 to 115 inclusive indicate the average Saybolt viscosity in seconds for these four grades at 210 deg. fahr.

Corrosion Test.—The following corrosion test shall not cause discoloration of copper strip. Place a clean piece of mechanically polished pure strip copper about ½ in. wide and 3 in. long, and 10 cc. of the oil to be tested, in a clean test-tube. Close the tube with a vented stopper and hold for 3 hr. at 212 deg. fahr. Rinse the copper strip with sulphur-free acetone and compare it with a similar strip of freshly polished copper.

Mr. Begg:—I think there is no doubt that the people who steal cars would soon find such numbers. The suggestion was made that we have a thief on the Divisions to test out these devices. The people who are stealing cars are pretty well organized and are prepared to steal almost any car they want.

Mr. Burnett:—One of the main objects of such a numbering system is to have the number in plain view where a constable or policeman can see it. Covering the numbers up would, I think, probably defeat the whole project because no policeman or constable would be permitted to tear a man's car partly apart to inspect the numbers.

Chairman Johnston:—To devise a system that will be satisfactory to the insurance companies and the car builders is a difficult task. On the other hand, if the insurance rates can be reduced 20 per cent, the attempt is certainly worth while from the economic standpoint.

Mr. Begg:—If anybody has further suggestions to offer, he should submit them in writing to the Society offices in New York City. We will make up samples for testing. The assistance of all the members is solicited. It will be necessary to have such suggestions promptly, however, because Jan. 1, 1923, is the time limit that has been set by the Underwriters Conference on the development, testing, approval and adoption of such a method of numbering engines and frames of automobiles.

LUBRICANTS DIVISION

Crankcase Lubricating Oil

One of the most important subjects before the Standards Committee is the establishing of definite specifications for different grades of engine crankcase oil used by the several groups of the automotive industry. The Lubricants Division was reorganized on an active basis in 1921 and work started on formulating practical specifications for crankcase oils, cup greases, transmission greases and other classes of lubricant. The work of the Federal Government, conducted by the Bureau of Mines in the Interdepartmental Committee on the Standardization of Petroleum Specifications, and the standard methods of testing adopted by the American Society for Testing Materials, have been considered carefully by the Division.

The accompanying table is the last revised specification proposed by the Division as the result of the discussion at White Sulphur Springs. The first specification was circularized, together with a questionnaire, by the Society in November 1921 and revised at a meeting of the Division in April 1922, as printed on p. 533 of the June issue of The Journal.

THE DISCUSSION

H. C. Mougey:—A meeting of members of the Lubricants Division, including H. C. Mougey, chairman, Sydney Bevin, P. J. Dasey, Dr. W. H. Herschel, K. G. Mackenzie, J. W. Stack and a number of guests representing producers and users, was held this morning (June 20). As a result of the viewpoints expressed by different engineers, some changes were made in the table of proposed specifications that was printed in the June issue of The Journal on p. 533. In Specifications Nos. 20 and 020 the minimum viscosity, Saybolt, at 210 deg. fahr., was changed to 42 instead of 43; in specifications Nos. 30 and 030, to 44 instead of 46; in No. 40, to 46 instead of 49; and in No. 50, to 50 instead of 52. The limit on the viscosity for No. 50 at 100 deg. was extended to read from 450 to 575 instead of from 450 to 550.

In Specification No. 80 the minimum flash-point was lowered to 380 and the maximum pour-test raised to 55. In Specification No. 95 the minimum flash-point was lowered to 390, the maximum pour-test raised to 55. The Conradson carbon-residue was raised to 1.75.

A new Specification, No. 115, was added, having a minimum flash-point of 400, nothing being specified under fire-point or viscosity at 100 deg. fahr.; the viscosity

at 210 deg. is 110 minimum and 120 maximum. Nothing is specified for color. The pour test is 60; acidity, 0.15; Conradson carbon-residue, 2.00.

The precipitation numbers for the entire series of oils were eliminated.

A column for corrosion tests, required on the entire series of oils, was added. The corrosion-test temperature, previously specified as 210 deg., was changed to 212 deg. fahr., to correspond with the corrosion test specified by the Interdepartmental Committee on the Standardization of Petroleum Products.

The revised specifications are presented in a tentative form and the Division is anxious to have as much help as possible from the members in making these specifications of the greatest possible use to the industry.

One question that has been raised is the drafting of these specifications so that they will conform with those prepared by the Interdepartmental Committee and with other specifications that are largely used, and a meeting is being planned to be held in July at which the Society, the Interdepartmental Committee and the American Petroleum Institute will make such progress as can be made along those lines.

Another question is that of designating names for the several grades of oil, such as "light," "medium" and "heavy," instead of the proposed series of numbers. Unfortunately, nobody has found enough names to go around. Suggestions in this connection will be appreciated by the Division.

CHAIRMAN JOHNSTON:—Is there any discussion of this progress report of the Lubricants Division?

P. J. DASEY:—If these recommendations are adopted as we have them now, how will we get the oil commercially on the market to conform to the specification numbers? I am not looking at the matter from the engineering point of view so much as from the standpoint of the everyday user of the product.

MR. MOUGEY:—That is the particular problem that is before us. We hope to have suggestions that will make for progress in this connection at the joint meeting of the Society, the Interdepartmental Committee and the American Petroleum Institute.

MR. DASEY:—We have 10 different numbers in the series. That would mean practically two light, two medium, two heavy, two extra heavy and two extra extra heavy grades, to include them all. I doubt whether many oil companies can meet these specifications.

MR. MOUGEY:—These specifications were drawn up in such a way that they can be met very generally.

C. W. STRATFORD:—In the discussion of these specifications this morning the only point to which I objected is that the acidity value of 0.15, given in the tabulation, cannot be met by any California oil at the present time. It would be rather a costly operation and would mean a fundamental change in processing methods to do it.

I represent the Associated Oil Co. of California and am familiar with what the other producers are able to do. We have produced oils with zero acidity. These oils give better results in engines than those of higher acidity. It seems that such specifications as these will ultimately force the oil companies to refine their oil so as to produce the zero acidity.

There is generally relatively little difference in the durability of the oils, as usually they are drained out after 500-miles' running. The difference is very marked in the operation of tractors, as at the end of 2 hr. a breakdown in the oils of high acidity is worse than in the case of oils of low acidity.

To produce the highest grade of heat-resisting oils from California crude, we manufacture a very heavy oil and blend it in certain proportions with the Eastern light-stock. The effect of adding Eastern light-stock seems to be protective. There is a sort of flaky carbon-deposit with this blend and yet the heat-resisting properties in regard to the crude from which it is made seem to be fully equal to that of the highest grade of paraffin-base or any other crude oil.

As to the grading, I think this is a serious specification as applied to California oils. The range of grades seems to me to be rather indefinite, each grade varying considerably. But there is a good reason for that variation. Lubricating oils made from different crudes vary widely. The viscosity drops very much more rapidly with an increase of temperature in the case of California oils than of straight-run oil from Pennsylvania. If we do not adopt specifications such as these, the only alternative would be to draft specifications for lubricating oils made from the Pennsylvania and from the Mid-Continent crudes; with perhaps a separate classification for the Gulf-Coast oils, and a distinctively separate classification for California oils.

DR. E. W. DEAN:—With regard to the organic acidity, the present limits seem to cover pretty well the Eastern and Mid-Continent oils. The acids found in the California oils are, according to information, present in sufficiently greater quantities to make a special refining process necessary to bring them within this specification. From that point of view it might be that the acidity test should be dispensed with, because it is doubtful whether the gain in the wearing quality of the oil is sufficient to make up for the additional expense in refining.

An automobile oil is not treated very well after it is put into an engine. For instance, some of the light oils have colors specified, but after one has been run in an engine a very few miles it cannot be determined whether it started at No. 1 or No. 67 color. The only value of color specification is that it is a certain index of the care in refining. In the same manner the organic acidity is an index. Eastern and Mid-Continent oils will give an acidity inside the specified limit, whereas the same degree of care in refining California oils results in an acidity several times greater. I think it doubtful whether it would be a benefit to the consumers to enforce the acidity limits on the Western oils.

With regard to the grades of oil it is rather hard to draft specifications on account of the differences in chemical constitution of oils, but I think the present figures are about as good a compromise as can be had. It is difficult to draft a general specification for lubricating oil that would be suitable for a given automobile to be produced from either an Eastern or a California crude.

W. H. HERSCHEL:—These specifications were reviewed rather thoroughly this morning and the changes that were decided upon make them less stringent. There should not be any objection to them from the oil-refiner's point of view.

Although there is a specification as to color test, I think that every oil-refiner will agree that there is not an oil on the market, or a probability that there will ever be, that will be ruled out on account of this, because the color is determined after dilution with 50 per cent of kerosene.

I think that sufficient emphasis has not been put upon the fact that the acidity test is probably the only one among all those specified that gives any indication of the probable durability of the oil. Emphasis should not be put on carelessness in refining. It is not a matter of concern to users of oil how much care is necessary to get oil

refined. The question is whether the product as finally offered for sale is of a serviceable nature. I do not think that the high acidity in California oils indicates that the California refiners are a bit more careless in refining their oils than the Eastern refiners are. It does indicate that the California crude is less stable chemically, and therefore will deteriorate in the refining and in the use. This question of acidity is one of great importance to the users of oils.

MR. STRATFORD:—What does acidity mean in terms of service? What do the emulsifying properties have to do with service, since after running non-emulsifying oil for 5 min. in an engine, there is a considerable emulsion of water? I would also like to ask Dr. Herschel whether the California oil treated with 0.1 per cent of soda, before it is treated with the acid, will have zero acidity, and whether the soda removes all of the unstable compounds?

DR. HERSCHEL:—To answer the question categorically, I should say, no. Mr. Stratford said this morning that this extra treatment was necessary to get the low acidity, and would cost only about 2 cents per gal. Therefore, this is not a very severe requirement on a refiner, or on the public which must bear the cost. Mr. Stratford said that the oil of lower acidity is more durable in tractor service; also, I believe that by mixing California oil with Pennsylvania light-stock a more durable oil could be produced. With regard to automobile-engine crankcases being drained at the end of 500-mile runs, this theoretical figure can be found in instruction books issued by automobile builders, but I question very seriously whether the majority of users of automobiles drain the crankcase at the end of every 500 miles. Therefore, I believe that durability of oil is as essential for an automobile as for a tractor.

MR. MANLY:—This discussion brings up the question of whether the test for acidity is made at the proper time. Mr. Stratford asks whether the addition of soda, bringing the oil within the specifications, suffices. It does so far as the test is concerned, but the important thing is whether it has made the oil any better for its intended use.

MR. STRATFORD:—This whole matter of acidity has come to us from the Navy specifications, the reason for its use in them being the limited tankage on-board ship, requiring the Navy to have practically the same class of oil for all purposes, that is, non-emulsifying oils. That does not apply at all to the lubrication of internal-combustion engines.

The divergence between California-oil acidity and paraffin-base oil acidity is approximately in the ratio of 1.5 to 0.1 in the case of a light oil. After several years' experience in engine-oil testing under service conditions, I believe that the only real test for an engine oil is actual use under service conditions. Driving at the usual touring speeds only about 10 or 20 per cent of the attainable horsepower of the average automobile is used. The temperature of all engine parts is low, and it would be foolish to use an oil with a 450-deg. flash-point. The result would be rapid formation of carbon on the piston-heads and in the combustion-chamber at the end of 2 or 3 weeks causing premature ignition and necessitating taking the engine down. If an oil of the correct loading-point range is used, that does not occur; the carbon formed is light and flaky, and is largely blown out with the exhaust.

The paraffin-base oil producers have learned that by burning the paraffin-splinters with Gulf-Coast, California or some Mid-Continent stock, the resultant engine oils come much nearer to meeting the operating conditions of the average automobile than the straight oil of Pennsylvania meets them.

Lubricating tractors is a matter of operating temperatures. If it is desired to operate at high temperatures, as in an airplane engine, an oil of very much higher loading-point must be used, and it becomes almost necessary to use the paraffin-base oil.

ATTENDANCE AT MEETING

The members of the Standards Committee and the Society and the guests in attendance were

Standards Committee Members

Azel Ames
F. W. Andrew
B. B. Bachman
R. S. Begg
S. Bevin
R. J. Broege
A. K. Brumbaugh
T. V. Buckwalter
R. S. Burnett
Clarence Carson
E. R. Carter, Jr.
G. S. Case
C. F. Clarkson
A. L. Clayden
J. Coapman
J. R. Coleman
H. M. Crane
L. A. Cummings
P. J. Dasey
G. W. Dunham
H. E. Figgie
J. B. Fisher
E. S. Fretz
D. E. Gamble
H. B. Garman
G. E. Goddard
W. S. Haggott
W. S. Harley
S. P. Hess
C. E. Heywood
M. C. Horine
F. G. Hughes
H. S. Jandus
E. A. Johnston
L. S. Keilholtz
W. C. Keys
C. H. Kindl
B. M. Leece
A. D. T. Libby
K. G. Mackenzie
C. M. Manly
C. A. Michel
H. C. Mougey
C. T. Myers
W. M. Newkirk
I. M. Noble
G. L. Norris
Leonard Ochtman, Jr.
Charles Oppe
Charles Pack
L. C. Porter
O. J. Rohde
A. J. Scaife
J. W. Stack
L. M. Stellmann
J. G. Vincent
W. G. Wall
E. W. Weaver
R. E. Wells
J. A. White
F. G. Whittington
Ernest Wooler

Society Members and Guests

H. W. Alden
V. G. Apple
C. E. Banta
David Beecroft
George Bell
O. C. Berry
R. E. Blossom
J. T. Caldwell
Herbert Chase
K. H. Condit
H. Copleston
J. T. Crawford
E. W. Dean
W. A. Dick
Ernest Dickey
H. C. Dickinson
J. B. Dillard
A. H. Ehle
F. H. Ford
G. W. Gilmer, Jr.
V. H. Gottschalk
Claude Greenhoe
A. J. Hall
G. A. Hancock
P. M. Heldt
W. H. Herschel
J. W. Hobbs
R. J. Hoffman
Russell Hoopes
G. E. Houser
R. M. Hudson
M. L. Hull
H. W. Jarrow
R. P. Lansing
T. J. Litle, Jr.
W. L. McGrath
Neil MacCoull
H. H. Magdsick
J. F. Marshall
W. H. Miller
F. E. Moskovics
W. C. Munson
A. L. Nelson
Harold Nutt
T. Y. Olsen
Gust Olson, Jr.
O. F. Ostby
B. S. Pfeiffer
N. B. Pope
H. R. Portugal
I. J. Reuter
H. E. Rice
H. J. Saladin
J. S. Schneider
R. H. Sculis
L. R. Smith
M. A. Smith
T. D. Smith
C. W. Stratford
G. E. Strohm
H. W. Sweet
P. S. Tice
E. E. Turner
W. C. Voss
T. A. Waerner
H. L. Walker
E. P. Warner
W. W. Wells
C. E. Wilson
H. F. Wood

Activities of the Sections

Meetings Committee Chairmen

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W. W. Slaght, Pierce-Arrow Motor Car Co., 1695 Elmwood Avenue, Buffalo
CLEVELAND SECTION
John Younger, 13,515 Lake Shore Boulevard, Bratenahl, Cleveland
DAYTON SECTION
Iskander Hourwich, Antioch College, Yellow Springs, Ohio
DETROIT SECTION
O. E. Hunt, Chevrolet Motor Co., General Motors Building, Detroit
INDIANA SECTION
O. C. Berry, Wheeler-Schebler Carburetor Co., Indianapolis
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C. B. Veal, Manly & Veal, 250 West 54th Street, New York City
MID-WEST SECTION
George W. Cravens, Hotel Lafayette, Clinton, Iowa
MINNEAPOLIS SECTION
L. A. Emerson, 4852 South Washburn Avenue, Minneapolis
NEW ENGLAND SECTION
V. A. Nielsen, V. A. Nielsen Co., 701 Beacon Street, Boston
PENNSYLVANIA SECTION
E. L. Clark, Commercial Truck Co., Hunting Park and Rising Sun Avenues, Philadelphia
WASHINGTON SECTION
William M. Wallace, Bureau of Construction and Repair, Navy Department, City of Washington

FOR the information of prospective authors of automotive engineering papers, there are given above the names and addresses of the members who will select the Section meeting programs during the coming year. If you are considering the preparation of a paper for presentation at a Section meeting, it is important that you write the Meetings Chairman of that Section, making certain that your topic fits in with the general plan he has conceived for the Section year. Submit an outline of your contemplated paper along with the subject. The Chairman is thus enabled to suggest possible extension or revision of the paper.

Photographs of some of the Section Chairmen and Vice-Chairmen are reproduced on the following pages. The following list gives the name of the office and the Section for the various officers.

NAME	OFFICE	SECTION
C. A. Criqui	Vice-Chairman	Buffalo
Victor Gauvreau	Vice-Chairman	Minneapolis
George E. Goddard	Chairman	Detroit
Iskander Hourwich	Vice-Chairman	Dayton
J. H. Hunt	Chairman	Dayton
W. E. Kemp	Chairman	Metropolitan
Taliaferro Milton	Chairman	Mid-West
R. J. Nightingale	Vice-Chairman	Cleveland
O. A. Parker	Chairman	Cleveland
Col. F. H. Pope	Chairman	Washington
H. W. Slauson	Vice-Chairman	Metropolitan
Lon R. Smith	Vice-Chairman	Indiana

Secretaries of the Sections

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A. J. Fitzgibbons, 168 Claremont Avenue, Buffalo
CLEVELAND SECTION
E. W. Weaver, 5103 Euclid Avenue, Cleveland
DAYTON SECTION
R. B. May, Dayton Engineering Laboratories, Dayton
DETROIT SECTION
Thomas J. Litle, Jr., 733 Seyburn Avenue, Detroit
Mrs. B. Brede, Assistant Secretary, 1361 Book Building, Detroit
INDIANA SECTION
B. F. Kelly, Weidely Motors Co., Indianapolis
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Phil N. Overman, 10 South 10th Street, Minneapolis
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PENNSYLVANIA SECTION
Edward L. Clark, Hunting Park and Rising Sun Avenues, Philadelphia
WASHINGTON SECTION
Benjamin R. Newcomb, 211 Victor Building, City of Washington

W. E. KEMP

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TALIAFERRO MILTON

COL. F. H. POPE

O. A. PARKER

VICTOR GAUVREAU

ISKANDER HOURWICH

C. A. CRIQUI

H. W, SLAUSON

LON R. SMITH

R. J. NIGHTINGALE

AN ECONOMIC FOUNDATION FOR HIGHWAYS[1]

IT has been accepted as a tenet that highway bonds shall be issued for a shorter period than the life of the road. The Cumberland Pike, the first and only great national highway of the early days, was opened to traffic about 1818. Today, after a full century, that highway is still in existence, rehabilitated and carrying an ever-increasing traffic. It is true that, due to changes in overland transportation, there was a long period when this highway fell into disuse and disrepair, but today, with the surfacing repaired and rebuilt, the same road-bed and most of the same waterway structures are serving traffic. The extension of this road from Baltimore to Cumberland, originally a state project, was resurfaced in recent years at a nominal cost. This light surfacing was placed for horse-drawn traffic, but with continued maintenance and necessary reconstruction from time to time, is now carrying heavy automotive traffic. Under the theory just mentioned, for how long a period would the bonds, had the road been so built, have been proper? Apparently the road is in better condition today than it has ever been in the past.

The term "permanent roads" is a fallacy that is responsible for more or less of the unsound theory that has been advanced with reference to bond issues. There are certain of the essentials of road construction, such as the road-bed and drainage structures, that can be built so that the deterioration is so slow that they may be properly termed "permanent." Road surfaces of whatever type deteriorate. Maintenance should begin as soon as the surface is thrown open to traffic, and the higher the cost of the road, the more careful in detail should be the maintenance. Even in a single State conditions vary to the extent that loadings which can be carried without deterioration of some road surfaces will inflict untold damage upon others. Yet under the average regulatory law the utility of the best roadways is not realized in full, nor is the safety of the lesser capacity roadways assured.

Research in the field of the weight of loads that can be carried by different road surfaces is revealing definite information. The influence of subgrade soils, tire equipment, distribution of the loads to the wheels, the speed and many other variables is too complex to be written into law. The seasonal variation alone in the carrying capacity of the road-beds, due to moisture conditions, is one of the most serious of all causes of road damage, and for this single reason the highway authorities must be given wide discretion in traffic regulation.

Our total expenditures for all highway purposes in the United States last year were approximately $600,000,000. An examination of the progress in highway building over the past 10 or 12 years indicates a very serious lag in the development of the road-bed in comparison to the increase in rolling stock. This lag is so serious that there is an insistent demand on the part of the large body of vehicle owners that the providing of improved road-beds shall be hastened. This demand in turn is followed, and properly so, because of the very large expenditures required, by the equally insistent question of who shall pay.

With the exception of a very few States it is apparent that the major highway development for the next decade and probably longer will be upon the so-called 7-per cent system that is now being established under the requirements of the Federal Highway Act of November 1921. We have now on a national scale the inauguration of the first cardinal principle of efficient highway administration, the classification of the roads into selected systems. On the 7-per cent system Federal and State funds will be concentrated until completion. This classification, recognizing with full credit the progress made by many of the States before the federal enactment, is fundamental. Through its operation the roads of greatest importance are first improved and all work done is accumulative toward the completion of a system of interconnected and correlated traffic lines as between the communities within the States and between the States themselves. As the highway systems are completed the traffic will become more and more organized and concentrated on the improved roadways. If these systems are not properly selected they will eventually have to be revised and so much of the investment jeopardized.

It is believed that a proper financial policy will require first that all of the maintenance funds be met from the revenues derived from the road user; second, that the costs of reconstruction be met from the revenues derived from both the road user and from State taxes, the relative percentages being different for different States and for the different types of roadway built; third, after deducting the federal aid, the cost of new construction should be divided between the road users and other State taxes from both urban and road sources.

The total expenditures for rural highway purposes in the United States for the past year came from the following sources:

	Per Cent
Local Road Bonds	33
Federal Aid	14
State Road Bonds	7
State Taxes and Appropriations	12
County, Township and District Taxes and Assessments	14
Motor-Vehicle Revenues	19
Miscellaneous	1

Federal aid and motor-vehicle revenues constituted 33 per cent; the remaining 67 per cent either comes directly or will eventually be paid from State and local taxes. It is believed that a very considerable readjustment of these sources of revenues must be made, so that a larger percentage will be paid by the road user and a lesser percentage from State or local taxes. This statement of motor-vehicle revenues is not fairly representative of the total funds collected directly or indirectly from the motor vehicles, for our estimates show the following revenues paid during the fiscal year 1921:

Federal Taxes Paid by Builders on Passenger Cars and Motor Trucks	$115,546,249
State Registration Fees	122,478,654
Personal Property Taxes	52,500,000
Wheel and Privilege License	3,636,243
Gasoline Tax, 17 States, Calendar Year 1922, Estimated	11,000,000
Total	$305,161,146

The total of $305,161,146 is equivalent approximately to $30 for each car and truck registered during 1921. This total sum, it will be noted, amounts to over one-half of the total estimated road expenditure for 1921.

For over 11,000 miles of federal-aid roads involving a total cost of over $188,000,000, the surfacing cost is 60 per cent and the remaining 40 per cent is made up of: Grading, 22 per cent; structures, 14 per cent; shoulders, 1 per cent; engineering, 3 per cent. That is, 40 per cent of the expenditures were for the permanent features of the road. The greater part of the expenditures for surfacing can be rightfully considered in the same class, for it is being successfully demonstrated that by supplementary construction, such as the placing of new surfaces and widening by addition of shoulders, the greater part of the investment can be continually maintained and thus preserved indefinitely. Expenditures of this class fall into the category of maintenance and reconstruction, and if continuance of these two can be relied upon the original investment is preserved indefinitely. Therefore, little fault can be found with the soundness of the financing if bonds are issued for new construction provided that the original investment is thus continually preserved.

Of all these requirements it appears that the funds for the

[1] From an address delivered at the annual meeting of the Chamber of Commerce of the U. S. A. by Thomas H. MacDonald, chief of the bureau of public roads, Department of Agriculture.

new construction in the amounts that will be needed will be the most difficult to secure. The public should not be called upon to bear this entire cost as an annual expense. Rather is it fair to distribute these costs over a period of years extending beyond the time of the maximum expenditures for new construction. It is not doubtful that succeeding generations will enjoy the use of the highways that are being built now, and there is nothing unfair in pursuing a financial policy that distributes a portion of the cost beyond the immediate period. No enterprise requiring funds in the large amounts that they will be required for the new construction of highways can go forward without anticipation of revenues.

Experience in the administration of highway work has already established certain definite principles without the observance of which it would be highly unsafe to resort to bond issues. The major principles may be outlined as follows:

(1) As a general proposition, State, not local, bonds should be issued for the building of the State highway system. In any case the total amount of bonds to be issued for any one year or over a period of years in any State should be subject to the control of one central body

(2) Sufficient revenues must be derived from the users of the highways to pay all of the maintenance and a percentage of any reconstruction charges

(3) All bond-built highways ought to be maintained under the direct supervision of the State highway department, which also must have jurisdiction of the revenues from the road users. The proper maintenance of all bond-built roads must be considered a first lien upon such revenues

(4) The proceeds of bonds must be devoted to a system of roads so devised that as the system is completed these roads will in fact continually serve the major traffic in the vicinity. This means that the only safe sequence of improvement is in the order that the different sections of the system serve the traffic

(5) Bonds should not be issued in an amount beyond the ability of the traffic of the present and immediate future to pay revenues sufficient for the proper maintenance of the roads built from the proceeds

(6) The types of road built with bonds should be adjusted to the traffic that they will be called upon to bear within the reasonable future

It will be noted that little has been said as to the uses that are to be made of the highways, but it must be remembered that the whole question of highway transportation is in its infant stages. Careful investigations and studies are now going forward to determine both the economic limitations of highway transport and of the necessarily adjusted highway construction. We have made rapid progress during the past 2 years toward such determination, and the suggestions that have been made are based upon an administration of a highway program predicated upon the results and determinations of these scientific researches and investigations.

If permanent policies can be adopted and carried forward without change over a period of years the necessary roads can be built without undue burdens upon the public. The result will depend upon foresight and adherence done by foreseeing now and adhering to safe and sound financial policies.

OBITUARY

EDWARD VASSALO HARTFORD, president of Edward V. Hartford, Inc., New York City, and vice-president of the Great Atlantic & Pacific Tea Co., died June 30, 1922, at his residence in New York City, aged 52 years. He was born May 28, 1870, at Orange, N. J., and following his public school training received his technical education at Stevens Institute, Hoboken, N. J.

From 1897 to 1900 he resided in Paris, France, and when on a visit to the United States in 1898 he brought with him one of the first motorcycles. He was an originator, designer and patentee of shock absorbers in the United States and abroad, being for some time president of the Hartford Suspension Co., Jersey City, N. J., which has manufactured shock absorbers and automobile jacks for a number of years. He is survived by his widow, a daughter and a son.

Mr. Hartford was elected to Member grade in the Society Oct. 30, 1911.

LIONEL N. LOUPRET died from an attack of heart disease on the golf course of the Country Club at Kokomo, Ind., July 1, aged 34 years. He was born at Lowell, Mass., on Nov. 1, 1887, and when only 16 years of age entered the service of the Heinze Electric Co. of that city. This was in April, 1904, a short time before the company began to build ignition apparatus, and Mr. Loupret assisted J. O. Heinze in the production of the first ignition coil made by the company and worked directly under Mr. Heinze's personal supervision. With the exception of a little over a year when Mr. Loupret was in charge of the Detroit branch of the company he spent all of the 16 years he was connected with the organization in the plant at Lowell. He had charge of the inspection, testing and production and from 1915 to 1920 supervised the engineering and experimental department. In March, 1920, he became associated with the Kokomo Electric Co., Kokomo, Ind., in its engineering department and at the time of his death was sales engineer of the firm of the Byrne, Kingston Co., which is also located at Kokomo.

Mr. Loupret was elected to Member Grade in the Society Dec. 4, 1920.

RECENT COUNCIL MEETINGS

AT the Council meeting that was held at White Sudphur Springs, W. Va., on June 19, members in attendance were: President Bachman, Past-President Beecroft, Vice-President Young, and Councilors Crane and Smith.

One hundred and twenty-five applications for individual membership and 30 for student enrollment were approved. The following transfers in grade of membership were made: From Junior to Associate, John Pollitt; Associate to Member, George L. Appleyard, W. H. Hollister, Warwick Ray, James W. Wilford, Linus J. Parker, Charles M. Kearns; Junior to Service Member, E. W. Rounds, J. F. Alcure; Junior to Member, Richard W. Fulton, Edward C. Bartlett.

Various matters connected with the Automotive Production Meeting of the Society to be held in Detroit on Oct. 26 and 27 were discussed.

The following appointments to the Standards Committee were made:

LUBRICANTS DIVISION—K. G. Mackenzie
PASSENGER-CAR BODY DIVISION—S. J. Baum
STORAGE BATTERY DIVISION—J. L. Rupp

The subject of Tail-Lamp Illumination was assigned to the Lighting Division.

A meeting of the Council was held in New York City on July 26.

It is expected that the next meeting of the Council will be held in September.

Applicants *for* Membership

The applications for membership received between June 15 and July 14, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ANDERSON, IVAN L., student, University of Utah, *Salt Lake City, Utah.*

BEAN, WILLIAM LLOYD, mechanical assistant to the president, New York, New Haven & Hartford Railroad, *New Haven, Conn.*

BEAUMONT, JOHN SYDNEY, metallurgist, Ford Motor Car Co. Ltd. of Canada, *Walkerville, Ont., Canada.*

BOSCHAN, ALEXANDER S., student, University of Michigan, *Ann Arbor, Mich.*

BURN, WALTER P., sales promotion and advertising manager, Transcontinental Oil Co., *Pittsburgh.*

BURROWS, LEES J., special service representative, Oakland Motor Car Co., *Pontiac, Mich.*

BUTCHER, HAROLD E., sales manager, Champion Spark Plug Co., *Toledo, Ohio.*

COFFIN, O. W. L., division manager, White Co., *Cleveland.*

COLEY, GLENN, metallurgist, Timken-Detroit Axle Co., *Detroit.*

DAVIDSON, ROBERT P., business manager, *Motor, New York City.*

DAVIDSON, WILBUR S., manager branch office, Rich Tool Co., *Detroit.*

DAVISON, WALTER W., factory manager, Wire Wheel Corporation of America, *Buffalo.*

DESCHAMPS, ROGER P., manager, Standard Automobile Co., *Brussels, Belgium.*

DUBOIS, CHESTER B., mechanical engineer, Jones Speedometer, Inc., *New Rochelle, N. Y.*

ELLIOTT, HARRY C., engineer, Packard Motor Car Co., *Detroit.*

ENFIELD, W. L., manager, lamp development laboratory, National Lamp Works, *Cleveland.*

ERSKINE, ALBERT R., president, Studebaker Corporation of America, *South Bend, Ind.*

GORDON, JOHN RUTHERFORD, body builder, Holdens Motor Body Builders, *Adelaide, Australia.*

GOTTSCHALK, V. H., research engineer, Society of Automotive Engineers, Inc., *New York City.*

GRANT, HUGO B., mechanical engineer, Turbulator Corporation, *Chicago.*

HANCOCK, G. A., engineering and research work, L. H. Gilmer Co., *Tacony, Philadelphia.*

HERMANN, HENRY E., draftsman, Phelps Light & Power Co., *Rock Island, Ill.*

HOLT, HARRY W., secretary, Charles B. Bohn Foundry Co., *Detroit.*

HOOVER, H. EARL, chief engineer, Hoover Suction Sweeper Co., *Chicago.*

JACKSON, H. GARDNER, assistant general manager, Wire Wheel Corporation of America, *Buffalo.*

JAGGER, NORMAN A., salesman, Kelsey Motor Car Co., *Newark, N. J.*

KERBER, JOSEPH H., shop superintendent, Wisconsin Highway Commission, *West Allis, Wis.*

KING, TOWER W., materials testing engineer, engineering division, Air Service, McCook Field, *Dayton, Ohio.*

KNISKERN, WALTER H., chief engineer, Atmospheric Nitrogen Corporation, *Syracuse, N. Y.*

KRAUS, SYDNEY M., bureau of aeronautics, Navy Department, *City of Washington.*

LUTZ, EARL C., assistant superintendent, International Harvester Co., *Chicago.*

MCCARTY, HARRY HARPER, research engineer, General Motors Research Corporation, *Dayton, Ohio.*

MCGOLDRICK, WILLIAM J., vice-president and chief engineer, Power Truck & Tractor Co., *St. Louis.*

MACKENZIE, RUSSELL E., carbureter engineer, Detroit Lubricator Co., *Detroit.*

MACKIE, HORACE S., treasurer and general manager, Electrocar Corporation, *New York City.*

METZ, ALBERT G., editor, Chilton Co. *Philadelphia.*

NEAL, JAMES BENSON, manager, Norton Laboratories, Inc., *Lockport, N. Y.*

NORWOOD, H. E., general manager and engineer, Perfect Window Regulator Co., *New York City.*

OHTSUKA, CAPT. N., Imperial Japanese Artillery, office of Japanese Consul General, 165 Broadway, *New York City.*

PAYNE, F. G., chief mechanical assistant, Shanghai Municipal Electricity Department, *Shanghai, China.*

ROBINSON, FRANCIS T., office manager, Society of Automotive Engineers, Inc., *New York City.*

SALISBURY, C. E., service manager, Hupp Motor Car Corporation, *Detroit.*

SCOVEL, ALFRED J., sales department, Marvel Carbureter Co., *Flint, Mich.*

SEELINGER, W. J., works manager, Ohio Body & Blower Co., *Cleveland.*

SIMPSON, H. W., engineer, Ford Motor Car Co., *Dearborn, Mich.*

STEPHENS, ELTON S., mechanical engineer and inventor, International Harvester Co., *Chicago.*

STOCKINGER, W. L., representative engineering and production departments, Studebaker Corporation of America, *Detroit.*

STREETER, EDWARD L., JR., general manager, J. N. Lapointe Co., *New London, Conn.*

STUTTER, CLINTON L., J. Linek, 6 Washington Street, *Maspeth, N. Y.*

TIMKEN, H. H., president, Timken Roller Bearing Co., *Canton, Ohio.*

WILLS, C. HAROLD, president, C. H. Wills & Co., *Marysville, Mich.*

WITMER, GEORGE, branch manager, Sterling Motor Truck Co., *Brooklyn, N. Y.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between June 10 and July 10, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

BERKOWITZ, SAMUEL (A) president and engineer, Automotive Products Corporation, *Hazleton, Pa.*, (mail) 325 West Diamond Avenue.

BOCZ, ALEX J. (M) mechanical engineer and designer, 2169 Richton Avenue, *Detroit.*

BROSSEAU, A. J. (A) president, Mack Trucks, Inc., 25 Broadway, *New York City.*

BROWN, GARNET C. (M) research engineering department, United & Globe Rubber Co., Trenton, N. J., (mail) 125 Pleasant Street, *Royal Oak, Mich.*

BROWNING, W. E. (J) civilian instructor, Air Service Mechanics School, Chanute Field, *Rantoul, Ill.*, (mail) P. O. Box 556.

CASON, PAUL B. (J) draftsman, Kimball Motor Truck Co., *Los Angeles, Cal.*, (mail) 1739 South Bronson Avenue.

CLARK, OLIVER H. (M) body engineer, Zeder-Skelton-Breer Engineering Co., 24 Mechanic Street, *Newark, N. J.*

COWAN, WILLIAM ARTHUR (A) assistant chief chemist, research laboratories, National Lead Co., 129 York Street, *Brooklyn, N. Y.*

CREAMER, CHARLES DELL (E S) student, Ohio State University, Columbus, Ohio, (mail) *Washington C. H., Ohio.*

DALE, E. B. (A) assistant professor of automotive mechanics, Colorado State Agricultural College, *Fort Collins, Col.*

FUHRER, MAX (E S) student, Armour Institute of Technology, *Chicago*, (mail) 3227 Fullerton Avenue.

GARY, C. E. (M) draftsman, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*, (mail) 630 West Newell Street.

GRUNDMANN, WILLIAM R. (A) service manager, Mercer Motors Co., *Trenton, N. J.*

HABERKOST, NOEL F. C. (E S) student, Ohio State University, Columbus, Ohio, (mail) *Munroe Falls, Ohio.*

HEMEON, J. RUSSELL (J) student, Massachusetts Institute of Technology, Cambridge, Mass., (mail) 9 Standish Street, *Boston, 24.*

KELLER, K. T. (M) manufacturing manager, Chevrolet Motor Co., *Detroit*, (mail) 1177 Edison Avenue.

KURTI, A. (M) *Doylestown, Ohio.*

LEWENTHAL, ALLEN (A) service manager, Smith Motor Sales Co., *Phoenix, Ariz.*, (mail) 146 North 11th Avenue.

MARQUIS, ALBERT C. (A) salesman, Dayton Iron & Steel Co., *Dayton, Ohio*, (mail) P. O. Box 7.

MARTY, WILFRED H. (A) assistant chief inspector, Wright Aeronautical Corporation, *Paterson, N. J.*, (mail) 454 East 31st Street.

MEDDAUGH, RAY C. (J) Sutter & Meddaugh, *Columbus, Ind.*

MOOSBRUGGER, JOSEPH (J) airplane structural designer, Dayton Wright Airplane Co., *Dayton, Ohio*, (mail) 1005 North Broadway.

OSBORN, GEORGE W. (A) superintendent, bearing department, Hoyt Metal Co., *Granite City, Ill.*, (mail) 2132 G. Street.

PERRY, JAMES W. (A) general manager, automotive equipment and electrical departments, Johns-Manville, Inc., 296 Madison Avenue, *New York City.*

SHERRARD, JOE O. (E S) student, Ohio State University, *Columbus, Ohio*, (mail) 4 East Norwich Avenue.

SKELTON, O. R. (M) vice-president, Zeder-Skelton-Breer Engineering Co., *Newark, N. J.*, (mail) Robert Treat Hotel.

SKRIBA, LOUIS S. (E S) student, Armour Institute of Technology, *Chicago*, (mail) 726 West 17th Street.

SPRINGER, RUSSELL S. (M) vice-president and factory manager, Holt Mfg. Co., *Stockton, Cal.*

TUTHILL, LEON H. (A) shop superintendent, Reo Motor Car Co., New York City, (mail) 552 East 34th Street, *Brooklyn, N. Y.*

VERPLANK, A. J. (E S) student, Armour Institute of Technology, Chicago, (mail) 100 West Ridge Road, *Gary, Ind.*

WALWORTH, RICHARD H. (E S) student, Armour Institute of Technology, Chicago, (mail) 41 Broad Street, *Hillsdale, Mich.*

WHITE, ROBERT B. (J) lubrication engineer, Taxman Refining Co., Chicago, (mail) 116 South Michigan Avenue.

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XI September, 1922 No. 3

Chronicle and Comment

Stationary-Engine Vice-Presidency

L. W. WITRY has been nominated to serve during 1923 as second vice-president for stationary internal-combustion engineering. In reporting to this effect, the Nominating Committee, of which Cornelius T. Myers and B. S. Pfeiffer were chairman and secretary respectively, completed its work. The other nominees for the offices of the Society next falling vacant under constitutional provision were listed in the July and August issues of THE JOURNAL.

An Ordnance Exhibition

THE Ordnance Department of the United States Army has invited the members of the Society to witness a demonstration of ordnance materiel at the Aberdeen Proving Ground, Md., on Friday, Oct. 6. Members who attended a similar demonstration held in 1921, strongly praised the program as being interesting and instructive. All types of ordnance are assembled for inspection and firing. The latest types of mobile artillery mounts, tanks and trailers are demonstrated. This annual exhibition is arranged to acquaint engineers in the industry with the progress made in ordnance design, thus fitting them to serve the Country efficiently in a national emergency. Details of the meeting are given on p. 281.

1923 Budget

THE tentative figures being considered by the Finance Committee in connection with the preparation of its report to the Council on the financial budget of the Society for the fiscal year beginning Oct. 1 next, are shown in the last column of the accompanying table. The other columns indicate the experience during the last 3 fiscal years to date, except that the figures in the 1921-1922 column represent estimates for the portion of the present fiscal year that had not expired at the time the table was prepared.

COMPARATIVE STATEMENT OF INCOME AND EXPENSE

	1919–1920	1920–1921	1921–1922	Estimate 1922–1923
INCOME				
Initiations	$19,940.00	$19,265.00	$12,471.00	$15,000.00
Dues and Subscriptions	67,299.50	76,456.50	74,354.67	75,000.00
Affiliated Appropriations	8,125.00	7,500.00	8,000.00	7,500.00
Interest	4,716.95	5,567.21	4,841.89	5,000.00
Advertising Sales	139,090.18	123,994.54	127,944.20	150,000.00
Miscellaneous Sales	8,034.92	10,062.86	11,927.26	12,000.00
	$247,206.55	$242,846.11	$239,539.02	$264,500.00
EXPENSE				
Publications	$65,078.57	$61,425.33	$57,373.91	$57,700.00
Sections	8,807.03	10,630.48	10,995.29	10,000.00
Research	....	1,004.98	15,351.89	19,500.00
Employment Service	2,976.35	1,324.77	5,454.66	7,000.00
Membership Increase	6,915.95	8,614.79	8,495.11	8,850.00
Standards	20,047.23	27,960.60	29,019.24	27,500.00
Cost of Advertising Sales	21,302.27	23,209.19	36,647.94	41,750.00
Cost of Miscellaneous Sales	3,824.12	6,881.31	5,924.56	6,000.00
Meetings	15,131.79	10,379.17	12,741.65	10,000.00
General Administration	83,567.12	80,864.09	77,384.24	76,200.00
	$227,650.43	$232,294.71	$259,388.49	$264,500.00
Unexpended Income	$19,556.12	$10,551.40		
Deficit			$19,849.47	

Internal-Combustion Engine Nomenclature

THERE is no doubt of the logic and propriety of the adoption by the Society of the word "engine" to denote the internal-combustion unit of motor vehicles. The reason for the action is, of course, to avoid

confusion, the word "motor" being applicable to the electrical unit usually installed on passenger cars. Many engineers use habitually the word "motor" instead of "engine" in the sense under discussion. Some companies manufacturing engines use the word "motor" in their official name. "Electric motor" and not "electric engine" is, of course, the name for the electric unit; and the term "steam motor" is not used as applying to a prime-mover.

"Engine trucks" and "engine vehicles" would be equally anomalous. There is, however, good reason for the terms "motor trucks" and "motor vehicles" on a basis of logic as well as of usage. The word "motor" in this connection can be taken to connote the fact that a moving vehicle is meant.

The Society has conducted no special propaganda in the connection other than to use the word "engine" consistently in its publications to indicate the internal-combustion unit of motor vehicles. The Society has no police power in connection with its standards work, but on the other hand that work should be supported as faithfully as advisable and feasible.

St. Lawrence Seaway Project

THE report of the International Joint Commission that has conducted a long investigation of the St. Lawrence Seaway project contains the following statement:

> The Commission believes that while it is physically practicable to bring both railroads and terminals up to the point where they could handle the traffic of the United States without serious congestion the expense involved would be enormous, amounting, in the opinion of experienced railroad executives, to $2,000,000,000 per annum over a series of years, and it is perhaps questionable if in the end the relief afforded would be comparable to that promised by the creation of an all-water route from the interior of the continent to the Atlantic seaboard.
>
> Of the alternative routes suggested, the New York State Barge Canal, the projected Oswego ship canal, the Richelieu and Lake Champlain route, the proposed Georgian Bay canal, the Hudson Bay route, the Mississippi route and the Pacific and Panama Canal route, the commission is convinced that none of them offers the advantages of the St. Lawrence route either as a means of relief for the acute transportation situation or as a channel for the carriage of ocean-borne commodities.

Support This Good Work

SAVINGS in manufacturing costs estimated at $124 per passenger car and $171 per motor truck are believed to have resulted from the adoption of the standards and recommended practices developed by the Society of Automotive Engineers. This is a splendid showing. The standards work of this Society deserves every encouragement and support, even if for no other reason than the resulting lower costs of manufacture. But better cars have resulted from this work, too. The engineer has been able to pay more attention to fundamentals, knowing that much of the detail work can be eliminated through the use of the existing standards. Originality in design has not been submerged. Rather it has been fostered by allowing the engineer to give his whole attention to the more important problems of design and development.

Secretary of Commerce Herbert Hoover is authority for the statement that the automotive industry makes a greater use of existing standards, or simplified practices as he prefers to call them, than perhaps any other industry. This is fine, but we all know that even the present-day standards are not so universally used as they might be.—*Automobile Trade Journal.*

Detroit Aeronautic Congress

AT Detroit on Oct. 12 to 14 the Second National Aeronautic Congress will be held, during the airplane races. It is announced that an unparalleled number of modern airplanes for a flying event will be shown, affording a rare opportunity to study present development. Many members of the Society will take part in the conduct of the Congress and the races, including representatives of the National Air Association and the Aero Club of America.

The purpose is to organize an American association to advance the interests of aviation in the realm of business and from the standpoint of legislation. The attendance of all interested is urged. Railroad reduced-fare privileges have been obtained, as local ticket agents can explain.

The principal races, the conduct of which has been endorsed by the Secretaries of the War and the Navy Departments, are scheduled to be run, in addition to the Curtiss Marine Flying Trophy Race, on Oct. 7, on the following dates:

Oct. 12—Aviation Country Club of Detroit Trophy Race for Light Commercial Airplanes
Detroit *News* Aerial-Mail Trophy Race for Large-Capacity Airplanes
Oct. 13—Liberty-Engine Builders' Trophy Race for Airplanes
Oct. 14—Pulitzer Trophy Race

Fuel Research

THE fuel-research program, in which the Society is cooperating with the National Automobile Chamber of Commerce and the American Petroleum Institute, is making progress. Of the 10 firms that undertook independent tests of the four experimental fuels, five have already completed their tests and sent in their results.

The results so far show that for the models tested there is surprisingly little difference in the mileage obtained even with fuels very much heavier than present-day commercial grade gasoline. However, the figures for crankcase dilution, while not as complete as those for fuel-consumption, show much more dilution with the heavier fuels, suggesting that crankcase dilution may be the "neck of the bottle" so far as fuel volatility is concerned.

The tests at the Bureau of Standards, using the very ingenious recording apparatus shown at White Sulphur Springs by W. S. James, are now in full swing. There are at present seven engineers in Washington, loaned by various companies in the automotive and petroleum industries, to assist in carrying out this test program, which it is expected will be completed within the next 2 months.

The Detroit Production Meeting

THE forthcoming Production Meeting of the Society in Detroit, Oct. 26 and 27, has met with general approval among production men. Several prominent executives have commended the plan in recent interviews and have offered to contribute papers for the meeting program. The papers will be concerned with vexatious manufacturing problems, and methods to combat them, the attainment of greater precision in manufacture and special production studies. These papers

(*Concluded on p.* 264)

Vol. XI September, 1922 No. 3

Aluminum Pistons

By FERDINAND JEHLE[1] AND FRANK JARDINE[1]

DETROIT SECTION PAPER *Illustrated with* PHOTOGRAPHS AND CHARTS

THE lightness and high thermal conductivity of aluminum pistons are conceded and the paper deals principally with their thermal properties, inclusive of the actual operating temperature of the pistons, the temperature distributions in the piston, the effects of the cooling-water temperature and the piston material on the piston temperature. The apparatus is illustrated and described, and charts are presented and commented upon in connection with a discussion of the results obtained.

Theories affecting piston design are presented and discussed, reference being made to diagrams relating to design procedure. The work is supplementary to that done in 1921 by the authors, which they presented in a similar paper to which they refer.

ALUMINUM possesses two inherent advantages over all other metals that are used for piston materials, lightness and high thermal conductivity. Engineers and the public in general realize the value of lightness in a piston, as is proved by the various thin gray-iron pistons now being manufactured, the weight of which approaches that of the aluminum pis-

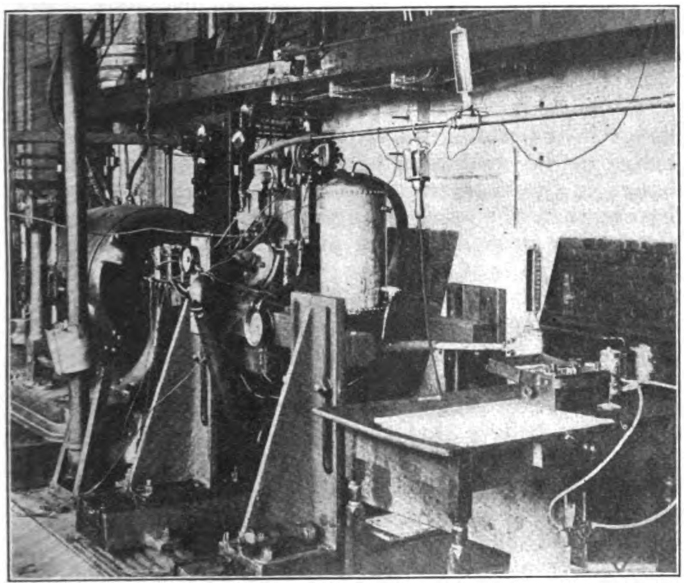

FIG. 1—APPARATUS EMPLOYED TO MEASURE PISTON TEMPERATURES

tons. The advantages gained by making the piston a good heat-conductor are not appreciated so thoroughly as the desirability of lightness, which again is demonstrated clearly by the numerous light iron-pistons. This paper deals principally with the thermal properties of the piston, such as the actual operating temperature of the piston, the temperature distribution in the piston, the effect of the cooling-water temperature on the piston temperature and the effect of the piston material on the piston temperature. We hope that it will be received as an additional chapter in the work of piston development.

[1] M.S.A.E.—Engineer, Aluminum Manufactures, Inc., Cleveland.

[2] See THE JOURNAL, May 1921, p. 397.

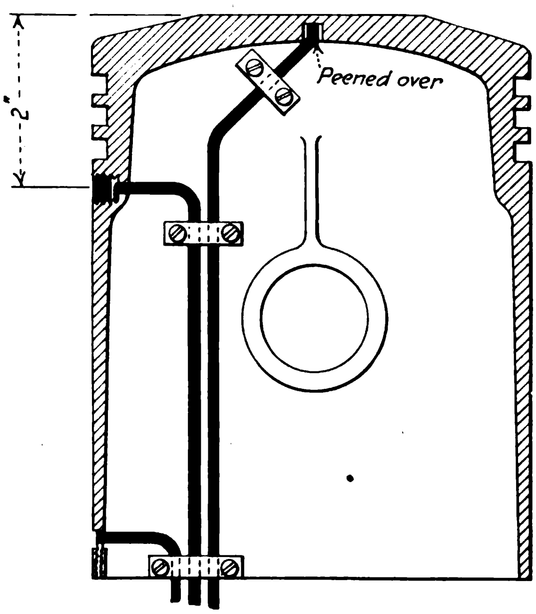

FIG. 2—METHOD OF FASTENING THERMOCOUPLES IN PLACE IN THE PISTON

In studying piston temperatures, the same set-up was used that is described in our previous paper on Aluminum Pistons.[2] Fig. 1 shows the apparatus. It consisted principally of a single-cylinder Liberty engine, a Sprague dynamometer and temperature-measuring equipment. The details of the temperature-measuring apparatus were illustrated in the previous paper and are not repeated here. The center head thermocouple was fastened by a slightly different method from that used in the previous work. In those experiments the head thermocouples were held in place either with a screw plug or a welded plug inserted from the outside of the piston. In the present work the head thermocouple was inserted by drilling a small hole only slightly larger than the thermocouple wires into the head, from the inside, for a short distance. The welded end of the couple was inserted in this hole

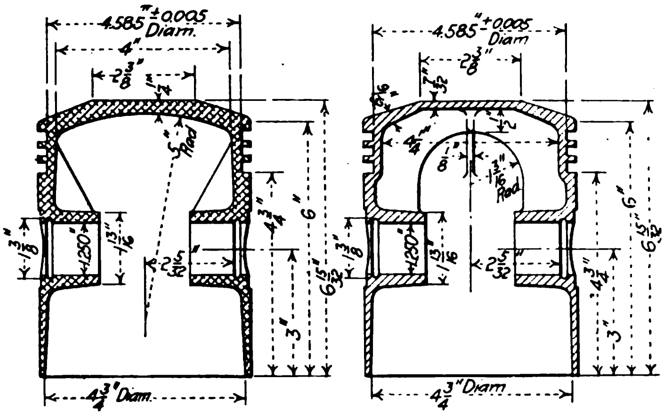

FIG. 3—DETAILS OF THE TWO PISTONS TESTED
The Piston at the Left Was Made of Aluminum While Cast Iron Was Used for That at the Right

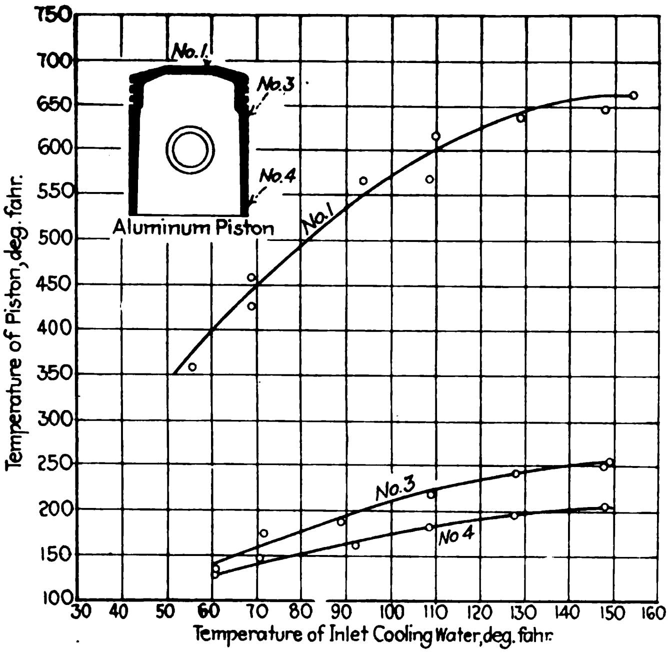

FIG. 4—TEMPERATURES AT SEVERAL POINTS ON THE PISTON PLOTTED AGAINST THE COOLING-WATER TEMPERATURE

and the surrounding metal was then peened over. Fig. 2 shows the method of fastening. The thermocouple had to be clamped securely to the piston, since the peened joint possessed no mechanical strength. By using this fastener for head thermocouples rather than the previous ones, there was absolutely no chance for hot gases to leak past the thermocouples and thus give a temperature reading higher than the true temperature of the piston-head. A comparison of the present with the former results indicates that the new method of thermocouple fastening was an unnecessary precaution. The thermocouples in the piston skirt were held in exactly the same manner as in the previous work, since there is no possibility of gas leaks at these points. During all of the tests reported in the present paper the engine speed was held at 800 r. p. m., with wide-open throttle and maximum spark-advance.

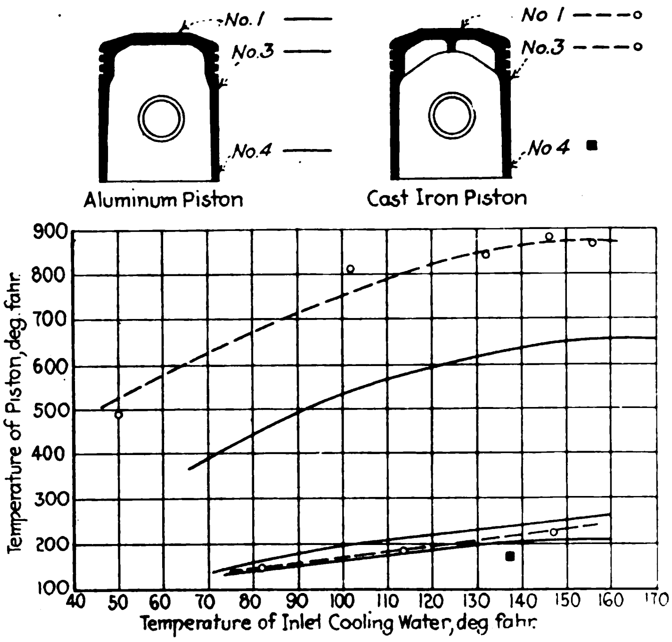

FIG. 5—COMPARISON OF TEMPERATURES OF THE CAST-IRON AND ALUMINUM PISTONS PLOTTED AGAINST THE COOLING-WATER TEMPERATURE

The power developed was between 12 and 13 b. hp. Care was taken to keep the fuel-consumption constant throughout the tests. Only readings obtained after engine conditions were constant were used in the computation of the results.

The view at the left of Fig. 3 is a detailed drawing of the aluminum piston used. It was 5 in. in diameter and had a head ¼-in. thick, with ample metal behind the rings. Without rings or pin it weighed 3 lb. Attention is called to the fact that this was not a regulation Liberty-engine piston.

EFFECT OF COOLING-WATER TEMPERATURE

A rather thorough study was made of the effect of the cooling-water temperature on the piston temperature. In the previous paper enough work was done on this subject to lead one to believe that the cooling-water temperature had an important effect upon the piston temperature. In the earlier work, piston-temperature readings were taken at only two temperatures of ingoing cooling-water; namely, 126 and 48 deg. fahr. In the present work, the highest temperature of ingoing cooling-water was approximately 160 deg. fahr., and measurements of piston temperatures were made at intervals of about 20 deg. until an ingoing-cooling-water temperature of 70 deg. fahr. was reached. The results obtained absolutely checked-up the indications of the previous work. The curves on Fig. 4 represent the temperatures of several points on the piston, plotted against the ingoing-cooling-water temperatures. The insert in the figure shows the points at which the temperature was measured. The temperatures of three points were determined; namely, in the center of the head, in the skirt immediately below the last ring and in the lower extremity of the skirt. The skirt temperatures were taken on the intake side of the piston.

The temperature of the head is affected most by a change in the cooling-water temperature; that is, a given change in the cooling-water temperature brings about a change in the head temperature of a greater number of degrees than the same change in ingoing-cooling-water temperature produces in the piston skirt. However, the rate of change is of the same general magnitude. It will be noticed that the curves flatten out at the higher cooling-water temperatures, which indicates that the piston temperatures under these conditions are not so susceptible to a change in the cooling-water temperature as they are at lower cooling-water temperatures. This would lead one to believe that the piston-head temperature in an air-cooled engine might not be so very much higher than in a water-cooled engine utilizing high cooling-water temperatures.

Comparing the different curves in Fig. 4, it is brought out clearly that there is an enormous drop between thermocouple No. 1, at the center of the piston-head, and thermocouple No. 3, immediately below the last ring. The drop from thermocouple No. 3 to thermocouple No. 4, that is, over the skirt, is not great. Unfortunately, thermocouple No. 2, immediately above the rings, was not operative. However, in the previous work the drop across the head was never 50 per cent of the drop across the rings and, since all of the measurements taken coincide so closely with the previous ones, it can be assumed safely that the greatest temperature-drop is across the rings.

PISTON MATERIAL CONDUCTIVITY EFFECT

In our previous paper a short series of tests was reported that showed the difference in the piston temperature was due to a difference in metal thickness; that is,

in the conductivity of the piston. Reference is made to some measurements of piston temperatures in a Diesel engine that were reported.[3] The pistons were made of cast iron and measured 400 mm. (15.75 in.) in diameter. Two pistons were tested, the difference between them being in head thickness. One had a head thickness of 52 mm. (2.05 in.); the other had a thickness of 61 mm. (2.40 in.) Under the same conditions of load and the like, the thinner piston showed a head temperature of 443 deg. cent. (829 deg. fahr.), while the thicker one showed a head temperature of only 402 deg. cent. (756 deg. fahr.). Unmistakably, the thicker the piston-head and the walls are, the lower the head temperature will be. This can be taken to mean that the higher the conductivity of the piston is, the lower the piston-head temperature will be, regardless of whether this increase of conductivity is brought about by thickening the section or by using a material possessing better thermal properties.

To prove definitely that cast-iron pistons operate at a higher head temperature than aluminum ones, we made some temperature measurements of a cast-iron piston running in the single-cylinder Liberty engine. The piston was operated under the same conditions as the aluminum piston while the results shown in Fig. 4 were being obtained. The drawing at the right of Fig. 3 gives details of the cast-iron piston, which is of somewhat lighter construction than the aluminum piston and has two ribs. These pistons represent about the usual relation between the cast-iron piston and the aluminum one that is used to replace it. The former weighed 5.9 lb. and the aluminum piston only 3 lb. Temperature measurements of this piston were made at different ingoing-cooling-water temperatures similar to those made on the aluminum piston. Fig. 5 is a chart that shows the difference between the cast-iron-piston temperatures and the aluminum-piston temperatures; they are just about what was expected.

The head temperature of the cast-iron piston is more than 200 deg. higher than that of the aluminum piston operating under the same conditions. For example, comparing the temperature of the two pistons when the ingoing-cooling-water temperature is 160 deg. fahr., we find that the head of the aluminum piston reaches a temperature of 660 deg. fahr. under these conditions, while the cast-iron piston-head registers 880 deg. fahr. When the ingoing-cooling-water temperature is reduced to 75 deg. fahr., we find the temperature of the aluminum piston-head is 420 deg. fahr., while that of the cast-iron piston reaches 660 deg. fahr. In other words, the cast-iron piston-head was as hot when the ingoing-cooling-water temperature was 75 deg. fahr. as the aluminum piston was at an ingoing-cooling-water temperature of 160 deg. fahr. A high piston-head temperature brings about certain well-known bad results such as carbonization above and below the piston head and preignition, which may necessitate lowering the compression. The point might be raised that, since the cast-iron piston is as hot with the cooling water at 75 deg. fahr. as the aluminum piston is when the cooling water is at 160 deg. fahr., an engine equipped with cast-iron pistons would operate better on a cold day than one equipped with aluminum pistons. Experience indicates that this is not the case. As A. L. Nelson points out in his paper on the Fuel Problem in Relation to Engineering Viewpoint,[4] a hot cast-iron piston-head does not assist vaporization more than an aluminum piston-head because the heat content

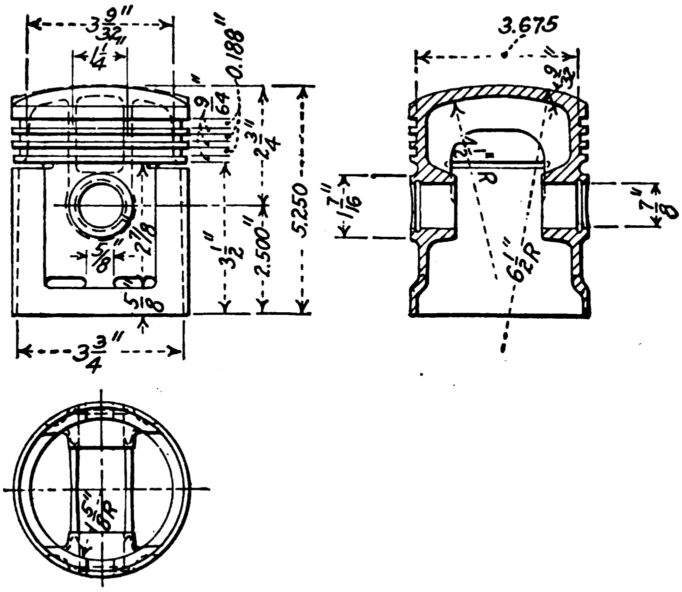

FIG. 6—IN THIS PISTON THE MECHANICAL EXPANSION DUE TO THE HEAD IS TAKEN CARE OF BY RELIEVING THE SIDES

of the two is about equal and the heat flow from the latter is much greater.

The temperature distribution in the cast-iron piston is somewhat similar to that which is obtained in the aluminum piston, except that it is exaggerated. The drop across the rings in the cast-iron piston is much greater; in fact, it is greater by something over 200 deg. fahr. than it is in the aluminum piston. The actual skirt-temperature is somewhat lower in the cast-iron piston. This can be accounted for by the fact that the thermal conductivity of cast-iron is much less than that of aluminum. Only one temperature-measurement was made on thermocouple No. 4; that is, at the bottom of the skirt. This great difference of temperature between the heads of cast-iron and aluminum pistons suggests that possibly the field for aluminum pistons is not limited to the high-speed passenger-car engine. The aviation engine is not generally considered a high-speed engine, yet aluminum pistons are used in the great majority of such engines. It can be stated that in aviation-engine practice the aluminum piston has not been used primarily for its

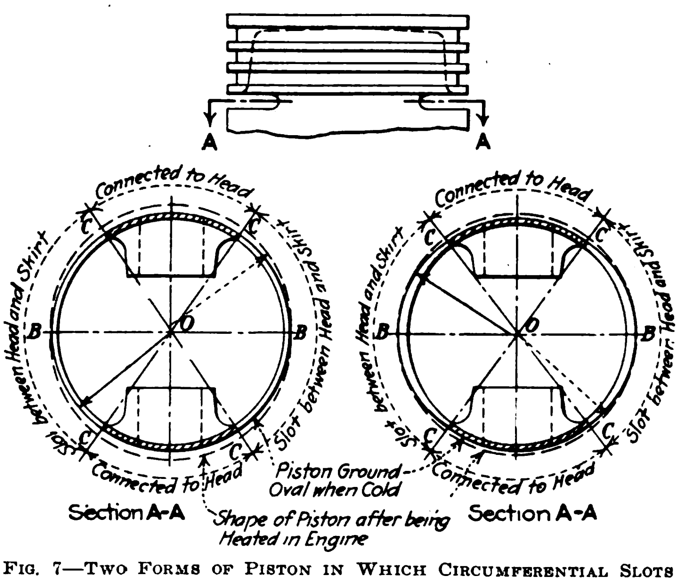

FIG. 7—TWO FORMS OF PISTON IN WHICH CIRCUMFERENTIAL SLOTS SEPARATE THE HEAD FROM THE SKIRT FOR A PORTION OF THE CIRCUMFERENCE

[3] See *Zeitschrift des Vereines Deutscher Ingenieure*, Aug. 27, 1921, p. 923.
[4] See THE JOURNAL, February 1921, p. 101.

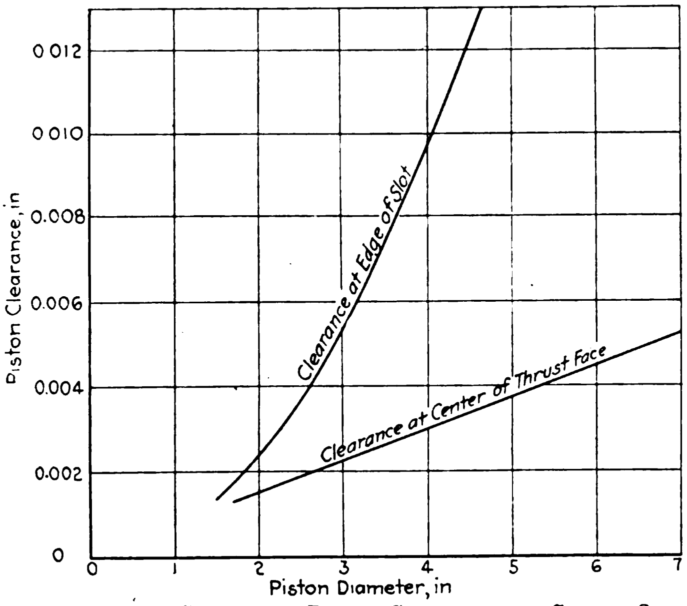

FIG. 8—CURVES GIVING THE PROPER CLEARANCE FOR SLOTTED-OVAL PISTONS

lightness, but because of its excellent thermal properties. The motor-truck engine is not a high-speed engine; but it is a heavy-duty engine; and so is the tractor engine. As a rule such engines are equipped with cast-iron pistons of fairly heavy section. The heating conditions in a truck and in a tractor engine are far more difficult to handle than in the passenger-car engine which runs at a higher speed. Much could be gained in the performance of slow-speed heavy-duty engines by the use of aluminum pistons. The cooler piston-head would permit higher compressions without a doubt, and would reduce carbon deposit and all its attendant evils. Incidentally, it would eliminate much vibration and many evils that accompany vibration, such as the loosening of joints. A reduction of bearing pressures would be assured also, and this is very desirable.

THEORIES AFFECTING PISTON DESIGN

The general theory upon which the separation of the piston-head from the skirt is based is given in detail in our previous paper. It is that the expansion of a piston skirt is due to two things; (*a*) the thermal expansion due to the temperature of the skirt and (*b*) the mechanical distortion of the skirt caused by the thermal expansion of the head.

To minimize the effect of the expansion of the head upon the skirt, circumferential slots separate the thrust faces from the head. This permits the head to expand free from the separated portions of the skirt. The sides of the skirt attached to the head are relieved to make the expansion in that direction harmless. The slotting of pistons in one way or another is not new, although the application to the aluminum piston, in conjunction with other things, is of more recent date. It has been found that a piston equipped with circumferential slots, as shown in Fig. 6, has a harder bearing along the edge of the bearing surface than along the center of the thrust face. This difficulty can be remedied.

Fig. 7 illustrates a piston that has circumferential slots separating the head from the skirt for a certain portion of its circumference. The view on the left-hand side is a section on *A A* of such a piston, which was machined to a true circle when cold. When the piston is heated during its operation in an engine, the effect of head expansion along the center-line *B B* of the slotted portion of the skirt is almost negligible; in fact, it may be negative. The expansion of that portion of the skirt connected to the head is exactly the same as in a piston without any slots. Therefore, while the piston is in operation, the portion of the skirt that is slotted away from the head is no longer a true circle having its center at the center of the piston. The radius *O C* to the edge of the slot has increased in length more than the radius *O B* to the center of the slotted area. The two thrust faces are, therefore, somewhat oval in shape and the piston will show a heavy bearing or score-marks along the edges at *C*. The dotted lines show the shape of the piston when it is in operation.

It is easy to give the piston skirt such a shape when cold that it will have the proper shape when heated during operation in an engine. In Fig. 7 it has been assumed that the proper shape of the heated piston is a circle concentric with the center of the piston. The right-hand view of the section on *AA* in Fig. 7 illustrates a piston machined oval for the entire length of the skirt to such a degree that it will expand into a true circle in operation. Again the dotted line shows the shape of the piston skirt in operation.

There are, of course, several forces at work controlling the expansion of a piston skirt and, even if the temperatures of the skirt and of the cylinder were known, it would be a very complicated problem to calculate the exact amount of clearance for the different portions of the skirt. These clearances must be determined experimentally. It is reasonable to expect that the point *C* in Fig. 7 on the bearing surface of the piston at the edge of the circumferential slots should have the same clearance as the ordinary trunk-piston, and this actually has worked out in practice. A very much smaller clearance can be used on the center of the bearing surface. The amount necessary has also been worked out experimentally. Fig. 8 shows two curves; the upper curve gives the clearance to be used at the edge of the slot as at *C* in Fig. 7, and the lower one gives the clearance for the center of the thrust face as at *B* in Fig. 7. While the curves in Fig. 8 are based upon experiments, the exact clearances necessary depend upon the type of engine and may, of course, vary somewhat.

It must be borne in mind that the edge of the bearing surface may not coincide with the edge of the slot as in Fig. 6. In this case the bearing surface is given the proper shape by drawing an arc between the point *B* and the imaginary point *C* in Fig. 7, proper clearances, as in Fig. 8, having been allowed at both points. The machining of such pistons need present no difficulties, since they can be ground easily by using a master cam.

THE DISCUSSION

A. A. BULL:—Perhaps the fundamental facts regarding slotted pistons are best represented by a statement that Mr. Baker made in presenting J. E. Diamond's paper, that of making the piston of such a construction that it will yield instead of sticking, which is a direct indication that it must expand. It is most undesirable that a piston should stick, but we must recognize also that, before it sticks, the pressure existing between the piston and the cylinder-wall must necessarily induce excessive friction. This is indicated best by installing some of the so-called non-expanding pistons in an engine running at say 2400 r.p.m. with the throttle set, noting how rapidly the engine speed begins to lag and then calculating the energy that is absorbed or wasted by the friction that is set up.

In the piston design illustrated in the Jehle and Jardine paper, having two circumferential slots below the

lower facing, the purpose was to create a differential expansion in the piston to the extent that at least one diameter would remain constant. That truly can be called a constant-clearance piston. The service in some 5000 cars, extending over a period of 3 years, has indicated the absolute success of that type of piston and the fact that it can be fitted satisfactorily to a clearance as low as 0.001 in. in a cylinder 3½ in. in diameter.

J. E. Diamond made reference to the wear on the aluminum piston caused by dust and described a test that had shown a very small amount of dust accumulating over a distance of 3000 miles. That is very probable but, unfortunately, we do not know the conditions under which the car was run. On the other hand, the car might have been run under conditions that probably would have accumulated 100 times as much dust in 3000 miles.

Our methods of measuring temperatures in aluminum and cast-iron have been somewhat different, perhaps, than those that have been illustrated and explained. Our method gives figures that agree closely with those given by Mr. Jardine. We first attempted to find what the average temperature of the piston-head was under actual operating conditions. This was accomplished by inserting a wire of a known coefficient of expansion through a piston-head. The wire was of a definite length when installed, and just long enough so that it would extend and rub on the cylinder-walls and become short to an extent such that measurement made upon it would indicate the average temperature. Simulating those conditions in a stationary set-up, with a flame-temperature on the head to give the same average head-temperature, thermocouples were installed in various parts of the piston to obtain the so-called heat-dissipation curve, and to ascertain also the effects of the cylinder temperature on the temperature of the various parts of the piston.

Concerning piston-rings, it was a common belief not very long ago that aluminum pistons were notorious oil-pumpers just because they were aluminum pistons; but, to consider the essentials, it is believed that so long as we have a material in the piston that has a higher coefficient of expansion, we necessarily must create between the ring and the piston groove a clearance under operating conditions which is greater than that which is obtained when the ring and the piston are fitted cold. That, in itself, is unquestionably the sole reason for any difference between cast-iron and aluminum pistons, so far as oil-pumping propensities are concerned. That brings us to the desirability of having a ring such as C. R. Manes has outlined, that will at least seal the piston-ring against one side of the groove so that it can adapt itself. When that is done, the question of lubrication and groove wear will have been eliminated.

Another factor that has not been given particular attention in this discussion of pistons is in regard to what the cylinder shape is under actual operating conditions. Fortunately, we have been able to determine what it is to some extent; it depends largely upon construction, and it has been demonstrated clearly that the problem of pistons and piston-rings is simplified greatly with a cylinder constructed so as at least to maintain its shape under operating conditions.

Another common belief is that the effect of vacuum in the cylinders is to increase the oil-pumping conditions. In observations of oil-pumping and of determining the best rings to use by running the engine without any cylinder-head and therefore having no vacuum to contend with, we find that the amount of oil that can be pumped up by the piston under those conditions is infinitely greater than it could ever be under actual operating conditions. Have we not been too prone to believe that we must depend entirely on pistons and rings to effect a cure for oil-pumping? Have we given full consideration to the conditions that surround the bearings or the amount of oil in the cylinder? I think not.

A. E. Damon:—We find that the circumferential slot around the head of the slotted piston is not there to check or retard the flow of heat down from the head of the piston to the skirt alone; it acts as a scavenger as well as a resistance to the heat flowing down to the skirt. We find also that, by adding a series of fins to the inside of the piston head which carries the boss, they conduct the heat very readily to the side-walls and the skirt, leaving the head, in the case of some of the air-cooled engines, practically cool; at least, so cool that we have been able to increase the compression and eliminate all spark-knock and preignition, this being indicative of no heat worth mentioning.

In regard to the slots being used to relieve the expansive effort of the piston against the side-walls of the cylinder, we are granting that the slot does that to a certain degree, no doubt; but, in the new type of E. C. Long piston of which I speak, the circumference of the skirt is actually smaller under heated conditions than it is when the piston is cold, practically relieving any tension whatever or any wall-pressure on the skirt or on the piston, other than enough to cause the cylinder-wall to act as a guide to the piston. There practically is no friction on the side-wall of the cylinder due to the piston skirt other than what we all know is a rolling friction of lubricating oil, which is the lightest friction it is possible to have.

F. Jehle:—It is true that there is a difference in the side-clearance between the ring and the groove, when the piston is cold and when it is hot. It is true also that this difference is greater with the aluminum piston than with the cast-iron piston. Therefore, if the width of the ring were reduced one-half, we would have a condition that more nearly approached the condition with the cast-iron piston. That is why I believe a narrow ring is a good thing.

Daniel Roesch:—Why limit the one-half-width ring to the aluminum piston?

Mr. Jehle:—I think a narrow ring in a cast-iron piston would be better than a wide one; this, however, for reasons other than to reduce the difference between cold and hot side-clearance.

Concerning heat throttling, while the slot separating the head from the skirt may throttle some heat, that is not the reason the slot is put there. Judging from the temperature measurements that were made on the piston without slots, there is not any great amount of heat that could be throttled off from the skirt; that is, most of it is thrown out by the ring section. The slot is put in primarily to give the head freedom to expand without distorting the skirt. If the small amount of heat throttling causes any difficulties, it is very easy to add a small amount of metal on the side where there is no slot, and thus compensate for the increase of thermal resistance caused by the slot.

I have never found any great difference between piston operation as regards heat, whether the piston had ribs and fins or not. Prof. F. C. Lea, who did much experimental work on pistons in England during the war, found that a multiplicity of ribs in the head of a piston merely adds to the difficulty of manufacturing it. He obtained results with numerous ribs that were no better than those obtained with some reasonable number, such as one or two ribs, according to his paper on Aluminum Alloys for

Aeroplane Engines, read before the Royal Aeronautical Society.

T. J. LÍTLE, JR.:—I should like to ask Mr. Manes how he determines whether a so-called pressure-proof ring is better than a plain ring. I take it that he refers to the use of these special rings, in many cases to replace the badly worn plain ring, and in such a case undoubtedly such a ring would be found to give improved results. I have made a very close study of various types of piston-ring, and am not very enthusiastic over many of the so-called pressure-proof rings. Many of them are constructed with tapering wedge-shaped sections that are very small in dimensions and extremely delicate and difficult to install. I have found that many such rings break during tests and have also had considerable breakage reported from service-stations.

I have noted other types of ring with three sections that were supposed to expand, not only diametrically but also to increase their total section to fill the ring-groove to compensate for wear. There may be rings of this type that function properly, but some of them at least fail to do so, and show very uneven bearing surface against the cylinder-wall.

The point I wish to make is that it is possible to produce a plain one-piece ring with very smooth bearing surface, preferably ground, carefully fitting them in the piston grooves. I prefer to use narrow rings on account of their lighter weight and their lessened tendency to hammer the piston groove and widen it. It is rather difficult to make an accurate determination of the relative values of these different types of ring even in the laboratory, unless rather elaborate preparations are made beforehand. One method that I have used very successfully consists of the following equipment:

A 100-gal. tank, filled with water, is placed alongside the engine being tested. A tube running from the top of the tank communicates with the crankcase. The breather of the crankcase is closed tightly, the engine is started and the water is allowed to run out of the tank just fast enough to indicate the absence of any pressure in the crankcase. In other words, all of the gas that flows by the piston and enters the crankcase is displaced into the tank as rapidly as it is formed, by allowing the water to run out of the tank as explained. By noting the time required to fill the tank with vapor and running similar tests with different types of piston-ring, a very good idea is obtained as to the relative efficiency. I have used this equipment to study piston design, as well as to compare different types of lubricating oil, and, while my experience has related mostly to the development of eight-cylinder engines, I believe that the above testing procedure would be just as useful in testing any type of engine.

In connection with the piston-rings used with the aluminum-alloy pistons, I have obtained the best results with narrow rings of deep section that fit the grooves in the piston closely and have 0.0005 to 0.0010-in. clearance. I prefer the narrow ring because there is a decided tendency in such a piston for a wide or heavy ring to peen and widen the groove at each end of the piston stroke. This action is more pronounced with loosely fitted than with tightly fitted rings. One manufacturer has gone to the extreme in this direction, and is using rings 3/32 in. wide.

One reason that excessive ring-wear is noted in certain engines is the rough cylinder-walls produced in some of the heavy-production shops. We believe that a cylinder surface should not only be ground with great accuracy, but be finished to absolute smoothness. This we accomplish by using very fine finishing-wheels with wide faces, and taking plenty of time for the operation. Each cylinder-bore is ground five times. The final operations are little more than polishing operations. It requires nearly 1 hr. to finish a cylinder block in this manner, but the results are well worth while. I repeat, therefore, that one of the reasons that piston-rings are short-lived is that they are used as laps to smooth-up the rough bores of some of our modern automobile engines.

C. R. MANES:—There is nothing wrong with the one-piece ring as long as it lasts. With a perfect groove and perfectly fitted rings no claims of efficiency for the multiple-piece ring over the plain ring can very well be established. The trouble is that the accuracy required to produce perfect grooves and fit rings perfectly to them is very difficult to obtain on any modern production job. Replying to Mr. Lítle's query as to how I determine that a so-called pressure-proof ring is better than a plain ring, I will say that if a man comes to me with an engine using from 5 to 10 times the oil it ought to use and I cure that engine permanently within a short time and at small expense by using pressure-proof rings, I surely have produced good results. If I can reduce his oil consumption one-half, one-third, or even one-twentieth, I have gotten results. We do not always get perfect results, because there are many types of engine and different conditions to be met, but in 95 per cent of the cases we know we get more than sufficient improvement in the operation of the engine to justify the expense to the owner. We do this with worn ring-grooves after the plain ring has ceased to function properly and the cylinder-walls are in many instances badly worn. In more cases than not we give that worn-out job a longer period of more perfect operation than it had before we put in pressure-proof rings.

Mr. Lítle cannot have made a study of pressure-proof rings. They are not difficult to install, they require no fitting, they do not break in use and service-stations do not report excess breakage, although breakage of plain rings both in installation and in use is common. It is possible to produce a plain one-piece ring with a smooth bearing surface and to fit it to the piston grooves carefully, provided the groove is not machined oversize, but this is a difficult operation in production jobs and one not likely to be obtained. After you have accomplished this difficult task to secure perfect operation, how long does this state of perfection last? It can be stated truthfully that with each mile the engine runs there is less perfection, while with a pressure-proof ring exactly the reverse is the result, as the ring will improve with use instead of deteriorate.

A MEMBER:—It seems to me that the best argument for the ring of narrow width has been missed. The thing that causes the groove to wear more than anything else is the force of the ring bearing against the groove at high speeds, due to the inertia of the ring, or its pressure against the side-walls; generally, the two operate together, causing the ring to slap from one side of the groove to the other. With a narrow ring, these forces are reduced very much more in proportion to the area of contact of the ring on the side grooves. From those considerations, the pressure caused on the side of the groove, due to the slapping of the ring in it, would be very much reduced per square inch; hence, the wear of the groove would be lessened to a very considerable extent. I believe also that the pumping of oil is a matter of inertia. Mr. Bull stated that an engine pumped more oil when the cylinder-head was removed than with the cylinder-head in place. The thing that is not taken into

consideration is that the pressure of the gases on top of the piston-head and their attempt to seep past the ring during at least two strokes of the cycle reduces that oil-pumping considerably. When the cylinder-head is removed, there is no pressure on top of the piston; hence, the oil-pumping goes on at every stroke, instead of at every other stroke. I believe that accounts for this part of the difficulty. The result of our experience has been that we get better results from four narrow rings than we do from three wide ones. They stay in the groove longer and last very much longer in the engine. I believe that the matter of workmanship on the ring has more to do with its operation than does the particular design. If the ring fits the groove properly at the start, there cannot be a hammer-blow on it because it has no space to move in until it begins to cut one.

MR. MANES:—The one-piece ring cannot be fitted very tight in the groove. It must have reciprocating space when the piston is cold, and that allows the clearance in the groove to increase more rapidly than if the ring were fitted closely. It is true that if the ring is fitted carefully, has deep enough grooves and a good wearing surface, it will wear a long time. In former days we had no trouble until an engine was 5 or 6 years old; but we had hand-fitted rings at that time. We had no 1/8-in. groove in a 3½-in. piston. We used a deep groove and the results told; but, with the present method of quantity production and the careless way in which the rings are installed in the piston, we certainly get rapid groove-wear and we pay for it by buying oil for our engines. We make the grooves apparently at random today, in most pistons, and that is where the trouble lies.

MR. DAMON:—Under some conditions, we use two-piece rings of the pressure-proof type and under other conditions, with the water-cooled engine, for instance, we use the 1/8-in. ring. We find we can get good results from both; but, in the majority of cases, we find that the two-piece ring will last longer and give more economical service for a greater length of time than a ring that is not ground to fit the groove. The practice we maintain is to equip every piston that is sent out with rings ground to fit the groove. We have yet to find a single-piece ring that we do not need to buy oversize and grind down to fit the grooves, on account of the warping that Mr. Bull mentioned.

J. E. DIAMOND:—Mr. Bull speaks of the friction set up by what are erroneously termed slotted pistons. In the partial circumferential separation of the head and the skirt resides the real merit of any and all of these newer types of aluminum pistons. As a matter of fact, in at least the case of one type of free-skirt piston the theory of differential expansion mentioned by Mr. Bull and featured in the elliptically machined piston was utilized a year in advance of even the conception of the latter. In this particular case, however, in addition to whatever benefit that might be derived from this differential expansion, was the suitable provision of circumferential skirt-expansion with mechanical construction that must be accepted. Mr. Bull's experience has been happy in comparison with others who have had an intimate association with the so-called constant-clearance type, which designation is a misnomer because the clearance is not constant.

I have been doing considerable work with one company employing this type, a company turning out very accurate work. Invariably, after every block test it is necessary to touch up each individual piston in varying degrees, and in many cases this operation is necessitated after a second or third test. In no two cases do the pistons show up the same. This surely is not particularly commercial. Furthermore, these pistons show excessive wear in service, and replacement is necessary almost invariably between 10,000 and 15,000 miles, which is much too frequent. A piston of the type advocated by Mr. Bull certainly does not have universal application as far as we have gone with it.

In connection with the determination of temperature, it was my intention merely to indicate the attempts that were made in the early days. Anent Mr. Jehle's remarks, it is the piston-ring that turns the trick with reference to the dissipation of heat. I am familiar with a series of experiments with a ring of unusual construction showing indisputably how important the bearing ring is in the matter of absorbing the heat from the piston-head and transferring it to the cylinder-wall. Circumferential slot or slots have very little or nothing to do with the matter of heat interference.

I am sorry that I apparently did not make clear the reasons for my advocacy of the narrow ring, since one of the speakers suggests that the best argument for its use has been missed. I agree entirely with him and regret I did not express myself clearly on the point. I also agree with his remarks on the subject of oil-pumping.

BRITISH PRIZES FOR AERONAUTIC PAPERS

THE Royal Aeronautical Society announces two prizes for papers on aeronautics. One of these amounting to 25 guineas (approximately $110 at the current rate of exchange) will be awarded annually for the best technical paper on aeronautics, preference being given to those dealing with an airship subject. The money for this prize will come from the society's R-38 Memorial Research Fund. The interest of the Usborne Memorial Fund will be devoted to the award of a prize in every alternate year unless the amount be such as to permit an annual prize to the value of £10 (approximately $44 at current rate of exchange) for a historical paper on any aspect of aeronautics. Both of these prizes are open to international competition. Details regarding the conditions of these competitions can undoubtedly be secured upon application to W. Lockwood Marsh, secretary of the Royal Aeronautical Society, London, England.

A MODERN PARISIAN GARAGE

A PUBLIC garage that embodies a noteworthy innovation in the arrangements made for accommodating cars on the upper floors has recently been completed in Paris. On the ground floor, cars can be driven in or out without maneuvering and are parked obliquely along the sides of the building in loose boxes which can be locked.

An electric elevator is used to take cars to the upper floor and to avoid maneuvering of cars to get on this elevator, a turntable is used. On arriving at the upper floor the car is driven off the elevator onto a sliding platform that is placed opposite the elevator and is carried to its own parking space. All these operations can be performed so quickly that as many as 150 vehicles can be moved in an hour and congestion at rush hours can be avoided.—*Omnia.*

Discussion of Papers at the Semi-Annual Meeting

WRITTEN discussions contributed in some instances by members who could not be present at the Semi-Annual Meeting and in others by those who were unable on account of the fullness of the program to participate in the discussion, supplement the remarks made at White Sulphur Springs. Efforts have been made in every case to have the authors of the various papers reply to the discussion, both oral and written, and these comments where received are included in the discussions. For the convenience of the members a brief abstract of each paper precedes the discussion with a reference to the issue of THE JOURNAL in which the paper appeared, so that those who desire to refer to the complete text as originally printed and the illustrations that appeared in connection therewith can do so with a minimum amount of effort.

The discussion of Principles of Motorbus Design and Operation, by G. A. Green, was printed in the August issue of THE JOURNAL, the paper itself having appeared in July, and the discussion of Fundamental Characteristics of Present-Day Buses, by R. E. Plimpton, was also published in the August issue together with the paper itself. In addition to the discussion given below, that of the Progress of the Research Department, by Dr. H. C. Dickinson, is printed elsewhere in this issue of THE JOURNAL together with the paper. The discussion following the presentation of the paper by F. C. Mock and M. E. Chandler, which was printed in the July issue, was not received in time for inclusion in this issue of THE JOURNAL. The papers by C. T. Coleman entitled Fuel Volatility and Its Effect on Motor-Vehicle Performance and by W. S. James describing the apparatus for measuring the road performance of engines have not been received as yet, both of these papers having been presented in abstract form only at the Summer Meeting. The discussion following these two papers will, it is expected, appear in an early issue of THE JOURNAL together with the papers.

OIL CONSUMPTION

BY A. A. BULL

THE object of the paper is to consider some of the fundamental factors that affect oil consumption; it does not dwell upon the differences between lubricating systems. Beyond the fact that different oils apparently affect the oil consumption and that there is a definite relation between viscosity and oil consumption, the effect of the physical characteristics, or the quality of the oil, does not receive particular attention in the paper.

The methods of testing are described and the subject is divided into (*a*) the controlling influence of the pistons, rings and cylinders; (*b*) the controlling influence of the source from which the oil is delivered to the cylinder wall. The subject is treated under headings that include the piston-ring; the effects of oil-return holes, side-clearance and ring motion; thin rings; influence of piston fit; efficiency of the scraper ring; ring and cylinder contact; carbonization and spark-plug fouling; oil-supply control; influence of oil viscosity; effects of dilution; external oil-leaks and breather discharge, and influence of controlling lubrication in proportion to throttle opening. [See June issue.]

OIL-PUMPING

BY GEORGE A. ROUND

OIL-PUMPING is defined and its results are mentioned. The influence of varous operating conditions is brought out, particular reference being made to passenger-car service. The factors that control the rate of oil consumption are described in detail and some unusual conditions are reported. Various features of piston grooving and piston-ring design are mentioned and the effect of changes illustrated. The relative advantages of the splash and the force-feed systems as affecting the development of oil-pumping troubles are set forth and improvements suggested. A new device for reducing oil-pumping dilution troubles is described and illustrated. [See June issue.]

THE DISCUSSION

F. F. KISHLINE:—Were the tests of the device illustrated in Fig. 4 of Mr. Round's paper made on a dynamometer with the engine running near its maximum capacity, or were they made on the road? I have in mind particularly the heat available for cooking the fuel. This might be very low when the engine is running at a low capacity, at which time it has been my experience that the greatest amount of oil-pumping takes place.

G. A. ROUND:—A majority of the tests were made on the road. A number of samples were sent to us from tractors operating in the field on kerosene.

A. L. CLAYDEN:—Has Mr. Round made any tests with the type of piston that is now being produced in large quantities by the Aluminum Manufactures, Inc., in which the ends of the wristpin have considerable clearance and there are two large slots instead of the usual row of holes? Of course, that is not applicable to the suction device; but it is a type of piston that is used extensively now. Has Mr. Round any figures as to how that compares with the more conventional type?

MR. ROUND:—Since the paper was prepared we have received several reports of excessive oil consumption with that type of piston. We believed at the time and still feel that this type of piston will help materially, but where the amount of oil that is going to the cylinders is excessive, we believe that this type of piston will not eliminate oil-pumping. This should be controlled by reducing the oil supply.

H. M. CRANE:—This paper has brought out a point on the question of oil consumption that I do not agree with at all. It states practically that we have to throw

very little oil upon the cylinder-walls or take the consequences. I do not think that is the case.

When we first used pressure lubrication, the thought occurred to me that is brought out here, that is, what will happen after the car has been in use for some time and wear and tear has taken place in the bearings? The answer to that seemed to be to flood the bearings and give the oil a free exit. That was done by providing a circumferential groove in the bearing lining, not in the crankshaft, that allowed a free exit from the oil-hole at all times, and by using 1/8-in. shims that did not come anywhere near touching the bearing; in other words, when the engine was stationary in any position, oil could be pumped through the bearing at a rate depending almost entirely upon the oil-hole in the crankpin. It is evident that the amount of oil would be very large, especially as the crankshaft is a very useful oil-pump; in fact, it is the main oil-pump in most pressure-feed systems, due to centrifugal action, the regular pump being used to keep the crankshaft filled with oil.

Different cars that we have built, depending upon the bore of the cylinder, stroke-ratio, crankshafts and bearings, have a pressure on the crankshafts of 2.5 to 25.0 lb. per sq. in. The lubrication is usually adjusted after the completion of the engine by determining the proper pressure to accomplish the required result. Under these conditions cars have been able to run under the same oil pressure in New York City at high speed, in European touring, and in the Alps, with no adjustment for throttling or for varying load. The reason is that the crankshaft, being a centrifugal pump, gives more pressure at high speeds than the direct ratio of power to speed.

With this system the use of return-grooves with holes under the upper rings seemed to be useless, it being difficult to determine any change that was due to the use of such grooves and holes. Sometimes it seemed that more oil went outward through the grooves and holes in the piston than went the other way.

The fundamental thing in the trunk piston is the crosshead. In our experience the length of the crosshead remains approximately the same as the diameter of the piston; with a piston-pin in the center of the crosshead and the walls of the crosshead parallel, we get excellent results. In passenger-car work the use of cast-iron pistons having a clearance of 0.0010 in. per in. of diameter or possibly a little more, gives satisfactory results; for full power at high speeds, up to 0.0015 in. per in. of diameter may be necessary.

Two additions must be made to the crosshead. The first is a set of rings at the top, usually three. The top flange is cut away until it will not bear. The second is a ring at the bottom. A land below this ring holds it in place, but it is cut away to such a point that a perfectly free release is left for oil. The cylinder-bore is designed so that in the downward position of the piston the lower ring overruns the bore slightly. This ring pushes the oil down. In a good set of rings the top ring will help to push the liquid fuel up.

Another thing about this type of piston that was wholly unsuspected and accidental is that an oil-film is trapped between the two sets of rings that seems to allow extreme piston clearances, with no piston slap. In aviation practice the pistons appear black after long periods of running, indicating no metal-to-metal contact.

A tapered piston, or one that is too short, does not give a ring a fair chance to do its work. With this type of piston, however, the simplest form of concentric ring of moderate width performs every desirable function, as soon as the ring approaches a bearing on the walls.

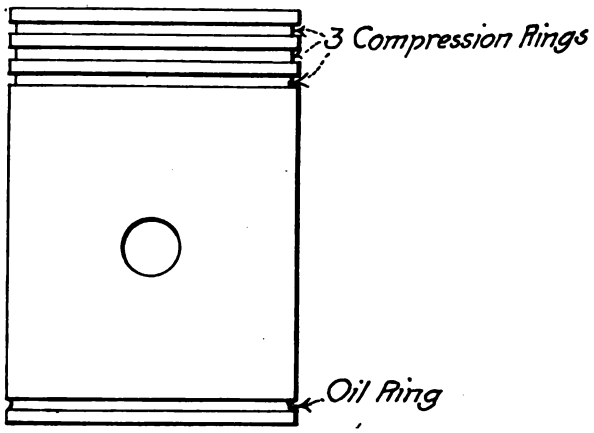

FIG. 1—PISTON EQUIPPED WITH THREE COMPRESSION RINGS AND AN OIL RING

Some time ago I installed in a car a set of 4 3/8 x 6 1/4 in. aluminum pistons. These pistons were of the general form already described and had a diametral clearance from 0.008 to 0.001 in. We were able to run with 50-per cent higher oil pressure with this clearance than previously when not using the lowest piston-ring. After running about 40,000 miles, although the performance of the car on the road was entirely satisfactory, the oil consumption very low, the formation of carbon in the cylinders moderate, and the porcelain of the spark-plugs white, we pulled the engine down and measured the cylinders. They were worn to the extent of 0.009 in. in some cases, tapering from the bottom to the top, yet, so far as operation on the road was concerned, they appeared to be satisfactory. The aluminum pistons were replaced by cast-iron pistons of the same design and, barring smoother operation of the engine, it was very difficult to tell the difference. With cast-iron pistons we use an oil pressure of over 20 lb. per sq. in., with a free flow from the crankpin, as has been described, and a crankshaft with three bearings. I cite that simply to show that it is possible to control the oil that is passed by the cylinders if the piston and the ring design is what it ought to be, and that it can be controlled successfully with cylinders that are badly out of shape and with ordinary rings. We do not need trick devices or peculiar joints, or anything of that sort, to get what we want.

Almost half of the piston-ring companies are specializing in special joint construction. If you will figure what the gap in the joints of these rings must have been at the end of 40,000 miles with cylinders 0.009 to 0.010 in. oversize, with normal wear, I think it will be apparent that patented-joint construction is of no use whatsoever and usually ruins a ring that otherwise would wear in a satisfactory manner.

MR. ROUND:—I have had no personal experience with the exact type of piston Mr. Crane describes. It may be that the deciding factor in this design is the relief of the piston below the skirt ring; for, with pistons otherwise similar in design and proportion, we have observed a number of cases of excessive oil consumption. It would seem that the use of this type of piston would be restricted to low-speed engines, because the increased length and weight would be seriously objected to by designers of high-speed engines.

W. L. DEMPSEY:—The proper solution of any problem depends primarily upon an analysis of the cause or causes producing certain results that it is desirable to change or prevent. In the last analysis, there can be but one *efficient* cause for oil traveling from the crankcase into

the combustion-chamber of a vertical automobile engine; it must be acted upon by a force in the direction of the combustion-chamber, and this force must be great enough to overcome the friction of the passage of the oil between the cylinder-walls, rings and piston. In addition, it must be great enough to overcome the force of gravity.

It is evident that this energy must be generated either by the engine itself or proceed from some source outside the engine. It is evident, also, that no oil can enter the combustion-chamber during the explosion stroke, because of the excessively high pressures that are exerted in all directions. It is equally evident that no oil can enter the combustion-chamber during the *latter* part of the compression stroke. It is very doubtful if any oil passes into the combustion-chamber during the exhaust stroke, although there may be certain conditions by which a vacuum is created during the exhaust stroke.

We must consider, therefore, only what occurs during the suction stroke of the engine. Let us examine, rather closely, whether there is anything in the construction of the piston, rings and cylinder that would warrant us in believing that oil-pumping is due to energy developed by the engine. First, there is ever present the law of gravity, which tends to carry any oil adhering to the piston or the cylinder back to the crankcase. This tendency requires an actual force to overcome it. If the viscosity of the oil is great, it tends the more to adhere to the spot on which it is thrown by the revolving crankshaft and dippers. This also requires force to move the oil either up or down. Is there any reason to believe that a well-designed piston and ring would carry more oil up against pressure and gravity than it would bring down? I think not. If there is any inherent tendency in the piston and the rings to force oil into the combustion-chamber, why does it not have a like effect upon the unburned fuel that causes so much crankcase dilution?

So long as the piston and the rings are fitted with clearance enough to permit a film of oil between them and the cylinder-walls, there is sufficient space for air and gas pressures to act upon the oil in either direction. Is it possible for the piston ring to carry up in one stroke out of four more oil than would be cleanly burned at the high temperatures existing during the period of the working stroke? It must be remembered that the force exerted by the revolving crankshaft, the connecting-rods and the dippers is always tangential to the axis of the piston travel, and never parallel to the walls of the cylinder and piston.

Mr. Round enumerates a number of factors controlling oil-pumping, but to my mind the one efficient cause of oil-pumping is the last factor stated in his excellent paper, namely, "vacuum in the cylinder"; or in other words, the force that causes oil to flow upward, against the law of gravity, into the combustion-chamber is the difference between the pressure of the atmosphere outside the chamber and the pressure of the gases within it. The examination of tables, giving the flow of air through an orifice under the influence of vacua of different degrees, may throw considerable light upon the subject.

For the last 3 years the company with which I am affiliated has been equipping cars with an automatic check-valve inserted in the cylinder at the limit of the downward piston-travel. We found that so much vacuum existed at the end of the suction stroke that a valve having a 7/32-in. opening opened with such violence that it was necessary to make the valve and the stem integral and of nickel steel to prevent the head from hammering off; and to deaden the sound caused by the rapid opening and closing of the valves, it was necessary to use a compensating spring or fiber washer. Nor is there any speed at which the valves do not open; hence, we conclude that the vacuum at the end of the suction stroke is more than sufficient to account for oil-pumping.

Mr. Round refers to an engineer who succeeded in reducing oil-pumping by maintaining a vacuum in the crankcase, equal to that in the intake-manifold. His paper also illustrates a device, in Fig. 4, that is intended to create a vacuum in a groove around the piston when it approaches its lower dead-center.

Tests that I made on a stationary engine, having a 4-in. bore and a 5-in. stroke, with a light spring and an automatic intake-valve, showed a vacuum, when turned by hand at a rate of 50 r.p.m., of 15 in. of mercury. One-half as much vacuum would be sufficient to pump more than 2 lb. of oil into a six-cylinder engine in a 10-hr. run, if the flow of the oil were not impeded by closely fitting piston-rings. I conclude, therefore, that the principal if not the only factor causing oil-pumping is the existence of low pressure in the cylinder at the end of the suction stroke, supplemented often by a pressure in the crankcase higher than that of the atmosphere, especially in engines having insufficient breathing area.

If we are to cure oil-pumping, let us lessen the vacuum at the end of the suction stroke and at the same time introduce the largest quantity of pure air possible by inserting an automatic air-valve that will unfailingly supply the cylinder with additional air, especially at low throttles, to burn completely and cleanly whatever oil passes into the combustion-chamber.

Mr. Round brought out the fact that they had more trouble with engines pumping oil on the seaboard than in the Rocky Mountain district; that tractors give less trouble than passenger cars; that passenger cars give less trouble when touring than when running in towns and cities. Does not each of these statements bear out the fact that the force which drives the oil into the crankcase is the weight of the external atmosphere? At sea-level the pressure of the atmosphere is 14.7 lb. per sq. in., while at great altitudes it is much less; consequently, there is more pressure or force at sea-level to drive the oil into a vacuum. When a passenger car is touring there is less vacuum, because the throttle is open wider, but when the same car is running in the city, it runs under an almost closed throttle and the vacuum is greater. A tractor when moving without a load requires one-half its engine power for self-propulsion; hence, the throttle is wider open and the vacuum is less than that of a pessenger car at low speeds.

The remedy, therefore, seems to be to lessen the vacuum in the cylinder, at the end of the suction stroke, and by ample breathing areas to prevent crankcase compression.

MR. ROUND:—I believe that the vacuum in the engine cylinder is in part responsible for the passage of oil into the combustion-chamber, as Mr. Dempsey claims. However, the fact that carbonization, which is a result of oil-pumping, gives less trouble in engines working under fairly heavy loads, is due also to the greater volume of fuel burned, the latter being adequate to consume cleanly much larger quantities of oil than is the case with lightly loaded engines.

J. F. WINCHESTER:—The questions of oil-pumping and oil-consumption are so closely related that the papers by Mr. Round and Mr. Bull can be covered in the same discussion. The experiences of these two men, in some ways, check up with the general experiences that I have had in the last 15 years, 10 of which have been closely

associated with various types of oil complaints submitted to one of the large oil companies, but, as a part of the discussion, permit me to present the following:

Mr. Round states that crankcase-oil dilution can be offset in two ways: (*a*) by the periodic draining of the entire supply; and (*b*) by the frequent addition of fresh oil to replace that used. To my mind, the solution of crankcase-oil dilution, from a trouble-preventative standpoint, is the periodic draining of the oil supply, depending upon the number of miles run. This recommendation should be made by local service-men, who understand the type of engine under consideration, for the length of time after which draining will be needed depends upon climatic conditions. In some cases the frequent addition of oil did not overcome or tend to relieve the trouble in cars and trucks that I had under my observation, for the reason that the dilution was in such proportion that it equalled the daily consumption of oil. Therefore, the addition of fresh oil tended to flood the engine.

Mr. Round states that ring end-clearance is of minor importance in controlling oil consumption. This statement does not check with my experience. It is my belief that ring end-clearance is a very important factor in controlling oil consumption. In the early days ring-fits were based largely upon steam-engine experience where wide clearances were needed. It is generally conceded that this practice is no longer accepted; that definite recommendations are made by the designers of various types of engine; and that definite clearances are insisted upon in the better-built products. The end-clearance of the ring, to a large degree, depends upon the clearance of the piston and upon the type of work on which the engine is to be used. On the average American engine it will be found that much better results will be obtained from the standpoints of both oil consumption and carbonization if all the rings below the upper ones are fitted with less than 0.003-in. end-clearance. It has been my experience that end-clearance controls the amount of oil that will accumulate under the ring, and has an important bearing on the amount that will be pumped into the combustion-chamber.

I have found that the amount of oil consumed is controlled to a large degree by the ring-pressure. In a number of cases where complaints have been made that certain types of lubricating oil were entirely unsatisfactory for a given engine, investigation has proved that the trouble was eliminated by employing piston-rings that had the proper tension and were correctly fitted. Mr. Round states that, in his experience, 2 lb. per in. of diameter for a ring of ½-in. width is the tension that gives the best results. My experience, which apparently checks with that of Mr. Bull, based on Table 3 in his paper, that gives ring-pressure values, indicates that a 50-per cent greater pressure, or 3 lb. per in. of diameter of ring, is the proper tension. There have been a number of cases where a ring-pressure of 2 lb. per in. caused trouble, whereas 3 lb. per in. for the respective sizes has been very satisfactory.

Attention has been called to the non-interchangeability of the rings manufactured, and to the indifference in the inspection given many rings before they are placed on the market. This experience checks with mine. I know that very many complaints of crankcase dilution and excessive oil-consumption are brought about by (*a*) the manufacturers missing this important point, and (*b*) the patent-ring manufacturers laying great stress upon the features of construction that they can visibly demonstrate to the buyer, yet neglecting to manufacture the articles within close limits. My experience agrees with that of Mr. Crane who stated that the majority of so-called special piston-rings on the market do not give as good results as the plain concentric ring. Mr. Bull believes that the two-piece ring is the most desirable.

Little stress is laid upon the oil-control groove in the piston-ring test. I have found that with the type of ring shown at the right in Fig. 1 the passing of oil into the combustion-chamber is controlled very satisfactorily. Mr. Bull states that the type of piston-ring having an inherent twist, so that the unit pressure on the bottom edge is in excess of that on top, may be beneficial. It is appreciated that rings of this type can be manufactured, but there is a serious doubt in my mind whether this theory will work out in practice. If a ring of this type has proper clearance in a groove, this inherent twist is controlled. Therefore, it fails to come into action. I have tested a large number of rings of this type and fail to see wherein they give better results than a properly fitted plain ring.

Considerable stress is placed upon the operation of the lubrication system. There is no doubt that, if a force-feed system becomes worn, there is a decided tendency for an excessive amount of oil to be thrown into the piston and to find its way into the combustion-chamber in various ways. This difficulty is particularly noticeable in so-called high-speed engines operating in city traffic. Under these conditions, a drop in the original pressure becomes necessary. As brought out by Mr. Round, the average manufacturer or service manager is inclined to insist that the original pressure recommended in the instruction book supplied by the manufacturer be adhered to. Careful instructions covering this point, by various manufacturers to their service-stations, would result in many of the present-day difficulties being overcome. Illustrations of the variations of individual spring-pressure furnished with pressure-feed systems, as brought out by Mr. Round, coincide with those of my experience. It is surprising to find the number of manufacturers of high-grade equipment that overlook this important point by failing to calibrate properly the tension of the springs of the lubrication system.

Mr. Round advocates the positive control of oil pressure under varying loads. I agree with him on this and submit that the best possible control is that which is connected with the throttle.

The question of employing auxiliary devices to control crankcase dilution or oil-pumping is naturally one that requires research work. But this difficulty could be taken care of properly by the further improvement of the oiling system of today, without the addition of auxiliary devices that tend to complicate the car's mechanism.

In the discussion Mr. Round advocated abandoning the use of shims. I believe this would work a hardship on the average American manufacturer. It would also result in increased up-keep cost in car and truck operation. The trouble that Mr. Round cites is brought about through a lack of education on the part of the mechanic. Therefore, education is what is needed, not a change in manufacturing methods or design. Neglect on the part of manufacturers to educate properly a suitable number of all-around mechanics is resulting in all of us puzzling over problems that we should never meet. The industry as a whole probably is suffering more from this cause than from any other.

My recommendation is that proper steps be taken to provide the future generation with a sufficient number of trained men to maintain economically the types of car being put out at the present time. We should not abandon present-day practice, which has accomplished eco-

nomical results from the standpoint of both manufacturing and up-keep, because mechanics have not been educated sufficiently to make the proper repairs.

Great improvement in oil-pumping and crankcase dilution would be effected if manufacturers, service-men and instructors dwelt more thoroughly upon the subject, and if the average garage-man were able to buy standard piston-ring gages and tension testers that would enable him to determine the quality of the product before it is applied. He should be supplied also, by the car builders, with detailed instructions regarding the fitting of shims and other parts that have an important bearing upon the results to be obtained after the car has been overhauled.

MR. ROUND:—Mr. Winchester's point regarding the overcoming of crankcase-oil dilution is well taken, but there are cars in operation in which oil is consumed so rapidly that replenishing the oil supply is sufficient to maintain the oil in reasonably good condition, thus offsetting any excessive dilution. This, however, is not the general experience.

Regarding the education of service managers and mechanics, too much emphasis cannot be given to this. Even today, we find branch-office service-managers, to say nothing of foremen and mechanics, who do not believe that the thinning out of crankcase oil is due to fuel dilution, and who show an equally meager knowledge of other fundamentals. A series of service bulletins to reach not only the managers but the individual mechanics as well, written in an interesting way, would be a great help along this line.

A. A. BULL:—I agree with Mr. Crane in reference to the necessity of presenting the ring squarely to the cylinder-wall. However, I hardly see how he can attribute to this feature alone the success of the construction to which he refers, because, with the wear and with the 0.009-in. taper of the cylinder that he found after use and under which conditions he states the rings and pistons function perfectly, it is difficult to understand how, under such conditions, the rings could remain square and parallel to the walls of the cylinder.

Referring to Mr. Winchester's comments, it is established almost beyond doubt that end-clearance is of no importance. If it were as important as he states, we would be in a serious position in respect to the use of a plain ring because we must recognize that, with the wear on the ring and the cylinder, the gap will increase very rapidly; and, under such conditions, we certainly would require something different from a regular angle or step-cut joint.

In the analysis of the piston-ring pressure, it is shown clearly that there is an advantage in increasing the pressure between the ring and the cylinder, but this in itself is of little avail if side-clearance in the ring occurs; for, while the ring itself may be more efficient in displacing the oil from the cylinder-wall, it cannot prevent it passing around the ring instead of through the oil-return holes that may be provided. Under such conditions of side-clearance, however, and with holes at the rear of the ring-groove, the passage of the oil can be prevented to a large extent.

It is true that the features of construction, which the majority of replacement piston-rings possess, visibly demonstrated to the buyer, rarely have any characteristics controlling oil-pumping, and the importance of accuracy in manufacture is often neglected. It must be recognized, however, that there are factors in the cylinder itself which may nullify the benefits of extreme accuracy in the ring.

I stated that a plain concentric piston-ring when properly installed is generally as good or better than rings having fancy constructions of joints; but, in the final analysis of the functions of a ring and recognizing the vital necessity of proper fit sideways in the groove, I advocate the use of a two-piece ring of a construction such as is illustrated by the third ring from left to right in Fig. 12, which has the ability to expand sideways to fill the groove properly. I have used rings of this construction made especially for us with aluminum pistons, for 3 years and cannot fail to recognize the fundamental necessity for a construction of this kind.

Regarding the reduction of oil pressure on pressure-feed lubrication systems, there is no doubt that this will help considerably after the bearings have become loose and it is to be recommended, but, as stated, the service-man usually endeavors to maintain the original pressure recommended, for safety.

It is to eliminate such conditions that the method of oil control from the crankpin is discussed at length. With the proper location of the oil-distributing hole sealed by the bearing throughout the greater part of the cycle, the oil discharged from the crankpin is not influenced appreciably by the bearing clearance. The main point is that the conditions existing when the engine is new should be maintained throughout its life, regardless of the wear that inevitably takes place.

With regard to positive control of oil pressure under varying loads, I think that the connection with the throttle is not proper, as it will not exercise any control over the oil consumption unless accompanied by the control of the oil discharge from the crankpin as outlined.

While I did not make any definite statements on the manner in which I believe crankcase dilution can be controlled, I am satisfied that this can be minimized to an almost negligible degree by the use of proper manifolds, preventing the entrance of large quantities of liquid fuel into the cylinder. The difference between manifolds in this respect can be demonstrated easily by providing suitable dams at the entrance to the cylinder and collecting the liquid fuel that ordinarily flows into the cylinder under low temperature and speed conditions. Further, the use of a choker for the carbureter that prevents sucking raw fuel into the manifold is necessary. The large differences in dilution that occur as between different engines can usually be attributed to such conditions.

NEW SYSTEM OF SPRING-SUSPENSION FOR AUTOMOTIVE VEHICLES

BY H. M. CRANE

THE author indicates what the history of spring suspension has been but discusses only the conventional type of four-wheeled design in which the front wheels are used for steering and the rear wheels for driving and braking. The problem of front-axle spring-suspension is mentioned, but that of proper rear-axle spring-suspension, especially for passenger cars, is discussed in detail because it is a much more difficult one.

The advantages of the Hotchkiss drive for shaft-driven cars and some of its distinct disadvantages are

stated, shaft-driven, rear-axle mountings being commented upon in explaining the factors that influenced the design of the spring-suspension device developed by the author. The advantageous features of this device are enumerated, inclusive of the effects of tire reactions. [See June issue.]

THE DISCUSSION

H. M. CRANE:—The history back of this spring suspension begins with a shaft-driven car equipped with distance-rods, a torque-arm and springs that were free to turn on the rear axle. This construction was followed by a practically similar construction in a somewhat larger car, about 1907. In the second construction we first had our attention called to the effect of the torque-arm in causing a reaction on the driveshaft, as described in the paper. An experiment we made there to ameliorate this condition consisted in providing a very considerable spring-resisted motion at the end of the torque-arm; that is, the fixed end on the chassis frame. Examination of this spring action showed that, over a comparatively smooth road, with the engine just pulling the car at about 25 m.p.h. on the level, you would occasionally bottom over 200 lb. of spring pressure on a torque-arm about 45 in. long. It gives a crude idea of the shocks that are produced in the driving mechanism by this effort of the rear-axle pinion-shaft to remain parallel to the gearbox shaft at the forward end. It is evident that any angularity that the rear-axle pinion is forced to take in relation to the fixed shafts on the chassis frame tends to result in an angular rotation of that shaft, or of some part of the driving mechanism, and the shaft is about the only thing to take it up.

I should have called attention in the paper to the fact that the use of fabric universal-joints, which is becoming very common at present, will go a long way toward eliminating the difficulties arising from this type of reaction; but the answer that we made to it at the time when it came up was to adopt the modified Hotchkiss drive. In other words, we clamped the rear axle solidly to the rear springs, proportioned the rear-spring arms so that we had a practically parallel motion between the axle and the frame, and used in addition distance-rods to align the axle and prevent it from going out of position. We also used a platform spring, overhung on the axle. That spring is equipped with a very satisfactory device for cushioning side-shocks in its cross-shackles, as I now know from having departed from it. As many of us have often done in attempting to improve, we actually arrived at an arrangement that was less satisfactory than the one we had been using previously.

Our next movement from this device was to the semi-elliptic spring hung under the frame, about 1912. The first cars of this type had very light frames and were driven almost entirely as open cars. The thing in this particular design which began to cause us to open our eyes was a breakage of the rear-frame member. The frame was designed at the rear end very much as at the front end, and we broke off the overhanging part of the frame beyond the last cross-member. That was due to the lateral vibration set up by these very flat springs that were stiff laterally, and partly to the twisting of the springs tending to twist this rear extension of the frame. However, we believed that by making the frame sufficiently strong there we would overcome this difficulty, and we proceeded to a larger car of 142½-in. wheelbase, using the same general layout of a flat semi-elliptic underhung spring at the rear. The riding quality in the rear seat of this car was very disappointing at the start. The difference was not particularly noticeable in the front seat, but there was a disagreeable lateral chattering effect that could be felt on the floor-boards and the cushions and showed up very plainly on the battery hanging. The battery was suspended under the rear floor in a hanging of fairly light weight and not at all rigid, and the battery was trembling sideways sufficiently to shake the whole floor. It was at that time that we introduced the spring cushioning on each side of the rear spring at the front end and the ball shackle at the rear end; that is, a shackle in which the spring end and the shackle connection were of the usual form, but at the other end of the shackle was a ball-joint attachment to the frame. This allowed a considerable amount of lateral freedom to the spring and at the same time reduced the torsional reaction of the spring against the frame when one wheel was lifted higher than the other. It was a fair compromise but not entirely satisfactory.

This car had a straight Hotchkiss-drive; in other words, there were no distance-rods. All of the connection between the rear axle and the frame was made through the spring; the driving torque-reaction, brake reaction and everything. We have found that the springs are entirely capable of doing this satisfactorily; that is, so far as spring breakage is concerned. To give figures, the usual flexibility of rear springs on this car was of the order of 165 to 170 lb. per in. of deflection for each spring. This is on a car that weighed, without passengers but otherwise fully loaded, from 5000 to 6000 lb. The spring, relatively, was of a very soft character.

In driving many miles with chassis of this type, and even driving with cars with bodies on, with the floor-boards up and watching the rear-axle operation, it became evident that there were other conditions there that were not being met. In other words, we had a very heavy axle tied to the frame in a vertical direction, with a considerable degree of flexibility; but, longitudinally to the frame, the springs being practically horizontal, there was almost no flexibility whatever. We ought to have realized this from the fact that we broke the front spring-eyes in a great many springs, in this case when we were using 1¼-in. bolts at the front end of the spring, the spring eye not being welded, but simply turned over and acting as a hook. We found it necessary to reduce the diameter of the bolt to 1 in. to make the spring stand up.

I began to feel that there ought to be some way of letting the rear axle accommodate itself to a certain extent in speed to the longitudinal-horizontal speed of the chassis. That had been done before and there was no question about it; the thing had been recognized as a desirable feature and, as I stated in my paper, there have been springs introduced in distance-rods, or the springs have been canted at an angle with the horizontal so that the rear axle would tend to move toward the rear of the car when the springs were compressed, and there have been a number of similar ways in which there has been an effort to overcome this defect that was generally recognized.

Some of the defects of the Hotchkiss drive have also been recognized. It is easier on tires, I am very sure, on ordinary roads, but we all know the effect that is produced on sandy, gravelly hills with the lightly loaded Hotchkiss-drive job; there is a tendency for the rear axle to jump clear of the ground and set up a vibration period in jumping and in rotation that necessitates an entire abandonment of any attempt to go up the hill rapidly, and simply necessitates reducing the application of power until it is just barely possible to crawl up the hill.

In thinking these various things over, I evolved the layout that is shown in Fig. 1. The first thought I had

was: Suppose we use a distance-rod but attach it at the level of the ground; that is, opposite the tread of the tire on the ground. Apparently that would practically eliminate the torque-reaction effect; in other words, we would transmit it directly to the frame through the distance-rod. At the same time I saw that if we attached the distance-rod at a point not at the center of the axle but at a point substantially below the center, it would be possible for the axle to travel longitudinally with respect to the frame to an extent permitted by the stiffness of the spring itself. The arrangement was first tried on a car of about 4200-lb. weight and, as an experiment, the point of attachment was placed at about half the distance from the center of the axle to the ground. In this job we were able to vary the position of attachment vertically so as to get a longer or a shorter lever-arm, and our experiments covered lever-arms rather more than half of the radius of the tire. The results of the device were two. First, there was a very noticeable and continuous variation of the position of the axle with relation to the frame in a horizontal direction on any kind of road, even a comparatively smooth, newly built highway. On a rapid reversal in a vertical direction, such as a "thank-you-ma'am" or a gutter in the pavement, there was a very violent and extended movement of the rear axle in the longitudinal direction and at the same time a very much reduced vertical throw of the body. I have tried to explain in my paper why I think that reduced vertical throw occurred; that it is due to the reduced tire-reaction. It is very difficult to tell exactly what is going on under these conditions, but I think that is the reason. As to the facts, there was no doubt whatever; there was a very great difference.

Another thing we found in an early application was that if the longitudinal stiffness was made too small by lengthening the lever-arm, it was wholly possible to obtain a serious periodic vibration longitudinally of the axle with respect to the frame on a washboard road; in other words, the axle would move backward and forward violently in a period determined by the stiffness of the spring reaction and the weight of the axle itself.

I am not convinced yet that we know exactly the best length of lever-arm to use with the arrangement. It must depend very largely on the weight of the axle compared to the stiffness of the spring and to the weight of the vehicle itself, but it is apparently possible to obtain a relation of parts that will practically never cause this unfortunate effect, and it certainly is most unpleasant when it occurs.

What I have just described is the new part of this suspension. I have called attention in the paper also to the lateral flexibility provided by swinging links. It will be recognized that this is simply an extension of what is provided in the ordinary platform spring-suspension. Our experience with various spring-hangings indicates that the higher the spring is above the ground, the less difficulty is experienced from the lateral vibration; that is, the very low underhung spring seems to affect the body very much more than the higher spring. But there are undoubtedly very heavy stresses put in with any height of spring, due to both the side action of the axle and the effect produced by attempting to raise one wheel higher than the other with respect to the ground. The torsional effects produced by this are very great and also the bending effect that must take place, due to the fact that the projected length between the springs when the axle is at an angle is less than the distance between spring-eyes on the frame, unless some means of accommodation is provided either on the axle or on the frame. I may say that the use of some arrangement of this type on front springs had a most noticeable effect in reducing the disagreeable rattle that we get in so many cars in the front springs. I take no credit for finding out this point in spring-suspension. It was called to my attention very emphatically by a chauffeur of one of our early owners. I did not believe him when he told me, but he promptly convinced me by rocking the car back and forth and allowing me to put my hand on the shackle. We found there was from 0.003 to 0.005-in. play in the shackle and yet the rattling from it was very unpleasant.

W. C. Keys:—What types of universal-joint and propeller-shaft were used and which types have been found best? Also, has Mr. Crane made a really fair comparison between this suspension and a very flexible full-elliptic suspension that is used on two or three cars with which we are familiar, and which gives fore-and-aft and side flexibility?

H. W. Alden:—There are two features of this spring-suspension that I wish to question Mr. Crane about. We used a somewhat similar construction in 1905 or 1906 in the Columbia car; it was not exactly the same but it had some of the elements. We had a radial arm on each end of the axle and a link running from there to the frame. The springs were overhung. The forward end of the spring was rigidly attached to the frame at a point some 4 or 5 in. below the point of attachment of the side-rod. What we were striving for was to get a sort of pantagraph action, so that the pinion shaft would remain substantially in a horizontal position all the time; in other words, parallel with the transmission shaft. We found that we got a rather more vertical whip of the universal-joint at the rear end than we did with the straight Hotchkiss-drive. Also, we developed another very serious defect that we tried to overcome, although it never was overcome satisfactorily. It was the varying angular positions of the two vertical arms on the ends of the axle, when the plane of the chassis departed from a position parallel to the central line of the rear axle; in other words, when the car rolled sideways. Our side-rods were very nearly in horizontal and vertical positions. In the construction that Mr. Crane uses, the side-rods are at a considerable angle. It would seem that when the car rolls sideways the side-rods in Mr. Crane's arrangement would exert a very considerable torsional effect on the body of the axle housing. In other words, if the car rolls to the left, the lower end of the left-side axle-bracket would be pushed back and the one at the other side would be pushed ahead considerably. We were continually shearing off those brackets on the ends of the axle. I would like to know whether Mr. Crane has had any of this same experience. It seems that the strains would be terrific at times, because the coupling is absolutely rigid and something would have to give.

I notice that one of the reasons Mr. Crane favors this construction is that it decreases the vertical whip of the rear universal-joint. It would seem from the diagram that, with the chance for the rear axle to surge forward and backward and with the two lower ends of the arm rigidly held with respect to the frame, this surging action would introduce a very considerable vertical whip of the universal-joint at the forward end of the rear pinion shaft.

G. W. Cravens:—In 1917, when I designed the Elcar, the question of the Hotchkiss drive came up and we finally adopted it; the car is now in its fifth season and no change has been made.

In Fig. 1 of Mr. Crane's paper, I notice that he has made the rear end of the spring higher than the front.

In riding a car, the discomfort of the passengers is greatest when the body has a front-and-back movement or tendency to change its rate of speed while traveling; in other words, if the body moves along smoothly at a uniform rate and almost without swing, it is not uncomfortable. When the wheels strike obstructions they tend to stop momentarily; then, through the rebound, they come back, which means that they are tending to move ahead with relation to the travel of the body. If the rear end of the spring is lower than the front, the bound of the axle or the tendency to jump backward does not pull the body back, as is the case when the compression of the spring makes it necessary for the axle to move forward, and that inevitably tends to retard the movement of the body. The question presents itself: Is the excessive flexibility, due to having everything on shackles, provided to overcome that condition; and, does it do so?

How does Mr. Crane feel in regard to using the propeller-shaft tube as the torque-arm and mounting the springs on the axle by spherical bearings or joints like those of the Lanchester car, so that the axle is free to move and the spring is free to move on it, with the torque or push of the rear axle transmitted to the body through a spherical joint at the front end, similar to that used on the Lafayette and some other cars in this Country?

W. W. WELLS:—It seems to me that the angularity of the drive-rod is an important consideration. When I read Mr. Crane's paper I immediately wondered what would happen if on our 1-ton truck we mounted the rear end of the drive-rod 9 in. from the center of the axle. We have a 550-lb. spring, and I find that a horizontal force of 9200 lb. applied to the axle would move it back 1 in.; that is, the change in location of the drive-rod would be equivalent to introducing a spring of 9200 lb. per in. in the drive-rod. I notice also in Fig. 1 of Mr. Crane's paper that the drive-rod stands at an angle of about 1 to 8, which means that an upward movement of the axle of 1 in. would cause a backward movement of $\frac{1}{8}$ in. Comparing these results shows that it would take an upward thrust of 1100 lb. to compress the spring 2 in. and shove the axle back $\frac{1}{4}$ in., due to the action of the drive-rod. A horizontal force of 1300 lb. would be required to rock the axle back $\frac{1}{4}$ in.; that is, if the direction of the blow on the axle is 45 deg., the backward movement of the axle in relation to the frame is due more largely to the vertical component of the force and the angularity of the radius-rod than to the horizontal component acting through the rocking action of the axle. Are any definite data available as to the correct angularity of the drive-rod or of the front spring?

MR. CRANE:—Answering Mr. Keys' question, the type of universal-joint used in the shaft of this arrangement at the rear end is the jaw type; that is, it has long, parallel jaws, in which rollers operate. There are two rollers in dumb-bell form on the ends of the propeller-shaft which are roller-bearing supported and the combination provides almost unlimited angularity with a very great freedom of longitudinal movement; the last being very desirable to keep excessive strains from being placed on the ball bearings either in the rear-axle pinion-shaft or in the gearbox.

I am very glad Mr. Keys spoke of the full-elliptic spring. I thought that I had done pretty full justice to this type of spring in my paper. There is no question that it is an excellent type for absorbing road shocks. The chief difficulty with it is that it is not satisfactory for absorbing torque reaction. This is due to taking it on only one-half of the spring-steel instead of all of it; in other words using the lower half of the full-elliptic steel spring to take the torque reaction as against the full spring in the semi-elliptic. I think that point is covered in my paper.

Replying to Mr. Alden's questions, I think an inspection of this device will indicate a complete dissimilarity from the one that he describes. I remember that old pantagraph arrangement very well. We considered it at the time but finally decided to use a modified Hotchkiss-drive instead. We discarded the Columbia device for exactly the reasons Mr. Alden gave in connection with it; that is, the extreme rigidity of the two sets of arms made it impossible for the axle to take an angular position with relation to the frame, or required that something should spring very considerably to allow the axle to assume such a position.

The fundamental difference in the two devices is that in the original arrangement described by Mr. Alden the front end of the semi-elliptic spring was rigidly attached to the frame, and in this device both ends of the semi-elliptic spring are shackled to the frame with relatively long shackles. There is, therefore, in the latter no very large strain set up by the axle taking an angular position with regard to the frame. Any such strain that is set up is simply spring-reacted, or partially taken care of by the swinging side-links. As a matter of fact, the car on which this was first used had the lightest rear-axle construction that I have known of being placed under a similar car. It had an aluminum center casting, carrying the gears, with axle tubes bolted to the casting on flanges of relatively small diameter. The tubes were parallel and simply flanged at the ends and, according to my recollection, they were not over $2\frac{1}{2}$ in. in diameter as regards the tube and not over 5 in. as regards the flange. That car weighed 4700 lb. without passengers and developed fairly high speed under the conditions in which tests were made, having an engine of from 80 to 90 hp. It never exhibited the slightest signs of distress in any of the axle parts; due to the loading from this device, the tubes did not loosen on the axle housing or show any tendency to do so, and they were not keyed but were simply fastened with fairly well-fitted bolts. I am well satisfied that, due to the spring reaction opposing such strains, there is no deleterious action produced in the axle itself and no excessively heavy construction is required to take care of such action.

As to the vertical whip of the universal-joint, in the ordinary Hotchkiss drive this is due to the straight torque-reaction; that is, if the brake is put on sharply, the front end of the pinion shaft ducks; if power is put on violently, the front end rises. Due to the fact that we reduce the torque reaction in this device, that is, absorb it directly by a change in the angular position of the axle, the vertical whip of the end of the pinion shaft is very greatly reduced. The difference is very noticeable. This device was substituted on a car for a typical Hotchkiss drive and back to a typical Hotchkiss drive, so that we could measure up very closely the relative action of the two arrangements.

In response to Mr. Cravens' question regarding the position of the rear springs, I realize that in a regular Hotchkiss drive it is not possible to place springs in this position, due to the fact that we depend on the position of the spring to make the axle travel in the correct relation to the frame as the springs are compressed. In this arrangement, however, the distance-rod controls this feature to an almost complete extent, and we therefore in this particular job took advantage of what we felt we would find by lowering the front end of the rear spring to keep it entirely clear of the floor-boards. The result is

that a body placed on the chassis illustrated in my paper shows an entire absence of obstruction in the floor from any part of the spring-suspension. There is nothing that cuts the floor at all except the ordinary wheel housings for the fenders. I am not absolutely convinced that this is the best arrangement as shown, but in all preliminary tests it has functioned fully as well as we expected.

I also explain in my paper the reason for overhanging the rear-spring, which made this tilted position necessary; it is the fact that the higher the springs are placed, the less rolling action there is. It is conceivable that if we mounted a sort of gallows-frame construction on a rear axle and hung the body on that, and located the springs above the body, the latter would actually swing out on a curve, rather than the opposite. There is no reason it should not do that, if it were hung at the top instead of supported at the bottom.

Mr. Cravens brings up the question of the torque-arm drive, which is a very common European practice. The Rolls-Royce has used it for a number of years; as he said, it is incorporated in the Lanchester and several other cars; it is employed also on the Buick, the Lincoln and the Lafayette cars in this Country. My objection to this construction is based largely on its excessive rigidity in driving, in torque reaction. If the torque-arm is long enough, there probably is no very serious trouble from the lack of parallelism between the rear-axle pinion and the gearbox and engine shafts. If the torque-arm is short or the springs are very soft, there is a great amount of this trouble.

I overhauled one of the old four-cylinder Fiat cars after it had had about 18 months of use in this Country, a number of years ago. There was not a single joint in the driving system of that car that was not shot to pieces; every key was loose in its keyway, and the cost of bringing that car back to its original condition was excessive. Further, it was a perfectly futile expense because it was an absolute certainty that the car would again relapse into a condition of looseness as soon as it was driven for any length of time on our rough roads.

I was told by a man connected with the American Locomotive Co. that they even had difficulty with the loosening of flywheel bolts on the old Alco car that used a very stiff torque-arm construction. It seems obvious to me that, in a job where we want cheaper service, the softer we can make all reactions, the less danger we have of shaking something loose, either by road conditions or by rather rough operation on the part of careless drivers. I would say also that my chief objection to the cantilever spring is that it requires the use of a torque-arm construction. On the fairly smooth roads in Europe, that is not important; or if you use a very stiff rear spring, it is not particularly important. There is another practical disadvantage in the cantilever spring that is one of my pet aversions; it is the difficulty of mounting bodies with that arrangement. The overhang from the last point of support of the spring-suspension in an ordinary cantilever spring toward the rear of the car is extremely great. I have been dealing myself with large cars, and the weight figures are something of this order: We have two tires and a tire carrier at the extreme back of the chassis frame something like 5 ft. from where a cantilever-spring pivot would be placed. These tires and tire carriers together weigh more than 150 lb. We have a 24-gal. gasoline-tank that, with all its fittings, probably weighs 200 lb. This is not so far back as the tire carrier, but it is far enough. In addition, on a long body, there are three passengers on the rear seat who are nearly as far back as the tank; they may weigh from 500 to 600 lb. Due to the position of the rear door in any type of body, and because of other considerations, it is a very difficult matter to make the frame stiff enough to carry this load properly. It can be made strong enough, there is no difficulty about that, but all of us who have had much work to do with closed-body mountings on the chassis realize that strength in a body frame on a passenger car does not mean anything, but stiffness is the one necessary feature.

Mr. Wells brings up the question of the angularity of the distance-rods shown in Fig. 1 of my paper. We have not made any very considerable tests on this particular layout by varying the angularity. What we have attempted to do is to cause the ball end of the distance-rod attached to the axle to rise and fall substantially in a vertical direction with respect to the axle. Of course, there is always a radius due to length of the distance-rod, but the radius is approximately normal to the axle. This drawing may be a little misleading, due to the fact that it is shown with the spring not very heavily loaded. As an actual fact, the fully bottomed effect occurs when the axle tube, which can be seen in Fig. 1 with a rather small diameter, bottoms on the frame. If you will trace the motion of the distance-rods, you will find it is not exactly a vertical one; it is inclined slightly to the rear, about the same amount that we would incline a Hotchkiss-drive spring attachment of the normal type.

There is no question that the angle of tilted springs, where the rear end is lower than the front end, does help considerably; but it is evidently only a compromise that will be possibly correct for a given loading and speed of the chassis and a few other conditions. It is better than nothing; but, so far as our experience goes, it does not go nearly as far in ameliorating the difficult conditions that we have to meet as the arrangement shown in Fig. 1.

A METHOD OF DEVELOPING AIRCRAFT ENGINES

BY CAPT. GEORGE E. A. HALLETT, U. S. A.

THE general method of procedure taken by the Air Service before beginning the actual design and construction of the necessary types of aircraft engine is outlined and the four steps of the development subsequent to a very complete study of existing domestic and foreign engines are stated.

After checking over the layouts, if all the details are agreed upon by both the designer and the Engineering Division, the contract is placed, usually for two experimental engines, and the construction work is begun.

Acceptance tests are made to demonstrate that the engine is capable of running at normal speed and firing on all cylinders. These are followed by the standard performance test made on the dynamometer at McCook Field. The results of the latter test determine whether the engine can enter the 50-hr. endurance test. The engine is then torn-down and inspected for wear. Suggested modifications are embodied in reconstructed engines which eventually fulfill the requirements. Descriptions of the various tests are given and commented upon. [See June issue.]

THE DISCUSSION

PROF. E. P. WARNER:—The paper prepared by Captain Hallett and read by Mr. Jones seems to me to be pecu-

liarly important, first directly because of its explanation as to the method of developing engines at McCook Field, where they are certainly doing more development work than at any other agency in the Country, and second because we who are interested in aeronautics should have a clear understanding of what that method is and have a viewpoint of our own on what the best procedure in distributing the work of development between the Government and private enterprise is. I think that in the next few months, or in the near future at least, we are likely to see a strong drive from some quarters for a revision, in one direction or another, of the present method.

I have heard from various quarters in the industry the sentiment expressed that the money that goes into the Air Service in general is a good appropriation gone wrong, and that it would have been better to turn the same money over to the manufacturers to keep alive on. On the other hand, we sometimes find the view on the part of those connected with the Governmental work, or strongly sympathetic to the Governmental work, that the manufacturer should confine his attention in peace times to producing those military supplies that are turned over to him by a Government agency, which, in the case of the Air Service, would mean that all the research in engine design should have been handled by McCook Field. It is very important that such research should continue, and it was handled very ably during the war; but at the same time there is danger in going too far in one direction or another. Owing to the departure of some of those in the automobile industry, I think the chances now are rather strong that those who remain may be attracted to the Government work more on the airplane than the engine side, and that the design will fall more and more into Government hands, a condition that would be dangerous in case an expansion of production facilities again become necessary.

If we get all the engineers, particularly those who are handling design, into the laboratories and debar the factories and designing staffs from going ahead with original work, we will be in a very poor position to expand, as bad as we would be if we allowed the factories to do their own work and did not enforce the exchange of results in research and original investigation. It is desirable, I think, that the information come into hands from which and through which it will be, as far as possible, impartially distributed to those who can make use of it for the development of aeronautics in general and for the good of the industry as a whole. Take a hypothetical case: If we suppose that all Government agencies such as that at McCook Field were to cease their research, it would be very difficult to get into the hands of a small company trying to produce engines or airplanes, perhaps more particularly airplanes, the results of the work that is being done by others. I am a very strong advocate of Government research and of the maintenance of Government laboratories, not only for research of its own but for the collecting of information on a uniform and impartial basis, and for the distribution of research data from whatever source they may come.

H. M. CRANE:—I have had a pretty long experience with private and with Government development work, and with the two in combination. I have known Colonel Bane for a long time, and I think that he has worked out about as just a balance between the advantages of the two systems separately as it is possible to work out.

We must not overlook the fact that it is desirable to have an educated personnel in the civilian outside field. It is equally important not to overlook the fact that it is necessary and desirable to have a highly educated personnel in the Air Service, in both the Army and the Navy, that will be the nucleus for an expansion during war. All of us who had anything to do with the early supply of war material know what a handicap it was to deal with men in the Air Service who knew nothing about the work. At present, however, we have at McCook Field enough men of very great air experience, who are thoroughly acquainted with all the difficult problems of engine development, are sympathetic toward the difficulties of such work and know the length of time it takes and the importance of it.

When I first had anything to do with aviation-engine design, the question of development did not mean anything, any more than it did with airplanes. The airplane designer seemed to be expected to stand or fall on the first airplane he produced of an entirely new design; if it did not work, the attitude was to throw it away and make an entirely new and completely different design. But it has been proved by experience that the best airplanes and the best engines cannot be obtained in that way. Mr. Jones will undoubtedly tell you that the best engines in use today are the ones that have had the longest period of development; and the ones that started soonest with any sound basis on which to go, and have been most consistently developed, are still the most useful engines.

I was told by a man in Washington who should have known better that you could not keep up with engine design unless you lived in France; that the designs changed so quickly and were improved so rapidly that, unless you were right next to the front, there was no opportunity to keep up at all. As a matter of fact, if you look over the standard engines used in the Air Service in this Country now, you will find that all of them are based on designs that were made from 1915 to 1917. I cannot at the moment think of any service engine in regular use today in the Air Service, that was designed originally at any later time. In other words, the 4 or 5 years of development of a sound, original design has far outstripped anything new that could be produced in a year or two, based on all the knowledge of that development work.

ELMER A. SPERRY:—I think there is a general feeling in engineering circles that in Colonel Bane's organization at McCook Field we have a remarkable agency, paralleled probably by no other one in the world, where aid is extended not only to the recognized designers and to those that have been longest in service and whose product has been longest under study and has been perfected gradually and persistently, but where the radical designer also has a wonderful reception and is not only encouraged and sympathized with, but given every possible aid. I have known of photographs being taken especially for designers so that they could have the benefit of the very latest developments in the particular line in which their endeavor lay.

It is, I think, beyond the knowledge of the lay engineer how completely Colonel Bane's wonderful organization has been developed, how he inspires his men, extending this even to the workers outside, and how broad and perfectly just it appears to those who have to deal with it. We now learn again from Captain Hallett's paper how extreme performance is not demanded in early attempts and how the designs are treated sympathetically, working toward the proposition of determining first the soundness and ultimate utility of the principle. I think this stands in rare contrast to the treatment that some designers have received from other quarters. I, for one, think that everything should be done to strengthen the organization at McCook Field. It is in good hands, the

appropriations there in all probability reach farther and it accomplishes more than any similar organization. There is no organization in Europe that has attained the accuracy of bomb-dropping, the perfection in photography, and the ceiling that McCook Field has attained; or that has been as bold and progressive in a great many lines and as willing and daring to try even extremely radical things. In some cases these are successful and in every case they are extremely useful in teaching us what not to do, which every engineer knows is almost equally as valuable as finding out what to do. I think that as a body we should thank Colonel Bane and his whole organization to the very best of our ability and extend to them every support.

PROFESSOR WARNER:—Mr. Crane expressed by views perhaps better than I could myself, relative to the balance of justness and the accuracy of the balance that is being struck by the McCook Field organization at present. I hope I was not understood as offering any criticism of that or of any other present organization. I said rather that before any change is made in either of the two directions I suggested, we should consider very carefully what the desirable change is.

What I desired to emphasize mostly is the necessity of the development work, the necessity of sympathy between the Government and the industry and of an educated personnel in the industry in the development of a new type. I know of several airplanes used during the war that, according to records which we have received since the war, were very unsatisfactory when first brought out and did not really become satisfactory airplanes for pursuit use, although they were later among the best on the front, until they were modified and rebuilt 20 to 30 times.

If airplanes are to be designed by the manufacturer, if he is to do anything except produce them, it is necessary that he be in a position to undertake some of that modification. Of course, the present situation in the airplane industry is very unfortunate. I think Chairman Clark will bear me out in saying that the ideal condition would be one in which an order for an airplane, when given to a manufacturer, could be carried through by that manufacturer to the point where he considered the machine as satisfactory; in which he could do a large part of the experimental work and then put the machine out with a guaranteed performance and declare that it was satisfactory to him, instead of, as at present, having to receive an order for an airplane that has never been built, on the basis of design alone, and to submit that airplane which, as a rule, never has been flown before it gets into the hands of the Government.

That is the one point where we might perhaps hope for a better balance in the experimental work. It is a point regarding which it is very difficult to see any practical improvement at present. None of the manufacturers of airplanes is able to undertake that experimental work under present conditions; it has to remain in the hands of the McCook Field personnel. Some difficulty to the manufacturer in keeping in touch with the changes made and in incorporating what proves desirable in later design is likely to result. As has often been the case, a manufacturer may have to change from a pursuit machine to a day-bomber and then back to a ground-attack type, instead of taking a pursuit machine and being able to develop it himself, carrying it through half a dozen constructions, as has often proved desirable. It is necessarily difficult for the designer to profit fully by the results of experience with his designs in flight.

I think that is the one point in which a change in the present system might be made profitably. Recent practice in Government work has been somewhat in the direction of allowing the designer more freedom in profiting by the results of experiments on his productions, in allowing him to continue to build other machines that incorporate improvements suggested as the result of trials of first types.

MR. CRANE:—Professor Warner should not have thought that I was attempting to criticise his remarks. I was only trying to amplify them. However, his later remarks are along a line that is also extremely close to my heart. I think that his ideas are being worked out at McCook Field now as well as is feasible under our system of handling appropriations by Congress. There are a great many builders who would like to work at double their present prices; in other words, they would like to take a guaranty contract if they could get an appropriation large enough to pay for two or three failures before they came through with the final result. But under the scrutiny given all McCook Field contracts by more or less interested parties, it has seemed to be impossible, I understand, to give contracts on that basis. It is a much better basis without a doubt.

The size of this Country and the limited personnel that is available now in the airplane plants result in very serious lost motion between the designing and the development testing. The testing is bound to cover very considerable periods of time and it is impossible for the designer to spend adequate time at McCook Field when his factory is somewhere else; he has to be at home and active on his other work. This forces him more or less to take the printed or typewritten word on the results of tests, or information given him verbally after the tests have taken place. This prevents his seeing the actual results and is undoubtedly a very serious handicap.

The other handicap, an endeavor to overcome which was made during the war, is the year-to-year appropriation that makes it difficult to give a long-term development-contract to anybody with any certainty of it being supported through to the end. I do not know whether we can obviate those difficulties with our present system of Government, but I do know that in the last 2 years McCook Field has succeeded in overcoming one by one very many of the worst of the difficulties. They are not overcome entirely yet, by any means, and it probably will be impossible to do so under peace-time conditions.

FREDERICK E. MOSKOVICS:—It seems to me that there is something to guard against in Professor Warner's suggestion regarding the latitude to be allowed to the individual manufacturer for experimenting. I think there is much to be done in educating the individual manufacturer who is experimenting as to just what he should do. I recall vividly an incident that took place at McCook Field during the war, when one of these experimenters brought a new airplane there for a flight test. When the McCook Field authorities proposed to sand-load the wings and test it, the airplane representative immediately went to Washington to protest to a certain senator that McCook Field was going to "bust up his plane." They sent a commission down there; the Military Affairs Committee of the Senate took the matter up and almost peremptory orders were sent down there to let this fellow fly his machine without sand-loading. He was supposed to have done a certain amount of experimenting. Glenn Martin and others talked for hours with the pilot who was to fly the machine and begged him not to. We all talked with him and said it was absolutely sure death. The machine did everything in the air that the McCook Field authorities had predicted it would do, and the pilot was killed

within 3 min. after the plane left the ground. There is a perfect illustration of letting a jackass do his own experimenting, without some check upon him.

MR. SPERRY:—I do not believe in too much private experimenting with airplane engines, using Government funds. I can understand the lure of doing one's own experimenting, but where is the superman who can produce results comparable with those that are produced when the same machine is taken to McCook Field for test, where it is brought immediately under the observation of 20 experts instead of one. These can look at it from as many different standpoints and can give advice for improvement along many lines, thereby tremendously accelerating the development. I firmly believe that the system that is being pursued has very many advantages, and that it should be continued until some better method has been found.

PROFESSOR WARNER:—The solution for preventing the type of accident that Mr. Moskovics mentions seems to lie in having a trained personnel in the industry; the securing of engineers who know something about their work before they are turned loose to experiment. I think no one would advocate letting anybody who wants to do so build an airplane and then have the Government fly it; but I do think that the industry ought to be encouraged to build up and keep up that organization of aeronautical engineers that was started during the war, and to secure trained men who are capable of conducting experiments. McCook Field certainly ought to continue; it would be absolutely disastrous if it were to cease its work of sand-loading the wings and of making investigations of airplanes. However, there are some things we cannot calculate. The maneuverability of airplanes is one of those things. No one appears to know really at present whether an airplane will be maneuverable or not, when it is started. The only way you can tell is by trying it out. For example, in the case of the machines that I spoke of, where it was necessary to destroy and rebuild 20 to 30 times before a satisfactory machine was secured, it was necessary to rebuild these machines to improve things that could not be calculated in advance.

It is a legitimate thing, it seems to me, for experiments to be conducted by the manufacturer, provided he has a competent, trained personnel of his own. You are dealing with a very difficult thing when you are trying to build an airplane at points 1000 or 2000 miles apart; there is a great amount of lost motion in calculating the tests on planes that have been redesigned.

MR. MOSKOVICS:—The point that Professor Warner brings up, goes without saying; but who is to determine the capabilities of this experienced designer? As to the ability of a manufacturer to do a certain amount of experimenting, I was impressed recently at Port Washington while watching a flight of a Loening seaplane, the first monoplane seaplane Mr. Loening built in regular production. The machine was developed by Mr. Loening's experiments.

If McCook Field could keep in intimate contact with other research fields, I would say that nothing should be left undone that could be done in that line and that it would be a great step in the right direction.

PROFESSOR WARNER:—Mr. Moskovics has, I think, an excellent illustration of the value of experimenting with a competent personnel. The man who is constructing for the Government should be authorized to do Government work. As to the agency to select competent workers, I think McCook Field is in a better position to say whether or not it would be safe to fly the machines. The authorities there are the most competent judges.

MR. CRANE:—I think McCook Field found during 1918 and 1919 very clearly what they were up against; that the outside designers would take an order for a fighting machine like the pursuit-plane and not try to consider what a pursuit-plane is for. They took a certain set of written rules and went ahead. The main idea in those days was that if you got speed enough and a rapid enough climb, that would secure an order. At that time there was no way of getting a really good all-around fighting machine in this Country, because there was no designer in the field outside who knew what it was or seemed to care what it was.

All of the early machines that came to McCook Field were lacking in every detail of a good field-machine except possibly speed and climb. The engines were not accessible; it was impossible to do the most ordinary work on them; the machines were difficult to set up and take care of, and in most cases you could not fire a gun successfully out of any one of them, which of course was the final and limiting objection. Since that time McCook Field has been more and more willing to work with the designer who is willing to try to work out their problems for them. They are striking the happy medium between discouraging the outside designer who simply has something to sell, or a weird idea that has murderous possibilities, and the other designer who will really aid them in their work. They have gradually modified their system of contracting until it is about as good, I think, as it can be at present.

Mr. Moskovics mentioned the Loening organization. I know the form of contract it has taken in recent years. The contracts have been of a continuing nature; that is, for four or five machines of a given type, no two of them alike, or ordered for and delivered at the same time. This has given an opportunity to modify each machine from the results of the one preceding.

To revert to war times, a special dispensation was made in the case of the Loening organization on sand-loading. I think that had never been done before. But McCook Field very wisely provided the first sand-loading test on the wings, with the idea that those wings were an entirely experimental set and should be tested properly under the supervision of the designer and his men. The wings stood up well, but one metal fitting crumpled up, which would have caused a serious accident in the air. But the deflection of the wings that occurred during different stages of the sand-loading under the supervision of the designer permitted him to modify the design and obtain a factor of safety something like 15 per cent higher, with no added material at all. It was a point that could not have been covered in previous calculations.

What we are doing is part of the education of the outside personnel, and it is a very important part. The airplane is the toughest problem we have today, for the reason that it is an assembled machine, and a machine in which the correct assembly is of the utmost importance. Our best motor cars are practically designed completely and constructed in individual plants. The assembled cars are never in the same class in efficiency. The airplane is still an assembled proposition. In designing a new airplane for the Government, one result is that we cannot reasonably expect the installation to be good on the first one. The attempt to take the time required to work out the installation to the last little details on a drawing-board, on a machine as complicated as an airplane, would be time wasted. The most important thing on the first design is to have it strong enough, and to fly it and determine that the general type is suitable, that it has the basic fundamental requirements that justify the

development expense. When that has been done, by preliminary experimenting and preliminary sand-testing, and with those samples on hand that you can measure up and on which you can shift parts about, you can determine the desirable modification in structure.

Any other system than that is bound to result in a junk-pile that is all out of proportion to what we ought to have, or else a line of service machines that are really entirely unserviceable compared to what they might be under other conditions.

OVERHEAD-CAMSHAFT PASSENGER-CAR ENGINES

BY P. M. HELDT

THE gradual trend toward overhead valves in automobile engines, as indicated by an increase in their use on American cars from 6 per cent in 1914 to 31 per cent in 1922, has been accelerated, in the opinion of the author, by their successful application to aircraft engines and by the publicity given them by their almost universal adoption on racing machines. Tractor engines recently brought out show the advantage of this construction. Methods of operating valves in the cylinder-head; the advantages of the valve-in-the-head construction as regards the form of combustion space, engine cooling and high-speed operation; the reason for using an overhead camshaft to operate the valves on racing engines, the question of noisy operation and the possibility of having an overhead camshaft engine operate as quietly as one in which the camshaft is enclosed in the crankcase; the location of the drive in the various foreign engines of the overhead-camshaft type; the silent operation that is possible with a rear drive; the use of chains and spur, helical, worm and spiral bevel gears for the camshaft drive, with the advantages and disadvantages of each method and descriptions of specific applications; and some radical designs of overhead camshaft drive and valve-actuating mechanism that have been developed abroad are among the topics discussed. The three methods of operating the valves: (*a*) directly through the action of cams on followers secured to the end of the valve-stem, (*b*) through the interposition of single-armed levers or adjusting blades between the cams and the valve-stems, and (*c*) by the use of tappet levers, are also outlined with particular reference to the specific applications of each. Numerous illustrations supplement the text. [See June issue.]

THE DISCUSSION

F. E. MOSKOVICS:—As to the difficulty in the direct application of the cam, is not the chief difficulty due to the thrust on the stem rather than to the clearance? I have had considerable experience in that work. I have had the benefit of investigating practically all of the racing cars of foreign makes that come to Indianapolis for the 500-mile cup-race. That can be illustrated best by the lengths to which both the Peugeot and the Ballot designers have gone. They use a hardened cup that fits over the top of the stem, and that cup is in a guide which carries the thrust entirely free from the stem. Louis Chevrolet also uses a similar form.

I think there is no possible question that, from the technical standpoint, the overhead camshaft has certain advantages, but it is difficult to drive it. Chains cause trouble due to stretching. The bevel-gear thrust can be balanced but it is a pretty exact piece of work. All of these troubles are eliminated in the well-designed, well-built spur-gear drive.

One point that Mr. Heldt's paper did not bring out strongly enough is the service problem. The passenger-car engine, be the design ever so good, does occasionally become carbonized. It must be demounted and mounted again. I submit that, placed in the hands of the ordinary small-town garage-man, demounting and remounting and timing of an overhead-camshaft job is a vital problem in itself.

The conventional location of the camshaft in the engine has one other advantage in that it is in a large mass in the engine which you might say is an automatic muffler, for the noise of the camshaft is mingled with the other noises in the crankcase. I know of no place that is a better sounding-board for noises than the head of an engine that has a thin light cover. We have tried all sorts of things to deaden these noises, but it is just about one of the best drums for noise that there is. I believe the problem of the lubrication of the bearings of the rocker-arm type can be eliminated. I think almost every maker has succeeded in bringing oil up there now by using pressure; that is the general practice.

H. M. CRANE:—I will not attempt to speak for any particular type of engine, but a progressive history of three different types that I have been able to go through with myself is interesting. We first built some L-head six-cylinder engines of 4-in. bore and 5-in. stroke. Subsequently, we built push-rod overhead-valve engines of the same dimensions. During the past year I have been operating an overhead-camshaft engine of 4¼-in. bore by 5-in. stroke. I will state briefly the relative qualities of those three engines. I think that the mean effective pressure can be made almost the same in all of them, but the L-head engine is much more prone to detonation, not of an order that possibly causes loss of power, but of an order that is very unpleasant to the driver and will cause him to retard the spark to prevent it and thus reduce the power. On the other hand, the L-head engine has a distinctly greater turbulence, which gives the advantage of a shorter spark-advance, the difference being that with the L-head engine a spark-advance of 25 deg. on the flywheel is ample for engine speeds up to 2400 r.p.m., while with the overhead-valve engines using two spark-plugs per cylinder from 30 to 35-deg. spark-advance is required to obtain the same results.

The interesting thing about the first overhead-valve engine that we built using push-rods is that it seemed to be quieter per se than a similar L-head engine. This may have been due to the form of the cylinder-head. In our experience the most difficult noise to contend with is caused by the seating of the valve. If it seats slightly in a diagonal way so that the stem slaps across, or if the seat has a sounding-board effect, the noise is very considerable. The opening of the valve, compared with this, is almost noiseless. The overhead-valve cylinder-head construction, with its heavy water-jacketing and stiff ribbing, seems to help the deadening of the exhaust-valve sound.

In our push-rod job we used cylinder-heads in two sets of three each, which of course is very easy to do in an

engine of that type. With the overhead-camshaft job it was obvious that the only real good way to do is to use a single cylinder-head, and there we find the chief advantage of the overhead-camshaft job as a practical thing. Mr. Heldt speaks of this in his paper and I think it is an excellent way of deciding on an engine; that is, which is the easier to take care of.

The overhead-camshaft engine cylinder-head on this rather large engine weighed about 135 lb. That is not a one-man job to dismount. Evidently it needs at least two men, and it is apt to require a chain-hoist. On the other hand, the heads of the cast-in-three cylinders, using rocker-arms and push-rods, weighed about 45 lb. each. It is a very easy one-man job to dismount. It is also easy to handle on the bench for grinding valves and has every advantage.

Our experience on cylinder expansion on this engine, which of course is of a moderately short stroke, indicated that it was wholly possible to figure on not over a 0.01-in. maximum variation on the inlet-valve side, if any means were taken to see that the push-rods are warmed proportionately to the cylinders to some extent. We have found again and again that it is possible in modern cam design to provide for such a variation in expansion without affecting the rate of opening and closing. I think this is the type of design that is called using a ramp on the cam. The principle is to determine a range of angle on the cam through which the speed of action of the lifter is the same, and if that is done it is obvious that no matter at which point the cam takes hold, the shock of the blow or the shock of the valve seating is the same. We have used that arrangement on exhaust-valves and found no difficulty at all from timing variations.

When you add to the lack of accessibility of the overhead-camshaft engine the difficulty of making the drive quiet, in the first place, and keeping it quiet afterward, I cannot see that there is any chance for argument at all in the passenger car. That is partly because I have always been an advocate of slow-speed engines, and still am. I think that the passenger-car engine of the most efficient type is one that reaches its peak at about 2400 r.p.m. or possibly 2500 r.p.m., and any attempts to make a quiet engine capable of running very much faster will meet with very little success; in fact, if you wish a very high-speed engine and want to have it quiet, I am inclined to think that the L-head engine is the most suitable form. I believe that you can make the parts for suitable bearing areas lighter in an L-head engine than you can in any form of overhead-valve engine. With a moderate speed for the engine it is possible to use large bearing areas with large oil-films and, when that is done, I have no hesitation in saying that an overhead-valve engine can be made that is to all intents and purposes absolutely silent so far as the valve operation is concerned, but it is bound to be the result of a strenuous course of development.

The difficulty of predicting what the result will be with any form of cast aluminum and cast-iron and forged metal construction, as to its acoustic or sounding-board effects, is such that we do not know anything about it. We all know that the Cadillac engine is surprisingly quiet, considering the excessive pressure of valve-springs used and the fairly good weight of the parts. Examination of the design does not give any hope on the face of it that it should be quiet. The valve-operating mechanism is mounted on an excellent sounding-board, apparently, and there is nothing else there to give you the idea that it should be quiet; but I think actually it is extremely quiet in view of what is being done by the valve-operating mechanism. On the other hand, a design with very much lighter springs, lighter valves possibly, and smaller parts all the way through, may cause excessive noise because the parts happen to line up in about the form of a telegraph sounder that brings the full amount of noise out of any mechanical slackness.

J. G. VINCENT:—I wish to second everything Mr. Crane has said in regard to the characteristics of the three types of engine. It happens that I went through just about the same procedure about 2 years ago in experimental work on the straight L-head, the push-rod-operated overhead-valve and the straight overhead-camshaft engines.

Taking the overhead-camshaft job first, it seems to me it has been very well brought out what the advantages and disadvantages of this type of engine may be. I think that lack of accessibility is the most important element working against this engine, and I believe that we must all take into consideration the service end of the business. Granting that you can make a quiet drive to the camshaft, there is one other point that I think is important. It is that the tappets and all the operating parts are right on top of the engine, where they certainly are easier to hear than when they are located down in the crankcase.

In regard to the matter of losing the timing when you demount the engine, I think that is very important. In connection with the better location of the camshaft in the crankcase for lubrication, I think it gives the crankcase position unquestionable advantages, particularly as that is the easy place in which to drive the camshaft.

When you come to the push-rod-operated job, so far as drive is concerned, it is exactly the same thing as the L-head engine; whether you choose to use spiral gears of some kind or a chain drive is another matter. Either job can be made good, but I believe that the chain gives the best results from the manufacturing point of view.

So it seems to me that, as Mr. Crane pointed out, it is clear that it comes down to a practical point of view where any quantity of production is involved as between the L-head and the push-rod-operated overhead-valve engines. Undoubtedly, both those engines will be developed and used to a large extent, because they both have some advantages. Perhaps they can be made equally quiet. Of course, the L-head engine with the demountable head still has some advantage from an accessibility point of view, and the question I would like to raise is: Is it worth while? In other words, what do you really gain by putting the valves in the head? I have been unable to determine any advantage, and I think there are many other things about the engine that can be worked on, that may give as much or more advantage. For instance, very careful attention to the carburetion will accomplish wonderful results. When you come to consider that the passenger-car engine is operating most of the time at a very small percentage of its maximum output, you will find that with the compressions you can use there is mighty little to be gained in either power or economy with the overhead type of engine. I was not able to obtain anything in excess of a 1-per cent advantage; so, it is very questionable in my mind, because the overhead valves add some weight to the engine, whether the small increase in efficiency will make up for the added weight you have to put in the engine. The best way to get economy is to reduce the weight and, as Mr. Crane said, use an engine of slower speed with a relatively moderate gear-ratio. It will be interesting to see which one of these two types of engine will succeed in the long run. Probably both will always be used. So far as I can see, however, the overhead camshaft cannot be manufactured

in quantities on account of the cost and the difficulty of making the job quiet.

F. S. DUESENBERG:—In regard to the setting or the timing of the gears in the service-station, I believe that as we are building our engine at present there is less liability of getting the timing off than in any other type of engine, because we use the upper gears mounted in the head and on a vertical shaft that is driven by one jaw in such a manner that it cannot be put together wrong. If you take the head off, there is a possibility of turning the crankshaft over one complete turn, and in that way the ignition might have to be set but, so far as valve-timing is concerned, it is impossible to get that wrong. I really feel that there is not any disadvantage in this point. The quietness, of course, is something that needs to be considered, and I believe it is easy to get a rocker-arm or the overhead camshaft quiet, but of course the gear noises must be taken care of. I have been driving a Willys car for the last year and I bought it to test out the gears. When I first got the car it was very noisy, but no adjustments have been made and, after 10,000 miles, it has surprised me on account of absolutely losing all the noise it had when I first got the car. For this reason I feel that it will not be an impossibility to get a very quiet gear if it is properly made. However, I do feel that any gear that will be quiet after a few thousand miles of use is apt to be slightly noisy at the beginning, because it has to be set up very close.

The accessibility of an overhead camshaft I really feel is rather better than it is with the other type of construction; it leaves the crankcase very free. In the arrangement that we have with the overhead camshaft, the gaskets bothered us at first on account of the variation in their thickness, but that was easily overcome and we have experienced no other trouble. We find it possible to take the present type of head off, grind the valves and have it back on the job in 2 hr. Two men can do that easily. The weight of the head is to be considered and does not make it as accessible or as easy as some of the other forms; our head weighs 140 lb., complete.

I might cite an experience that Bill Murphy had in a race at Los Angeles about a year ago where they had four heats in the final. Murphy burned a valve in the first heat; he tried to run the second heat and found his carburetion was unsatisfactory, so he stopped and took the head off. He had a new head without a camshaft; so, he removed the camshaft from the old head, put it into the new one, put the head back on and had the engine running in 32 min. from the time he stopped. In the next heat, he broke the world's record by 3 m.p.h. This incident shows that those things can be made very accessible and very easy to handle. I feel that the combustion-chamber on an overhead-camshaft engine can be made so clean and so easily that it really is a good production job, but I think that a little greater accuracy is necessary if you use the gear drive than if a chain drive or the ordinary L-head is used.

R. ABELL:—Mr. Duesenberg's experience and my own have been about the same in regard to the overhead-camshaft engine; namely, that the dismounting of the head and valve mechanism and its replacement is a shorter job than with any other type of engine. With a helper, I have taken the head off of my engine, removed the camshaft, inspected the valves, reassembled the head, timed and adjusted the valves and had the engine in operation again in less than 1 hr. The principal difficulty has been with gear noise, but it has been amply proved in quantity production that this feature has been overcome and that overhead camshafts are now remarkably silent. I think the important thing in the design of the overhead-camshaft job is to make the adjustment very accessible; in other words, to have a cover-plate removable from the front so as to expose the entire drive in order that fine adjustments can be made easily, with good opportunity to examine the meshing and lost motion of the gears. If the noise is more than is desirable with a new set of gears, a slight amount of use will indicate the adjustment required and, as adjustment is permitted with this type of drive, it is important to have the adjustments as accessible as it is possible to make them.

In most overhead-camshaft engines, to make adjustments of the lower gears it has been necessary to disassemble the entire engine and then work in the dark; so, in an economical production design, it is very important to provide for accessible adjustment at this point.

R. E. FIELDER:—I have listened with great interest to Mr. Heldt's paper. Certainly its discussion is most illuminating. The point that strikes me most forcefully is the fact that, even at this rather late date, the poppet valve and its method of operation are still far from perfection. As a matter of fact, there seems to be a very general disagreement in regard to certain controlling design factors.

Nearly all the speakers have referred to certain disadvantages that are inherent in the various types of poppet-valve construction, and one instinctively wonders why it is that designers tolerate the disadvantages that they have mentioned, the most important of which are noise, inaccessibility and difficulty of assembly and of adjustment. Certainly, these disadvantages are not found with the sleeve-valve type, the pocketless combustion-chamber, simplified cylinder construction and accessibility of which stand out so prominently.

D. B. WEBSTER:—One of the cases cited in Mr. Heldt's paper is that the Leyland eight-cylinder-in-line engine is designed with a camshaft operated by connecting-rods. That seems to me to offer a way out of the gear noise. Has anyone had any experience with that type of operation? Are there some objections to it that we have not heard of?

MR. DUESENBERG:—The rod type of control shown on the Leyland engine has been experimented with by several people. It was tried out a few years ago by one of the Western engine makers and it was found absolutely impossible to use it with a separate head construction. The change of the gasket or the variation that occurred in the engine when it was taken down was such that the distance in length would change and the rods could not be changed, and there was no way to get them adjusted properly so that they could be operated satisfactorily.

P. M. HELDT:—That is being taken care of in the Leyland engine by using a short shaft. The connecting-rods do not drive directly on the camshaft, but to a shaft that has an Oldham coupling connected with the camshaft.

A. L. NELSON:—With regard to comparing the L-head and the valve-in-head engines, I think most of us will agree that very fine results have been obtained by both types so far as present-day practice is concerned. It is my belief that the requirements of today must be set aside in the light of the great advance that has already been made in the way of providing fuel "dope" that makes practicable the use of compression-ratios of about 7 to 1. To obtain an appropriate combustion-chamber on an L-head engine having a high compression-ratio, together with reasonably large valves, is an exceedingly difficult matter when giving due attention to port areas that permit the attainment of a fairly good volumetric

efficiency. We have observed very often an actual loss of power in experiments when the compression of L-head engines was increased. This we know is contrary to what should be expected. Invariably it is found that the charge is throttled on entering the cylinder. This throttling of the charge accounts for the lack of increased power.

In the case of the valve-in-head engine, the compression-ratio can be increased readily without throttling the charge. It is thought that designers should bear this important consideration in mind when looking into requirements for the near future. In short, the valve disposition that is universally approved in aviation-engine work should eventually become the universal practice in the high-efficiency engines that we will be called upon to produce in the near future. There appears to be no room for doubt that the highest economy, with the least amount of detonation or fuel knock, will be obtained by adopting valve-in-head engines.

MR. HELDT:—I can say only that there seems not to be very much sympathy for the overhead-camshaft engine among the scientists here. I have followed foreign periodicals pretty closely and have noticed that very many patents have been taken out on different constructions. If the overhead-camshaft engine is adopted here, I think it will be for the higher grade of cars where the cost of construction is not so very important. I believe it is not the proper kind of construction to be incorporated in a low-priced car.

A. C. WOODBURY:—A simple way to secure different periods of valve-opening from the same cam where a roller cam-follower is used is by varying the size of the roller. As an illustration, let us assume that it is desired to hold the exhaust and the inlet-valves open through periods of 230 and 215 deg. respectively, 115 and 107½ deg. of camshaft motion, and that the valve-stem clearance is to be 0.020 in. With a straight-sided cam, a 1¼-in. base-circle and a ⅞-in.-diameter roller for the inlet-valve, the angle between the two sides of the cam would be about 67 deg. On the same cam a 2-in.-diameter roller would produce the 115-deg. opening period required for the exhaust-valve, with the same lift as the inlet-valve.

So far as valve-timing is concerned, the slipper of the Leyland engine would act the same as a roller and the opening periods can be varied by changing the distance from the pivot-pin to the slipper face, but from Fig. 19 of Mr. Heldt's paper it appears that there is no such difference between these dimensions on the two slipper-blocks as would be required to make the customary difference between the two valve-opening periods.

DETONATION CHARACTERISTICS OF SOME BLENDED MOTOR-FUELS

BY THOMAS MIDGLEY, JR. AND T. A. BOYD

THE effects of admixtures of various percentages of alcohol and alcohol-benzene mixtures for reducing the detonating tendency of paraffin hydrocarbons have been measured by the authors. These results represent an extension of previous work in which similar determinations were made for benzene and other aromatic hydrocarbons. The bouncing-pin apparatus was used for making the determinations. The data obtained by its use are considered to have a high degree of accuracy.

In order that the effects of the blending materials might be measured through as wide a range as practicable, they were blended with kerosene for making the majority of the determinations. This made it possible to ascertain the characteristics of the materials up to a concentration of 80 per cent of benzene or 50 per cent of alcohol without introducing the difficulties due to excessively high engine compression. Because xylidine has the property of exerting a powerful suppressing action on detonation when present in a fuel in percentages that are relatively very small, the standard used as a basis of comparison in the tests was composed of small percentages of xylidine in the paraffin fuel. Tables and curves are appended that show the results of the tests in detail. [See June issue.]

THE DISCUSSION

PRESIDENT B. B. BACHMAN:—It seems to me that Mr. Midgley has shown that the dilution of kerosene with benzene is more effective than the dilution of high-test gasoline with benzene. Is that correct? If so, why?

THOMAS MIDGLEY, JR.:—Yes, the curve as plotted shows the increase of compression pressure that the fuel would stand. The benzol originally would stand a compression pressure of 50 lb. per sq. in. and the commercial benzene 20 lb. per sq. in. For sake of argument, let us say that the kerosene will now stand 70 lb. per sq. in. The increment is not so great and, as the fuels get better, the increment apparently decreases.

P. J. DASEY:—Do the increments of compression shown on the right-hand side of Fig. 2 in the paper indicate an addition to the normal compression, on the basis of calculation? What is that basis of calculation?

MR. MIDGLEY:—In this particular engine the compression pressure is 52 lb. per sq. in.; in the case of gasoline it is 75 lb. per sq. in., or thereabout. It is the increment to whatever the fuel stands.

MR. DASEY:—I noticed that they all started from a common point.

MR. MIDGLEY:—Yes.

MR. DASEY:—It is interesting to have that basic compression pressure put in.

MR. MIDGLEY:—It is difficult to get a reading that is really accurate or to state the compression pressure that the fuel will stand, or the pressure ratio, for the reason that when the engine starts you get a certain value; tomorrow you get a different value; and the next day still another value. You can get any value for any set of barometer readings, carbon deposits or temperature readings. It is affected by the smallest thing. You cannot take a reading and say, "That is it." The increments depend upon the relation between the different fuels, one being much better than another. That is the difficulty of getting down to an accurate basis.

HERBERT CHASE:—What method did Mr. Midgley follow in setting the spark when measuring detonation? I ask this because detonation is known to vary considerably with the spark position.

MR. MIDGLEY:—We set the spark, roughly, at the point of maximum power. On the other hand, when comparing two fuels, the spark-advance was the same for both; so that the effect is cancelled. If the spark were wrong for one, it would be wrong for the other; the method is essentially a comparison of two fuels, one being a standard of some sort. The actual readings show the relation between two fuels; the spark-advance is kept precisely the same in making the comparison.

AIRPLANE PERFORMANCE FORMULAS

BY EDWARD P. WARNER

AERODYNAMIC analysis relates mainly to questions of performance and stability, the latter including both maneuverability and control, but the designer's problems concern chiefly the prediction of the best possible performance. Accurate analysis, which would include a summation of the elemental resistances of an aircraft part by part and the making of many corrections, supplemented by tests of models in a wind-tunnel, involves much labor and expense.

When a preliminary choice of dimensions and specifications for a new type of an airplane is to be made or there is a question of the performance attainable with a given load and power, a shorter method becomes necessary. This is to be found in the derivation of simplified formulas and graphs.

The author illustrates by examples the process of deriving these formulas and considers in turn such elements as minimum and maximum speeds; climbing ability and the conditions under which an airplane will have a zero ceiling, that is, the limiting conditions under which flight is possible; rate and angle of climb, the latter controlling the possibility of getting out of a small field over a barrier; and the best ceiling possible. Under these heads he takes up such questions as fineness, which is the ratio of the parasite resistance to the wing area; and lift, weight and power and their relations in determining the various coefficients which are used. Values obtained theoretically from the formulas were checked by comparison with those of about 50 airplanes of various types of which the measurements and performance are known and application is made to specific examples. Numerous curves show the relation of the various coefficients that enter into the design and performance of the airplane. [See June issue.]

THE DISCUSSION

CHAIRMAN V. E. CLARK:—In connection with Professor Warner's curves on rate of climb and ceiling, he uses the pounds per horsepower times the square root of the pounds per square foot; in other words, the second member of the expression is $\sqrt{(W/S)}$. Did Professor Warner try to introduce into that $(W/S)Ky$ instead of using $\sqrt{(W/S)}$, where Ky is the maximum lift-coefficient of the particular wing-section and the particular arrangement as indicated in wind-drag experiments? If he did try that, did the points come any closer to a common curve?

PROF. E. P. WARNER:—I did not try that out because I was trying for the simplest possible formula for a preliminary prediction. There is no doubt that $(W/S)Ky$ would give a simpler calculation. We find that the machines with thick wings, those having the higher maximum wing coefficient, give a higher ceiling and those with the lower wing-curves give a lower ceiling. That was as close as we could get it without going outside those three factors mentioned. An American engineer has derived a formula for maximum speed that does take this factor into account and appears to give closer results than either this formula or any other that I have ever seen, although I have not checked it up with as many machines as I had here.

H. M. CRANE:—I am not educated to the point of being able to discuss this formula or any of these formulas from the theoretic basis on which they are worked out, but I do want to speak in appreciation of the type of thinking and the type of work that have been done in this paper. It is a very great thing for the average engineering layman to have placed before him in such a simple form the whys and wherefores of airplane performance without going into a long series of calculations. This is the simplest presentation I have seen, and I think that from that point of view it is a very valuable contribution, entirely regardless of its effect on the airplane designer who may use it to make a quick stab at the general proportions of a machine he is working on. It goes much farther than that; it allows the ordinary man who is not doing any of that sort of work to get a fair idea of why one machine does certain things and another machine does something entirely different. I am sure the paper will be of great value to the members of the industry who are interested only casually in airplane designing.

FRANK C. MOCK:—Why does not the parasite resistance cut down the ceiling? If it slows down the plane, it certainly will cut down the climb.

PROFESSOR WARNER:—That is true; it does cut down the ceiling. I did not mean to convey the contrary impression. At minimum speed the wing-drag is much greater in comparison with parasite resistance than it is at maximum speed, though the parasite resistance is apparently of less importance. In maximum speed, the parasite resistance is the big thing. The machines that go to the right of the maximum-speed curve are those in which every effort has been made to eliminate the external bracing resistance. Those that are off to the left of the curve are those that, like the JN-machine, have a large amount of external bracing.

I specified that the exponent used in that curve, if we reduce it to equation form, is 0.39. It is changed to 0.39 empirically, as the result of the plotting of the actual points; because, as a rule, the high-speed machines have a lower parasite resistance than the low-speed machines. Occasionally, however, we get a low-speed machine, such as the Junkers, that has very little parasite resistance. Then the formula gives results that are relatively bad, but even under such conditions the error resulting from the use of the formula is not over 10 per cent.

Progress of the Research Department

By DR. H. C. DICKINSON[1]

SEMI-ANNUAL MEETING PAPER

DR. DICKINSON outlines the history of the Research Department since its organization, indicates why the universities are the principal bases of operation for pure research, describes how the department functions as a clearing-house with regard to research data and comments upon the bright prospects for the future. He enumerates also the facilities the Research Department has for the coordination of research problems.

The practical achievements of the Department have resulted from its recent concentration upon the three major projects of study with regard to the tractive resistance of roads, with reference to fuel and to testing programs, and of an effort to render financial assistance to the Bureau of Standards and the Bureau of Mines that would enable these Bureaus to continue their elaborate research programs, details of all of this work being included.

Supplementary road tests now being conducted by nine different automotive-vehicle companies are outlined, and the factors governing the selection of fuels for test purposes are stated and commented upon.

SIX months ago, at the Annual Meeting of the Society in New York City, the work of the Department was outlined by H. M. Crane, as Chairman of the Research Committee. The Department was then 4 months old. Its program had been mapped out but little practical work had been done. The last 6 months have gone to show what the Research Department can do for the industry and how.

The automotive industry is one of the three largest in the United States. It is generally admitted that the American designer is far ahead of his foreign competitors. America created the industry and still has the lead. It is an industry composed of talented young men, remarkable alike for adaptability and initiative, with no traditions or precedents to hamper their progress. It is intimately bound up with, and dependent upon, a number of other great industries, being a very large consumer of iron, steel, gasoline and lubricants, rubber, paint, aluminum and all manner of accessories. If the source of supply of any of these materials fails, the automotive industry suffers a setback. During the war, when supplies of rubber, shellac and mica were controlled, and other commodities vital to our industry were hard to get, we realized, probably for the first time, our dependency upon other industries, and our general unpreparedness for abnormal conditions. Since that time, the engineers and manufacturers have taken a wider view. Realizing that a shortage of any one commodity may paralyze business, they engage research men to investigate the possibility of replacing that commodity by some other more plentiful material that will serve the purpose as well or better. Again, the automobile builder, anticipating a possible shortage of petroleum products, sets his laboratory men to work on the investigation of other fuels to be used in place of gasoline, and, if he is very foresighted, sets his engineers to designing engines that will burn substitute fuels efficiently and economically.

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The two problems I have just cited are problems affecting the industry as a whole. In addition to such problems as these, every manufacturer has his own particular problems, arising from his desire to make his product give better service than that of his nearest rival. He is constantly on the alert to develop new qualities that will make his car stand out as a good car. Any progressive manufacturer knows how necessary technical research is, if he is to make a success of his business. It is gratifying to note how much interest the manufacturer and engineer are taking in pure research, by which I mean the conduct of investigations not so much for the purpose of obtaining practical and profitable results, as for the advancement of knowledge in one particular problem affecting the industry as a whole. But, unfortunately, very few companies are blessed with a vision of the future combined with a money surplus; the rest cannot afford to indulge in pure research, which offers no prospect of an immediate monetary return.

PURE RESEARCH IN UNIVERSITIES

The principal bases of operations for pure research are the universities. The university instructors and research men are actuated rather by a love of knowledge for its own sake than by any pecuniary gain that may accrue. Their pursuit of knowledge is an end in itself and not a means to an end. The results obtained by university laboratories are very wide in range and inestimable in value. While the main object of the universities always must be instruction, the presence of active and enthusiastic research groups is an ideal, if not an essential, background for the training of young engineers. Many of the schools are suffering from a great handicap. Owing to the scarcity of good teachers and the large number of men seeking instruction of a more or less elementary nature, many of the best potential training-grounds for research men tend to degenerate into mere technical schools imparting textbook information instead of principles, and giving too little individual attention to the students. Under such conditions research languishes, and the teacher loses his morale and becomes a teaching drudge instead of an intellectual leader. I cannot impress upon you too strongly the necessity for a constant supply of competent research engineers, well grounded in principles, to carry on the work of research in the industry. There can be no progress unless we have such men. It is to the universities that we must look for them. I would recommend that before the close of the school year, every man in the industry who employs young research engineers send a list of his requirements to a selected list of universities, so that he may have an opportunity of finding the type of man he needs. I have received a number of letters from young men, most of them completing a post-graduate course, keen research men with ability and enthusiasm, all wanting to know how they are to find an opportunity to turn their training to advantage. One of the larger British universities has an Appointments Board composed of professors and business men, whose task it is to keep in close touch with the industries and find out what positions are open for

trained technical graduates. Any young man about to graduate is at liberty to go before the Board and state his case. Since practically every one of the large industries over there has established its research organization, the trained man is in demand, and none of the fine research material that the University produces is wasted.

A Clearing-House

Turning from the worker to the more pertinent question of the work itself, a review of the research field revealed a number of organizations, both schools and manufacturing laboratories, all engaged in interesting investigations of their own, without any relation to one another. Every day valuable results were being obtained and only a few individuals were getting the benefit of them. Manufacturers were looking vainly for information that was available, but they did not know where to go for it. Over and over again work was being duplicated because the people doing the work had no idea of what other people were doing, while more important and pressing problems were awaiting solution. Everywhere the need of organization and of centralization was apparent.

The first task of the Research Department was to establish a clearing-house of information for the benefit of the industry. Soon after the department was organized, 2 weeks were spent in visiting industrial and school laboratories. The results of these visits were very illuminating. Everywhere we found engineers, professors and students ready and eager to discuss their problems, enthusiastic over the prospect of exchanging information with other people, people who had common research interests, and anxious to cooperate with the Department in every possible way. We had a large index-card that was described in The Journal for December 1921, which we asked them to fill out for our files, so that we might know exactly what work they had done, what they were doing, and what information they needed. In some cases we were able to supply them with the information they required at once, from the material we had on file. They were glad to hear of the organization of a central body to which their problems could be referred, and to which they could apply for information as to what was being done by other laboratories in their own field. They all agreed that, while a certain amount of duplication was necessary and even desirable, the Research Department could serve a useful purpose in preventing unnecessary duplication. Almost every laboratory suggested problems that called for solution, and that could not be undertaken by them owing to scarcity of funds, or of time, or of personnel. Some of the laboratories asked for suggestions as to what problems could be undertaken advantageously and were provided with a list of some of the problems in which the industry is interested. There are on file with the Department a number of suggestions for problems to be undertaken in the future; questions that are of importance but not of immediate urgency. In most cases the report of the laboratories was the same, "the harvest indeed is great, but the laborers are few." The manufacturers, many of them, were suffering from the slump and had cut down to the bone. Many had dispensed with their research staffs, and the rest had had to retrench. There were notable exceptions, that will occur to all of you.

The Future

The prospects for the future are bright. The industry has come to realize the necessity for research and will soon be putting their principles into practice. More valuable still, as indicating the confidence that the Department has already inspired, engineers throughout the Country have developed the habit of taking the Department into their confidence, of writing as a matter of course describing what they are doing, and inviting suggestions as to the method of attack and the general conduct of their researches.

Recently another trip was made to some of the laboratories of the Middle West, and the progress was noticeable. The interest that the Research Department was taking in organizing research work had proved a real inspiration and the results obtained throughout the trip were most gratifying. We hope that eventually all the school and university laboratories engaged in automotive research will be visited, but it will be some time before trips to the Far West can be undertaken. Although the benefits of this field work, in bringing home to members the help they can expect from the Department, have been clearly demonstrated, it is doubtful whether the industry is aware of the special facilities which the Department has to offer.

Research Department Facilities

The Research Department is in close touch with the Engineering Societies Library, which is located on the thirteenth floor of the building in which the Society is housed. It is purely a reference library and contains most of the material that has been published on the automotive industry since its inception. The proceedings and journals of the various engineering societies, American and foreign, are kept on file, as are over 1000 current periodicals along engineering lines. For the benefit of engineers who cannot use the library themselves, the librarian maintains a photostat service, by which a complete copy of a magazine article referred to in another periodical may be had at a nominal cost. Bibliographies, references and translations may be had quickly, reliably and at the minimum expense. The fact that the great resources of this library are constantly at our disposal increases the efficiency of our Department enormously. If any one of our members wishes to know what has been published on the subject of brake-linings, or of pistons, for example, we can supplement the information we have in our own files by referring to the publications on the subject in the library. As a general rule, when we receive a query of this kind, we give a short bibliography of the subject, adding particulars of where the various publications may be obtained. For our own use, we have the Engineering Index, the Industrial Arts Index, and Science Abstracts, which list recent articles by subjects. This we supplement by our own indices, one covering the publications of other organizations, and individuals, and one covering the publications of the Society. These indices are kept by authors, titles, and subjects, and involve a careful scrutiny of all the periodicals, American and foreign, in the automotive and allied fields. In connection with the first-named index, we have a file of pamphlets and publications that are not allowed out of the Department, since they are constantly needed for reference. In the Members' Room of the Society, there is a small library in which copies of current magazines in the automobile, aviation and tractor fields are kept up-to-date for the use of members, as well as The Journal of the Society, *The Engineer*, *Engineering*, and a number of other very valuable publications. These volumes are kept for reference and are available at all times.

Inquiries of all kinds are received from engineers, arising out of everyday experience and covering a variety of problems in automotive engineering. Incidentally, we

have received two or three communications from Cuba and the Canal Zone, asking for information as to what adjustments should be made in an American car to enable it to run on alcohol, the only fuel obtainable at a low price in those territories. Reviewing the variety of problems that are presented to us for solution, we cannot fail to realize that the scope of the work is almost unlimited. Every effort is made to keep in close personal touch with engineers throughout the Country, who are engaged or interested in pioneer work. Many of them have developed a habit of dropping in at the Society's headquarters whenever they are in New York City, to talk over their problems.

In each issue of THE JOURNAL the Research Department discusses some topic that is of interest to the automotive industry, and suggests various problems for research arising from an examination of the subject. The discussion is always accompanied by a selected bibliography. The Department plans to publish in future issues of THE JOURNAL short abstracts of books and articles published here or abroad, which seem to be of interest to the industry, so that members may be sure of being informed of what is being published on their subject.

I have with me a supply of index-cards that were designed to afford ready reference to the activities and the questions of those firms and individuals who are interested in research, and I shall be glad to give one to anybody who is interested in cooperating with us and in getting the full benefit of the service we have to offer.

To pass from the resources of the Department to its practical achievements, we have been engaged for the past few months upon three major projects. The first of these is the cooperation with the Advisory Board on Highway Research of the National Research Council in the study of the tractive resistance of roads. With the advent of the commercial motor-vehicle as an essential part of our transportation system, and with the widespread use of passenger cars for purely commercial purposes, the problem of highways emerged from the "good roads" stage to become one of the most urgent of our national problems in industrial development.

While the problem may appear to be primarily the concern of the economist and the highway engineer, it is equally one for the automotive industry. The Department was originally drawn into this field through a request for cooperation with the Bureau of Public Roads and the National Research Council. A survey of the situation showed that there are many phases of the highway problem in which the cooperation of the Society is necessary to assure full consideration of the interests of the industry, as well as the benefit of its technical experience.

Recognition of this situation has led to the recent appointment of a Highways Committee, of which H. M. Alden is chairman. This will assure the needed contact between the Society and the other agencies engaged in highway research.

In addition to the foregoing activities, we have taken an active part in securing materials and financial support for a project which the Advisory Board on Highway Research has under way at the Massachusetts Institute of Technology under the direction of Major Mark L. Ireland. Extensive road tests have been made and valuable data are now in process of compilation.

FUEL RESEARCH

One of the major activities of the Research Department has been providing for the fuel-research program in progress at the Bureau of Standards. The two other main research activities of the Department have to do with the fuel problem. While many of you are familiar with the history of this undertaking, I shall explain it briefly for the benefit of those who are not. About a year and a half ago, a conference was called at the instance of Dr. Manning, director of research for the American Petroleum Institute, to discuss the joint responsibility of the petroleum and the automotive industries with regard to present and probable future supplies of motor-fuel. There were present at this conference representatives of the two industries, including the National Automobile Chamber of Commerce and the Society of Automotive Engineers on the automotive side, as well as representatives of the United States Bureau of Standards and the Bureau of Mines.

After consideration of a number of proposed research projects, the conference decided to concentrate the efforts of the available research laboratories on a single problem that appeared to be of the widest importance to the two industries. Recognizing that the supply of crude petroleum and hence of motor-fuel is limited, and that there is reason to expect the demand to more than keep pace with the supply, it is of importance to us that the price of fuel should be kept at a minimum or the production at a maximum. It is of importance to the petroleum industry that the maximum amount of motor-fuel should be had from the crude-oil supply, since motor-fuel is the highest-priced quantity product of petroleum. The result of the relation between supply and demand has been commercial gasoline, which we have been able to burn with some degree of satisfaction, but with much more satisfaction at some times than at others because its quality has changed.

It has been suspected for a long time and has recently been proved by experiment that as gasoline becomes less volatile, the amount used per mile increases, however, the amount produced per barrel of crude also increases. If we confine our attention to fuel-consumption, neglecting for the moment such things as crankcase-oil dilution and the like, it is obvious that there is a balance between the increase in production that can be secured through increasing the "end-point" of gasoline, and the decreased mileage that this fuel will yield; and it may be assumed that the interests of the two industries will be best served when the quality of gasoline marketed is such as to give a maximum mileage per barrel of crude oil used in its production.

It is important to remember also that for the purpose of this discussion we are concerned with the average vehicle in the hands of the average driver, and not at all with what vehicles or drivers might be under ideal conditions, or even very much with what they may be 5 or 10 years hence. It is the average driver of the average vehicle who will consume the next few years' supply of fuel.

To adjust the quality of gasoline so as to meet the condition of maximum utility as defined above, or in fact any other specified condition, it is necessary to know

(1) The relation between fuel consumption and volatility for the average vehicle in use
(2) The relation between volatility and the amount produced per barrel of crude under average refinery conditions

For an answer to the second question we can depend upon information to be secured from the petroleum industry, partly through the agency of the United States Bureau of Mines. The first question however, is purely

an automotive one and for its answer two distinct research projects are in progress.

Bureau of Standards Tests

The first of these, in point of time, is being handled by the United States Bureau of Standards with the cooperation of the Bureau of Mines. This work was initiated before the organization of the Research Department; hence we, as a department, had nothing to do with its inception. Plans for this research had been under way at the Bureau of Standards for many months and were substantially completed some time ago.

The Department has, however, devoted much time to consultation with engineers and executives in both the automotive and the petroleum industries to insure the necessary financial and moral support for this research, and to make the results as useful as possible to the two industries. I do not propose to describe them in detail, since W. S. James, of the Bureau of Standards, who is in charge of the work, and other members of the staff of the Bureau are to speak to you about them later. The work has been the result of a year of careful study and experimental development of apparatus and methods. In connection with the work the Bureau of Mines is in charge of the testing of fuel samples, and will compile the results derived from this part of the tests and will assist in the second part of the program as outlined above, determining the relation between fuel volatility and the amount produced from the average crude oil.

The principal task of the Society in this connection has been to secure funds for the work of the Bureau of Standards and the Bureau of Mines. This problem was presented to the National Automobile Chamber of Commerce and the American Petroleum Institute as a joint research program, with complete explanations of the nature of the work and the cost of materials, apparatus and assistants. The two organizations voted to share in the expenses, and very hearty promises of cooperation were received from individual companies. A number of research engineers have been secured by loan, one each from different firms representing both industries, and active research work on the program is now in progress.

Supplementary Tests

The second portion of the program was undertaken at the suggestion of a member of the Research Committee, to supplement the program of the Bureau of Standards. The latter, consisting of a series of road tests of a limited number of cars under as nearly as possible laboratory conditions as regards precision of measurement, could not include all conditions of use that the average driver encounters. It was thought, therefore, that much might be gained by a series of less elaborate tests which could include a much larger number of vehicles, driven by average drivers in normal service. Accordingly, a supplementary research program was drawn up along these lines, and visits were made to several companies, including those building passenger cars in the largest number, and at present nine different companies are each running, or have completed a series of tests that will be described later by the engineers in charge of the tests themselves.

The plans included observation of the comparative effect of the four grades of fuel in crankcase-oil dilution, by draining and refilling the crankcases of the test cars each time the fuel was changed. The used-oil samples are to be tested at the Bureau of Standards, as well as by the several companies, to check the amount of dilution that resulted from the use of each of the fuels.

Fuels for Test Purposes

The selection of fuels for test purposes presented some rather complicated questions, as to the number of fuels to be used, the range of volatility to be covered, the limitations to be imposed as to chemical composition and as to refining methods. These questions involved numerous conferences and visits to refineries, and resulted in the selection of four experimental fuels ranging in volatility from approximately aviation-grade gasoline to a fuel about as bad as commercial gasoline ever gets.

Provision for supplying these fuels was made by the American Petroleum Institute, the fuels to be sold to the experimental laboratories at a price that will partially distribute the cost of the research between the automotive and the petroleum industries. The supplies of fuel have been made-up by two refineries, one in Chicago, and one on the Atlantic coast.

The tests were intended to show under normal everyday running conditions with average drivers, what effect distinct differences in fuel volatility have on the total fuel used per mile of travel. They serve as a check on the results of the Bureau of Standards' tests, which are run with much greater accuracy than is possible under conditions such as these, but must necessarily include a limited number of cars. In this respect they are similar in plan to the tests described by C. L. Coleman that involve an even larger number of vehicles, but not so many different models. The main feature of the tests was the operation of a number of cars of each of several models, by their regular drivers in the course of their ordinary driving, but each car supplied successively for periods of 1 week with fuels differing by definite steps in their volatility characteristics. An essential point was that the drivers should not know what grade of fuel they were using at any time until the tests were completed.

It should be noted in connection with these tests that the information desired is not the actual fuel-consumption of the different cars, or different models, but the difference in fuel-consumption as produced by the differences in volatility of the four selected fuels. While the seventy-odd drivers of cars of 10 models included in this series might not afford a fair average of the fuel-consumption of these cars, it may be expected that they represent much more nearly a fair average of the difference in fuel-consumption with the different fuels, and hence afford reliable information as to the relation between average fuel-consumption and volatility for these cars. To make the results applicable to the average fuel-consumption throughout the Country, it will be necessary to average them with respect to the estimated number of cars of each model in use, or perhaps, to be more exact, with the estimated total fuel requirement of each of the several models.

The Research Department expects to compile the results of the road tests on this basis as soon as possible after the data are available.

THE DISCUSSION

P. S. TICE:—The avowed purpose of all this research seems to be to arrive at the fuel that will give us the greatest economy. The consensus of opinion appears to be that we have not as yet arrived, in average practice, at a method of carburetion or handling of the fuel in the intake that is capable of giving us the maximum possible utilization; so, are we not really wasting time when we try to judge the merits of several fuels from results obtained with them in carbureting apparatus that is admittedly only indifferently good? It seems to me that the data offered this morning tell more about the carbureting devices used than about the fuels that were presumably on test.

T. J. LÍTLE, JR.:—At the meeting of the American Petroleum Institute at Chicago the automotive industry was told, I understood, that the fuel of the future would be heavier and less volatile than that which we are getting at present, and to get ready for it. I think the most important work to consider in research is getting ready to use the heavier fuels. How many companies have done work along that line? Has Dr. Dickinson information as to how far we may be expected to go in that direction? We were told flatly that it was anticipated that we would have to use fuels of an end-point higher than 500 deg. fahr., and I am wondering how many companies will be ready to utilize these fuels when they get them.

CHAIRMAN H. M. CRANE:—In my opinion the results of the tests that are now being made will have the greatest possible bearing on the use of still heavier fuels, if we are finally forced to use them. We hope the comparison in actual service conditions of a number of different devices for handling the present fuels, which are fairly heavy, and the results obtained with them, will give us some kind of a curve that will indicate the utilization value of petroleum distillates of different volatility in engines of the present general type; that is, in general, four-cycle engines having float-feed or similar carbureters, various metering devices and more or less simple manifolds, with the application of heat in the quantities available. It is as necessary for us to know that as to know what we would have to do if we had a much heavier fuel. We may make it very plain before we get through that the fuel must be modified to suit the needs of the present general type of engine, or the present general type of engine must be completely changed.

I have spoken many times of the fact that the attempt to use a different type of engine is not a new thing; it is one of the oldest things in the industry. It has been carried out under the urge of an immense financial advantage to be able to use heavier fuel. From the time that we first began using gasoline commercially in automotive vehicles, the spread between the cost of gasoline and the heavier fuels has become greater and greater, at least until very recent years, and it has always been so great as to present a tremendous inducement to any one to use the heavier fuel if he could do it and give the service in so doing. There is the same inducement today. That is the reason I have felt that it is very probable that there is more hope in the proper modification of the fuel to suit the engine that has been developed in service, which is particularly suitable for general service because of its simplicity, than in altering the engine by increasing its complication to a very considerable extent to make it suitable for use with some arbitrary form of fuel.

That is why I am glad to hear that a number of engineers of the oil companies are going to the Bureau of Standards and will meet continually with engineers from the automotive industry, with the result that the

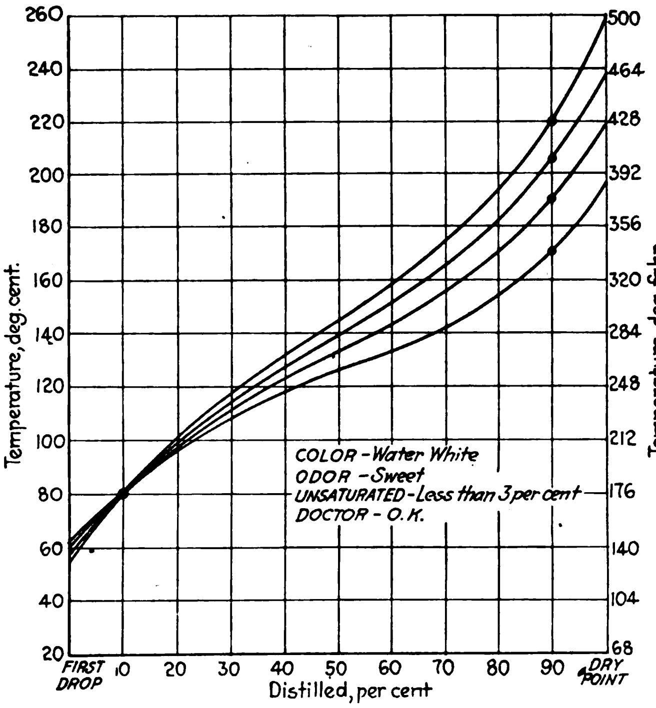


FIG. 1—DISTILLATION CURVES OF VARIOUS MOTOR-FUELS

knowledge of both ends of the problem will be much more thoroughly disseminated. I know that plenty of us do not half realize the difficulties of the production and marketing of petroleum products. I am equally sure that a great many of the petroleum people do not realize that the present enormous extent of the automotive industry today is based absolutely and entirely on the simplicity of the motive power that is supplied by the present type of engine. It is true that the education which the public is receiving from this simple engine has gradually prepared it to use a somewhat more complicated device, and the public is doing that every day. The present engine for using the present heavier fuel is a far more complicated device than the public had available in 1905 to 1908, and it is using it more or less successfully.

It is necessary to remember also that there are a tremendous number of automobiles in service today, that they are using liquid fuel very rapidly, and that they cannot be changed fundamentally. They can be changed in detail, they can have new manifolds possibly or new carbureters, but they cannot be changed into any other type of engine; they cannot be supplied with any very complicated carbureter-manifold system on account of lack of space under the hood. We all know how difficult it is to apply anything special in the form of a carbureter or manifold to a Ford, because the space is not available to put it in.

If these tests can and do succeed in clarifying the situation regarding the existing cars, the ones that are using 90 per cent of the fuel that is produced every day, they will have performed a tremendously useful service. It is wholly possible that they will result in an overall improvement of only 10 per cent in the use of fuel, but that 10 per cent would be about enough to provide the fuel for 1 year's production of cars and is very much worth getting.

I do not want Mr. Lítle to think that I say we should not do anything along other lines, but also I do not want the feeling to go out that there is imminently possible a sudden change in engine design that will make the use of heavier fuels easy and satisfactory for our kind of work,

because I am absolutely certain that there will not be present in any of those engine forms the simplicity that will meet the requirements of the widely distributed use of apparatus that is made by the automotive industry.

PROF. E. P. WARNER:—Is it planned to extend the service tests being made by the Bureau of Standards and the various companies to cover the question of mixed fuels? There is sold regularly a so-called gasoline that contains a large percentage of benzol. In Boston a great many car-owners use it exclusively.

DR. H. C. DICKINSON:—There are as yet no definite plans in this respect. It is rather difficult to lay out a series of tests that will cover a matter in which there are so many variables. Perhaps the most important characteristic of these blended fuels is their anti-knock property, which is a matter that Mr. Midgley has covered. We have considered what could be done in the way of determining the economy of various possible fuels, so far as this can be done in an experimental program, but no definite plans have been made.

I think Mr. Crane has practically answered Mr. Litle's question, but I would like to add just one thought. You will notice that the upper fuel in Fig. 1 has a 500-deg. end-point. Of course, the fuels were selected before any tests had been made. It was our guess, so to speak, that we had selected the worst fuel which would be likely to "get by" in service without offering so much difficulty that the tests would be meaningless; in other words, we hoped, after securing the results, to plot curves of crude-oil consumption versus fuel volatility which would show a minimum; that we had got beyond the point of economy. Evidently, from the tests, so far as they have gone at present, we were off on our estimates; because the cars that have been tested, and the trucks also, apparently have actually utilized the 500-deg. fahr. end-point fuel without any marked decrease in economy. Of course, that does not cover the question of crankcase-oil dilution, which may be the neck of the bottle; but, so far as the volatility alone is concerned, it looks as if we will have to make another guess, select two or three heavier fuels and do more testing along the same line before we determine the economic limit of volatility.

W. S. JAMES:—In connection with the point raised by Mr. Tice, there is one advantage in the type of work described by C. T. Coleman that I believe has not been brought out; it is that of fuel specifications. There is considerable controversy at present, at least in the Federal Specifications Board, between the petroleum refiners and the users of gasoline, as to what constitutes a suitable gasoline. The petroleum refiners maintain that the present type of gasoline is satisfactory and that the 90-per cent point can be raised with no detriment to the industry. On the other hand, the users are not sure; they do not know. This was one of the prime reasons the Bureau of Standards took this work up in Philadelphia in cooperation with the Post-Office Department. The Post-Office Department uses something like 4,000,000 gal. of gasoline per year.

There are practically no data on the advantage or disadvantage in actual service of fuels of varying volatilities, and these few tests, meager though they are, furnish at least indications as to whether the refiners' demands should be granted. Specifications that were laid down for use in Government purchases have been adopted by a considerable number of municipalities and large companies. This is a matter of great interest to the whole automotive industry. I believe that clarification of the correctness or incorrectness of gasoline specifications is one of the benefits of this kind of work. The Post-Office Department is using Government-specification fuel in about 5500 vehicles in service now. These vehicles represent an investment of capital of about $10,000,000. To change this equipment would be very expensive. It may be easier to change the fuel or the garage operation. Possibly, as Mr. Coleman has indicated, greater gains can be expected from care in carbureter adjustment and in car condition than from change in fuel.

P. J. DASEY:—Considering the rapid development of the automobile industry and the number of cars that are being built annually, it seems to me that it will be only a comparatively short time before we will have such an enormous volume of these vehicles in use that it will be very difficult to secure the proper amount of fuel. To supply the demand as the number of car-users increases, the oil companies probably will have to get a much larger production of crude or give us a larger percentage of the heavier hydrocarbons. If we are to continue producing the same types of engine that we have been building, we will find it increasingly difficult to handle the heavier fuels. It seems that the whole problem resolves into two items. One is that in the development of new engines we base the design on the necessity of using a heavy fuel, up to a 550-deg. fahr. end-point. The other is to continue the work being done by Messrs. Mock and Tice and a number of others and make devices applicable to the present-day types of engine which will enable them to handle the heavier types of engine fuel.

The experiments that are being conducted indicate that we are losing more of the heavier ends in dilution as the fuel becomes heavier. When the fuel is lighter, there is less dilution. On the other hand, we all know that we could get more power from the heavy fuel if we could handle it properly. I believe we will all agree that, if it were possible to handle 550-deg. end-point fuel, the condition would be better because of the much greater volume that would be available from our natural resources.

There is plenty of room for development in engines that would not make them radical departures or increase the cost of production materially, but go a long way toward solving the fuel problem. I will mention only extremely high compressions and operating at lower engine-temperatures. Compression in itself means heat. It has been stated repeatedly that every time the petroleum refiners introduce a series of heavier hydrocarbons into the fuel the engine builders must reduce the compression to handle it. Our experience is just the reverse of that.

I do not know of any oil company that will guarantee to give us now, or at any future time, a 420 or 428-deg. end-point gasoline without raising the price to an enormous figure. The automobile industry will not stand still. The number of cars will increase, and therefore the number of gallons of fuel we shall demand will grow enormously. We cannot grow enormously if we have a high-priced fuel.

MR. LITLE:—I think it would be a mistake to give the oil industry the impression that we can meet them no matter how high the end-point of the fuel goes. Mr. Dasey says he can handle 550-deg. end-point fuel; possibly he can, but we have about 12,000,000 automobiles and trucks running around the Country, and it will never be possible to fix them up. Naturally, that not being possible, they work in a very unsatisfactory sort of way when the end-point goes too high, or when the fuel is not sufficiently volatile.

The oil people absolutely can improve their fuel if they want to. It may cost them more to do it, but it costs the

automotive industry much to use the heavy fuel they are proposing to supply. I would rather see them crack more kerosene and mix less kerosene with the fuel. I think it is possible to work along that line, and others to whom I have spoken on the matter agree with that view. I had hoped to hear some one here, representing the fuel industry, say that if we were willing to pay a few cents more for better gasoline we would be guaranteed a supply.

CHAIRMAN CRANE:—It all comes down to the fundamental thing that the industry is fighting for. About 2 years ago I said that I refused to believe that a straight distillate of crude petroleum was the only fuel that we ought to use in an automobile engine. I did not know much about it, but on general principles I was sure that was to be the case. It took the General Motors Research Corporation to bring clearly before us the fact that it is not necessary for us to do it; that by the addition of various available products, which are not products of petroleum, the fuel that we use in our present engines can be very greatly improved, and a fuel can be made for which an engine of equal simplicity can be designed, wherewith we can expect to get enormous increases in economy and therefore increases in mileage per gallon of crude oil.

I am equally loath to believe that the refining industry has reached a point where its representatives can honestly say that they are giving us the greatest quantity of fuel of a volatility that we can use in present-day engines, that is available in a barrel of crude oil.

I am very glad that Mr. Lítle raised the question of price. I think that has been put too often before the oil people; that we must have the lowest possible price on gasoline, regardless of what the gasoline is. After all, the cost of operating a car is based not only on the cost per gallon, but equally on the mileage per gallon. In other words, the owner really is interested in how much a week it costs him to keep the tank of his car full of gasoline.

The California situation, in which they used distillate and a fairly high quality of gasoline side by side, was very significant. Certain classes of interests that were able to do so applied special equipment to their engines, especially in such cases as motorboat work and heavy trucking, where it could readily be done, and obtained economical operation from the lower-priced distillate, while the other passenger-car owners found they got more satisfaction per dollar expended out of the better grade of gasoline.

I was very much disappointed to hear last year that the refiners had decided to throw both of these products into the same tank and market them as one product. I am absolutely certain that the economic cost to the Country of doing that has been very considerable. They are now supplying a fuel that is not satisfactory for either of the two classes of trade; that is, not as satisfactory as the particular fuel before had been, and the overall price is undoubtedly higher and, equally, the mileage per barrel of crude oil is probably lower. It is the object of all these tests to bring all such facts out into the light.

I fully expect to be shown up as a false prophet by some of the recent tests, as to the ability to handle heavy fuel in some of the more modern cars. It looks as if they were doing it more successfully than I believed they could. I am entirely willing to be shown up that way also. I hope that the oil people, when they find they have been wrong in their contentions, will be willing to say so and show a disposition to change the attitude that they have previously maintained.

T. A. PECK:—I think we are face to face with certain fundamentals that we cannot talk around, desire it though we may. Speaking for the petroleum industry, we can give some of you a 78-deg.-test gasoline, if you want it; but there will probably be 9,000,000 motorists who will not get any, and they are desirous of using their cars. The petroleum industry must have the good-will of the motorist and the automotive engineer. If it is to succeed, it must strive to attain and retain this good will. The petroleum industry today is spending millions in experimenting, in enlarging its refineries, in changing its equipment and in trying to produce enough fuel to keep automotive wheels rolling. My own opinion is that there will not be any radical change in the design of the four-cycle engine we are using today for some time at least. We are all striving earnestly and ably all the time to improve the operation of the engine, because the motorist is becoming more critical every hour. I can remember the day when mixing-valves were good enough for an automobile. We seem to be getting away from the fact that, fundamentally, a carbureter is merely a metering device.

Speaking for the petroleum industry, I hope and believe we all want to do all that we can. For every step that you will take with the engine, we will take two steps to give you the fuel you want. But we must have the raw stock out of which to make it; please remember that it is impossible to get gasoline without crude oil.

FOREIGN TRADE

BY skillful handling of foreign credits, continued investment in foreign loans by the American public and a willingness to accept payment of our debts through the importation of goods, we may be able to sustain our export trade at approximately its present volume. This is of even more vital importance to our farmers and producers of raw materials than to our manufacturers. It is this question of the willingness of the American people to accept payment in goods and services of the debts owed them from abroad that is of the greatest moment in the present phase of the tariff discussion. The uncertainty of public opinion as to the most desirable kind of permanent tariff is evidence of the changing ideas that the condition of our foreign trade is forcing upon us.

The effect upon our foreign commerce, and especially upon our exports of foodstuffs, which compose about 30 per cent in value of our total exports, should be the major consideration in making up our new tariff. The producers of agricultural commodities are vitally interested in maintaining and advancing our export trade, for they produce a surplus over domestic consumption. It is, therefore, essential that under the new conditions of our position as a creditor nation we should take every reasonable precaution against unduly obstructing the growth of our imports of merchandise, upon which depends in such large degree our ability to sell abroad our surplus of domestic products.

Ultimately our imports may exceed our exports, even if the change to an import trade balance be deferred through the extension of American capital in investments abroad. Spread over a period of several years, this change can occur without disturbing our domestic, industrial or agricultural interests, and without decreasing our exports. If, in the course of time, such a change does not take place in our merchandise trade balance, some other means will have to be devised by which foreign debtors can pay what they owe us.—Guaranty Survey.

Research Topics and Suggestions

THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.

HOLDING THE ROAD

THE present demand for safety in transportation in general, and on the highway in particular, calls for much better knowledge than is had of some of the many factors that affect the safety of highway traffic. One of the most important of these is the ability of the vehicle to hold the road or to avoid skidding. A survey of the literature yields almost no information on the hold of tires on any sort of road surface, or on the effects of water and other agents on the road surface.

For convenience in discussing this problem, we shall use the term *hold* to designate the coefficient of static or sliding friction as the case may be, between the tire and the road, or, in other words, the resistance that is offered to sliding in any direction. For clearness, it is necessary to distinguish between this and the so-called rolling resistance or rolling friction of the wheels on the road. The latter is being studied by highway engineers as a part of a general program covering the tractive resistance of vehicles.

There is no direct relation between the tractive resistance of a vehicle, i.e., the force required to drive it in the ordinary way and the resistance offered to sliding, either transversely or straight ahead, when the wheels are locked. It is the latter sort of resistance only that we have designated as the hold on the road, which we wish to discuss here.

It is common experience that resistance to skidding may have two very different values even on the same road surface. The maximum amount of retardation of a vehicle is almost always obtained by applying the brakes only to such a point that the wheels just do not lock. When the wheels begin to slide, the retardation becomes less, often very much less. The reason for this is that so long as the wheels are not sliding there is static friction between the tire and the road. Under these conditions each portion of the tire that comes into contact with the road surface remains at rest there, with no relative motion or sliding in the plane of the road surface. As the static-friction coefficient is almost always greater than that of sliding friction, the maximum hold occurs when there is no sliding.

The difference between static and sliding friction has an important bearing on the problem of the hold on the road, as will appear later. Under some conditions this difference becomes even more pronounced than is the case normally. This often occurs when the hold is none too good, as on wet pavements. If sliding or skidding starts under these conditions, it is not easily stopped.

The lack of information on the subject covered by this survey may be due to the complexity of the problem. Obviously, the hold on the road is dependent not only on the nature of the road surface, and the tire surfaces, but also upon such things as road contours, tire inflation, spring-suspension and load distribution of the vehicle. There are a few figures for the distance in which a vehicle can be stopped on a dry hard road, from which a figure for the hold on the road can be deduced, but these do not apply to wet or slippery roads. The ideal road-tire combination would be one for which the hold is the same whether the road is wet or dry, and whether the wheels are rolling or sliding. This ideal condition does not exist and it is doubtful if it can ever be produced. But there is every reason to hope that it can be more nearly realized than at present, and we shall attempt a brief survey of some of the factors that affect the hold of tires under road conditions, in order to suggest, if possible, where progress can be made, or at least where further information is needed and where it may be obtained through further research.

The general factors which determine the hold are three

(1) The materials in contact; rubber for the tires and earth, concrete, asphalt, gravel, etc., for the road, with or without a layer of mud between them

(2) The smoothness or roughness of these surfaces; tire finish, either smooth or non-skid; road surface, smooth or rough, concrete or gravel

(3) Contour of the road surface, or what is usually termed roughness, as it is felt by occupants of the vehicle.

A word of explanation may be needed to make clear the more or less artificial line drawn between roughness of the road surface and roughness of the road, because the two have opposite effects on the hold of the vehicle. By surface roughness we mean such irregularities of surface as are too small to cause any appreciable vertical motion of the wheels, but are entirely "smoothed out," so to speak, by the flexibility of the tire, as for instance, a gravel or broomed concrete surface. Whereas by road roughness, or roughness of contour, we mean such inequalities of surface as cause vertical motion of the wheels or flexure of the springs. There is, of course, no sharp distinction between the two degrees of roughness. In fact, what may be only surface roughness for tires with low inflation pressure may constitute roughness of contour for tires carrying a higher inflation-pressure. A well-laid Belgian-block pavement is a good illustration of this effect. But the action of the surface, as regards the hold on the road, will also be entirely different in the two cases.

MATERIALS IN CONTACT

As for the character of the materials in contact, rubber and steel are the only two materials used in ordinary practice for vehicle tires and, for the purposes of this discussion, rubber is the only material that we need to consider. Road surfaces, on the other hand, may be composed of any of a large number of materials but, if we confine our attention to improved roads, these materials all possess the common characteristic of being rigid as compared with rubber tires.

If there were always good contact between the road and the tire, the problem would be much simpler, but in practice the two surfaces are often separated by a lubricating film of water, containing varying amounts of mud and other substances that even improve its lubricating quality.

The coefficient of static friction between rubber and most road materials, when dry, is ample for almost any purpose.

REAR-WHEEL-BRAKE STOPPAGE ABILITY

Speed, m.p.h.	Distance Required for Stop from an Actual Test, ft.-in.	Standard Given by Thermoid Rubber Co., ft.-in.
10	9-0	9-2.4
20	34-11	37-0.0
25	53-7	58-0.0
30	74-5	83-3.6
40	135-6	148-0.0
50	178-0	231-0.0

The figures on the preceding page were given for the distance in which a car can be stopped by brakes on the rear wheels only, when these wheels carry less than half the total weight of the car in a paper by J. Edward Schipper entitled Passenger-Car Brakes.[1]

On the basis of these figures, the coefficient of friction is less than 0.7. Other tests on dry concrete pavements have shown a coefficient of friction of about 1; that is, to start a car with all wheels locked would require a force equal to the weight of the car. However, on a wet surface, the coefficient of friction may have almost any value down to perhaps 0.01. The liquid layer acts as a lubricant and follows the natural laws of lubrication. Thus, the characteristics of a good bearing should be those of a bad road-tire combination, and vice versa.

The journal to be lubricated by a light or non-viscous lubricant like water should have a smooth surface, a very low bearing pressure and a high peripheral speed. While the lubrication of a plane surface is not entirely analogous to that of a journal, these characteristics are a dangerous condition for a highway. The various anti-skid devices, such as special tire treads and tire chains, are designed to decrease the area in contact and thus to increase the unit pressure between the tire and the road surface. The higher this pressure, the more readily is the water layer forced out of the way. Also, the smaller the individual contact areas, the better from this point of view. As regards the material of the road surface, much the same is true as in the case of lubricated journals; so long as there is lubricant present, the material of the surface is of very minor importance. So the main essential for hold on the road is that the lubricating film of water or mud shall not remain between the tire and the road surface.

Condition of Surfaces

It is in connection with the persistence of the surface film of water that the surface roughness of the road material probably becomes most important. A good bearing-surface must have no surface roughness that has a depth equal to the thickness of the oil-film; otherwise there will be a metallic contact and excessive friction. By the same token it appears that a good road surface should always have a surface roughness greater than the thickness of the water-film, so that there would always be contact with the tire through the water layer.

We recognize that this is only reasoning by analogy, but we venture to do so for lack of any real information as to the relation of the surface roughness or smoothness of the road and the hold of rubber tires on them when wet. Because of this lack of information and of the evident importance of the subject, we believe there is need of a systematic research program that will furnish information on this and on other points to be mentioned later.

Up to the present time, non-skid devices, such as tread forms and chains, have been applied to the vehicle. Is this the correct place for them? Can the road surface be designed so that it will have the desired non-skid quality or give the desired hold on rubber tires? May it not be practicable to make a better non-skid surface on the road than on the tire? The road material being rigid, if a given type of roughness is found to be desirable and practicable to produce, it will retain its shape under load, which formed rubber will not do.

Even in the matter of non-skid tire surfaces, innumerable varieties of geometric forms are used. Is any one of them better than another? What are the principles governing the effectiveness of non-skid treads? It seems that the answer to the foregoing and many other questions of practical importance in the design of roads as well as of tires, can be answered only by a systematic research program, such as is suggested here. The question of surface is already being considered by the highway engineer. It seems that the tire makers and the builders of highways could get together on this problem with the makers of vehicles to their mutual advantage and to the great advantage of the general public who use both.

We shall not attempt here to suggest the motive or scope of such a research program; suffice it to say that such a program need be neither long nor costly to yield important results in a virgin field.

Road Contour

As for the effect of road roughness or surface contour on hold, it is common experience that a vehicle may skid on a perfectly dry road, the surface of which has become corrugated. This is merely an exaggeration of what occurs on all roads that are not perfectly smooth as regards these major irregularities to which we have referred as road roughness. On even fairly good roads, the wheels often leave the road surface. While a wheel is in the air, it has no hold, of course and, when it returns, the static hold that it should have on an ideal surface may have been reduced to a sliding hold, which is far less effective, especially on a wet surface; in other words, the bouncing of a wheel starts a skid so far as that wheel is concerned.

But even if the wheel does not leave the road surface, the instantaneous pressure upon the road may vary from zero to many times the actual weight carried by the wheel. Whenever the pressure is reduced, the hold or resistance to skidding is reduced in about the same proportion. Here logic tallies with experience, that skidding is promoted by even slight irregularities in road contour, particularly at high speeds. The importance of this factor depends, of course, upon many things, such as tire-tread design, inflation-pressure, sprung and unsprung weight and the general balance of the vehicle. A study of the effect of the surface on the hold would be incomplete without some information on the general effect of road-surface contours.

The results of such a study are likely to reinforce the latest conclusions of highway researchers, that road roughness of this kind is one of the things to be avoided at all costs, because of the destructive effect on the road of the bouncing of the wheels. The tendency to promote skidding should be an additional argument, if one is needed, for studying road surfaces.

[1] See THE JOURNAL, April, 1922, p. 280.

EFFECT OF RATE OF HEATING IN HARDENING STEELS

ALTHOUGH the rate of heat for hardening is apparently of considerable importance, at least in some steels and under certain conditions where the control of dimensional changes in quenching is desired, it has rarely been accurately regulated. This question is therefore of vital importance to manufacturers of die-castings and in the production of thread gages. At the Bureau of Standards recently a series of 16 samples of chrome-vanadium die steel was hardened using various heating rates and the dimensional changes noted. The desired uniformity of results was not always obtained but in view of the discrepancies found in other types of test on the same steel, it is believed that the metal did not have the required uniformity. Similar tests of different steels are now being carried out.

The Aluminum-Alloy Piston

By JAMES E. DIAMOND[1]

DETROIT SECTION PAPER *Illustrated with* CHART

AFTER stating that the piston and its rings are expected to maintain compression, keep the gases of combustion within the combustion-chamber and prevent the heavy unburned ends of the fuel from diluting the lubricating oil in the crankcase to as great an extent as is possible, the author mentions the relation of the piston to compression and states the necessity that the piston shall move in the cylinder within the closest possible clearance-limit and discusses the thermal characteristics of aluminum in their relation to these requirements. Piston and cylinder relations are considered, inclusive of wear, and aluminum alloys suitable for pistons are discussed, tabular data and a chart relative to heat-treatment being given. The progress of design is outlined and ring-groove wear is commented upon, together with other factors affecting piston design.

IT is only comparatively recently that real progress could be reported in regard to the aluminum-alloy piston but, in view of my early connection with its commercial development and a rather intimate connection with it since then, I will sketch its history briefly, including past experience and subsequent development.

As a first step in recording the progress that has been made since the alloy piston came to be considered commercially, I will analyze the problem by stating the function of the piston, the service required of it and what happens to it while rendering this service. Admittedly, this is very elementary treatment. Aside from the mechanical considerations involved, the piston and its rings are expected to maintain compression and to keep the gases of combustion where they belong. Further, this combination is expected to do its share in keeping the lubricating oil out of the combustion-chamber and the unburned heavy ends of the fuel out of the crankcase lubricating oil to as great an extent as possible. These are the fundamental considerations, but there is an incidental one of rather vital concern in the relation of the piston to compression. A requirement that will help to meet the major premise is that the piston shall run in the cylinder within the closest possible clearance limit. The point as to compression is that the more readily a piston is able to yield up the heat absorbed by it during the impulse stroke and, likewise, the heat created during the adiabatic compression, the higher the compression can be without causing the phenomenon generally known as detonation.

Broadly speaking, it is a matter of general knowledge that the alloys of aluminum have thermal characteristics highly favorable to meeting the latter condition; and, at the same time, another characteristic that, in the past, has not been particularly favorable to meeting the former. In other words, the high thermal conductivity of aluminum is of great usefulness, but alloyed aluminum might have even greater usefulness as a material for the manufacture of pistons if the coefficient of thermal expansion were of smaller magnitude. Occasionally, it is rumored that a non-expanding aluminum alloy has been developed; but, so far, substantiation has always been lacking. As a matter of fact, it may come to pass that we shall find in this particular characteristic a valuable attribute.

Turning now to those difficulties experienced in the past with the alloy piston that are directly attributable to the hitherto not particularly desirable characteristic of high thermal conductivity, the main problem has been to fit the piston with a clearance sufficiently small to insure silent operation without piston slap at all times, and to keep the lubricating oil out of the combustion-chamber, yet at the same time to maintain sufficient clearance between the piston and the cylinder wall to avoid the danger of piston seizure and possible scoring in the upper speed-range. It is only fair to state that this problem is not present in every case. Engines, like human beings, seem to possess individuality. Another difficulty probably indirectly created by this same propensity of expansion has been the matter of wear in the ring-grooves.

PISTON AND CYLINDER RELATIONS

Attention is now directed to consideration of the relation of the piston and the cylinder. The problem of wear is the dominant one. If the problem is approached with an open mind, it will be conceded that much of the criticism heaped upon the alloy piston in years past has been entirely unwarranted in many respects and perhaps none more so than in this particular one. If the employment of the alloy piston is justified at all, and I hold that this is not open to argument, it is the duty of those concerned to see that this piston is put to work under favorable conditions; in other words, to see that the cylinder is properly prepared. In the last analysis this is squarely up to the engineer. His foremost consideration should be to see that the ideal condition is approached as nearly as possible. It is imperative that as finely finished cylinders as possible be produced. Perhaps it may cost more to accomplish this, and it may even be necessary, to get the best results, to lap-in the cylinder with special equipment approximating the running-in with iron pistons. I mention this because the best results I have ever observed in the way of minimum piston and cylinder-wear have been where alloy pistons have replaced iron pistons in cylinders that had been run-in with the latter. The alloy piston demands this consideration as its right, entirely regardless of the fact that a slight additional initial expense in producing a better bore and finish probably will be more than offset in the long run in the way of performance and service. The preceding has all related to a consideration of what might be termed avoidable wear. There are, of course, other angles to this matter of wear, but the data I have collected are so contradictory that I will draw no definite conclusions, although some of this contradictory evidence is submitted herewith as a matter of general information.

Much has been said recently about the abrasive nature of road-dust and its destructive tendencies, when drawn into the engine through the intake system, with special reference to its lapping effect and resultant piston and

[1] M.S.A.E.—Vice-president and general manager, National Piston Co. Inc., New York City.

cylinder-wear. This is undoubtedly a factor, but it is difficult to say how important a one. During the war I spent some time at one of the plants of the Holt Mfg. Co. The records show clearly that, in agricultural work, were it not for the air-cleaner with which its engine is equipped, it would be necessary to replace pistons and regrind cylinders several times each season. Contrasted with the foregoing, some experimental work has been done recently by an organization having unique transportation problems, to determine the amount of road-dust drawn into the intake system. This has shown an accumulation of less than 0.500 oz., or 0.032 lb. in a 3000-mile test. This is 0.010 lb. per 1000 miles, or about 75 gr., which is only a rather small teaspoonful. While this dust was not analyzed to determine the exact proportion of abrasive material, the presumption is that it was high, because the experiment was conducted over city streets. It would not seem that this amount of dust drawn into an engine per mile of service and divided among a number of cylinders would cause appreciable wear. Again, there is no certainty that most of this dust might not be blown out during the exhaust stroke. So, here we have two opposing results. If it is decided ultimately that this is a major cause of piston and cylinder wear, the employment of an air-cleaner or filter will be recommended strongly.

Other interesting facts have been noted in connection with this matter of wear. One is that the results are not at all consistent and a precise explanation of this inconsistency is still lacking. In some cases a piston shows no wear after considerable mileage, but the cylinder shows appreciable wear. Ordinarily, the wear is divided. Recently I examined a set of pistons approximately 5 in. in diameter that had been taken out of a truck engine after the truck had run approximately 80,000 miles. Strange to say, no measureable wear was discernible and the cylinder wear was normal. It is not outside the realm of possibility that some permanent growth compensated for the wear that actually took place, although this is more or less discounted by the fact that the original tool-marks were still visible, and that these particular pistons, which were of the Franquist type, were cast from an alloy that does not seem to have any tendency to grow under ordinary operating conditions.

Concerning the relation between piston hardness and wear, the results at present are more or less inconclusive because so many factors are involved. The Brinell-hardness test apparently does not tell the whole story. I have been loath to believe the tendency toward greater hardness a correct one, and I think that unusual hardness can be gained only at the sacrifice of other desirable qualities. This judgment may be entirely wrong, but I have not yet forgotten some disastrous experiences in the early days of development. In some respects the service exacted of a piston is related to that exacted of a babbitt bearing, which is soft and yet has remarkable wear-resisting qualities. Further, recent experimental work seems to confirm the opinion expressed previously in regard to the not entirely conclusive nature of the Brinell test. However, this experimental work has not been entirely completed, and so it is not safe to draw definite conclusions. These experiments have been conducted with pistons made from two of the newer alloys and, while one of the alloys shows only a maximum hardness of 55 and the other a maximum hardness of 65, in the case of neither alloy is there any measureable piston-skirt or cylinder wear. The condition of the ring-grooves may tell another story ultimately.

Further, I am convinced that design is a factor also. In connection with some experimental work being conducted with pistons of one of the newer designs, perceptible and therefore excessive wear has been noted in less than 3000 miles. This result is not based on the results with a single engine, but with two distinct groups of engines. It is interesting to observe that the wear was actually much greater than was the case with pistons of conventional design, but which could not be used because of inability to eliminate piston slap. An analysis of the design in question convinces me that there are elements in it that are not sound and contribute to the results noted.

Aluminum Alloys

The goal from the very outset has been to produce an aluminum alloy that would have those characteristics of the iron piston that are good and none of its bad ones, still retaining all the marked advantages inherent in aluminum as a basic material. Part of the problem is metallurgical, and part of it relates to engineering. The responsibility of producing an alloy having the proper strength and wearing qualities, one that would retain its hardness, would still be soft enough not to score the cylinder if seizure takes place, would prove entirely stable, would dispose of the maximum quantity of heat and possess all of these qualities at high temperature, was and is primarily within the province of the chemist. Proper design constitutes the responsibility of the engineer.

In reference to the metallurgical aspects of the problem, I reached the conclusion some years ago that the aluminum-nickel-copper-magnesium class of the alloys known then was the best suited as a material for the manufacture of pistons. No profound reasoning was involved in drawing this conclusion, since it was based on the observation that pistons cast in this class of alloy almost invariably seemed to give better accounts of themselves.

Another very valuable fact was discovered in the laboratory while settling an argument concerning heat-treatment and piston growth. An extensive series of experiments through a wide range of temperatures disclosed no tendency of this alloy to grow. The tendency was rather the reverse, strange though this may seem, and was wholly in contrast with the results obtained in a similar series of experiments with a piston cast from a different alloy in which growths as much as 0.002 in. per in. of piston diameter had been established. While the work had the determination of the proper treatment to insure initially a maximum permanent growth as its object, in addition data as to distortion were gathered. This also was ascertained to be a negative quantity. In passing, this phenomenon of permanent growth first came to my attention in 1915, in connection with its occurrence in the alloy pistons used in Stutz racing engines. It appeared at the time that this phenomenon would have an academic interest only.

With the above problem particularly in mind, a vast amount of research work in connection with light alloys has been done within the past 4 years. Probably Dr. Walter Rosenhain, a foremost metallurgist in Great Britain, has delved most deeply into various phases of the work, assisted by the staff of the British National Physical Laboratory. After a most thorough investigation of the various known alloys, a vast amount of original work in the development of new ones and an exploration into the possibilities of all classes down to the last detail, Dr. Rosenhain finds that an alloy having the following general analysis: copper 4.0, nickel 2.0, magnesium 1.5 and aluminum 92.5 per cent, has by far

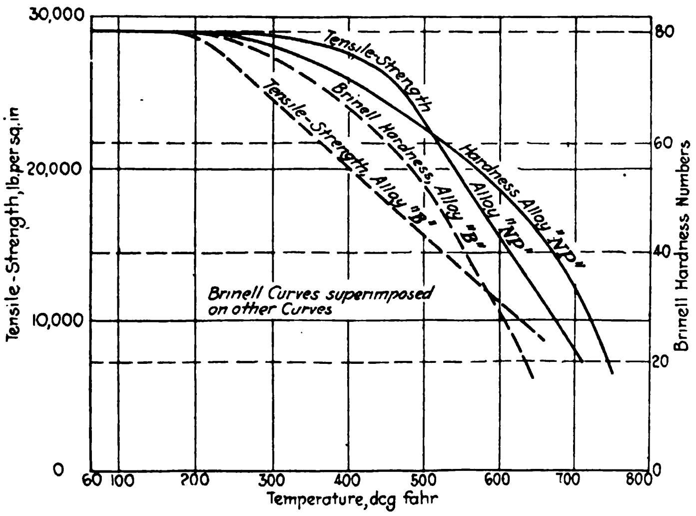


FIG. 1—CURVES SHOWING THE EFFECT OF THE TEMPERATURES ENCOUNTERED IN OPERATION UPON THE TENSILE-STRENGTH AND THE BRINELL HARDNESS OF TWO ALUMINUM ALLOYS

the greatest possibilities of any alloy thus far developed where work must be performed under an incubus of high temperature, not excepting duralumin. An alloy of this general analysis was developed in Germany approximately 20 years ago; but, apparently, its possibilities had not been explored so exhaustively before. This particular type of alloy has many characteristics that are desirable in an alloy piston particularly. This is the explanation for the improved performance previously described as having been noted with pistons cast from this alloy. Its physical properties are remarkable in a number of respects, none of which is more remarkable than its response to heat-treatment. In connection with the data given in Table I and the curves shown in Fig. 1, I had hoped to present figures showing to what extent the remarkable properties acquired by heat-treatment were affected while actually being subjected to temperatures approximating those prevailing in a piston while operating in an engine; but these are not available at present. The 2-per cent elongation of this alloy is taken at 60 deg. fahr. The decrease in tensile strength of it up to 400 deg. fahr. is negligible. Its tensile-strength decreases slightly above that temperature and the Brinell hardness is reduced as shown.

TABLE 1—PHYSICAL CHARACTERISTICS, AS CAST. ELONGATION 2 PER CENT

Temperature, deg. fahr.	Tensile-Strength, lb. per sq in.	Brinell Hardness
60	29,000	80 to 90
300	......	75
400	......	67
500	24,500	60
600	17,500	51
700	11,000	30
750	......	17

This alloy can be given various tensile-strengths by heat-treatment, from 33,500 lb. per sq. in. after a simple treatment to 47,000 lb. per sq. in. after a more complex one. This alloy is stable. Duralumin apparently is not. It is known that all of the acquired characteristics are not lost under the operating conditions in an engine; therefore, there is undoubtedly some advantage in the treatment. It can be presumed that the required properties will be affected much less in the skirt than in the ring-carrying portion of the piston. Of course, it is possible to obtain a much higher "as-cast" hardness, and an even greater hardness after heat-treatment with other alloys higher in copper. This can be done also with the alloy in question by making slight changes in the formula but, apparently, this is accomplished only at the expense of other more desirable qualities. Toughness is sacrificed for brittleness; consequently, such alloys are not particularly desirable ones to handle in the plant. Further, all the evidence indicates that, with the present known alloys having high hardness at normal temperature, the drop in hardness is very rapid with an increase in the temperature. Considering one of the other alloys used as a piston material with an initial as-cast Brinell hardness of approximately 80 at normal temperature, merely for comparison, this drops to 72 at 300 deg. fahr., 63 at 392 deg. fahr., 34 at 500 deg. fahr. and 13 at 750 deg. fahr. The tensile-strength of approximately 29,000 lb. per sq. in. as cast, drops to 15,700 lb. per sq. in. at 500 deg. fahr. and to 9000 lb. per sq. in. at 650 deg. fahr. This all has a great bearing on wear, particularly on ring-groove wear.

Having surveyed the phases pertaining to past experience with the conventionally designed alloy piston broadly, I will now discuss the progress in the field of design made within the same period. Probably the greatest advance has been made in the solution of the most troublesome problem, that of coping with expansion so as to prevent piston slap and piston seizure at all temperatures. Three attempts to determine piston temperatures were made in the very early stages of its commercial development; among these were two that, in three instances, personally known to me, really gave valuable data. The method used in two of the three experiments involved the soldering of small balls with solders of various melting-points at various spots about the interior of the piston. One fact particularly impressed me at the time, this being that the experiment showed a piston-head temperature of approximately 600 deg. fahr. and a skirt temperature several hundred degrees less. This did not harmonize with the belief commonly held then that, due to the high thermal conductivity of aluminum, the skirt temperature would be not very much less than that of the head. The other method was the insertion of polished steel discs in various parts of the piston and, upon completion of the experiment, the comparison of the discoloration with a standard color-chart.

RING-GROOVE WEAR

Among the other phenomena noted was the progressive decrease in ring-groove wear from the top of the piston down. This revealed the nature of the heat conditions prevailing in the ring-carrying portion of the piston, and showed that a tremendous amount of heat was being dissipated, or conducted into the cylinder, through the rings. Some further indications of the rapid drop of temperature from the top to the bottom in the ring carrier were given by the condition of felt packing placed back of the rings. This was in connection with some experiments made with a view to the elimination of oil-pumping. The felt back of the top rings charred badly in course of time; back of the second ring it did not char nearly so much and back of the third ring charring was absent. Incidentally, the general accuracy of the piston temperatures shown in the ball-and-solder experiments was confirmed in the experimental work of much more delicate nature and having exactly the same objective that was sought at the British National Physical Laboratory and attained just before the end of the war.

However, the significance of all of the foregoing considerations did not impress or influence the industry at

the time.[^] A number of conclusions were drawn as to what was actually effected by the partial separation of the head and the skirt. Among these was the conclusion that such separation interrupted the flow of heat from the head to the skirt somewhat in the same fashion that a check in a piece of steel visibly retards heat-flow as shown by the color at each side of the check. Analysis and a simple experiment showed that this conclusion is not tenable since, while the heat-flow was retarded, it was not checked permanently. An attempt had been made only a short time previously to arrive at some formula for determining piston-skirt clearances based on an average skirt-temperature of 250 deg. fahr., this figure having been found as approximately correct in the British National Physical Laboratory experiments. The calculation was simple enough, since the temperatures of the piston and the cylinder and the respective coefficients of expansion involved were all known qualities, but the clearances revealed by this mathematical determination as being necessary were in every instance less than those required in actual practice, in some cases by several thousandths of an inch. The only apparent conception that explained this was that the radical expansion of the head was so great that it carried the skirt wall, expanding to a much less degree with it. This hypothesis is supported by the fact that a well-designed piston, but machined with the skirt tapered an excessive amount from the bottom of the skirt to the top, can be fitted without the least trouble at the lower end with the clearances theoretically determined. The magnitude of the expansion force exerted by the head is indicated by the fact that, in many cases, an examination of the scored area discloses some evidence of fusion, this signifying a tremendous friction-loss. It should be accepted as fundamental that the skirt must be relieved wholly or in part of all stresses created by this mechanical expansion.

Piston Design

This discussion of development would not be complete without mentioning the design of piston well known as the Liberty type. I have never felt that this had any place in the automobile engine. It was created to meet a particular condition, and, in the light of present-day knowledge, it probably would meet it just as well if the design of the skirt portion had been along conventional lines, disregarding entirely the question of partial head and skirt separation. As far as ribbing is concerned, I went on record in 1915 that a piston is usually better without ribs, or rather with the equivalent metal used to increase head thickness. Ribbing usually is suggested only when die-operating conditions must be met.

In addition to the development of the free-skirt piston, it is felt there has been a marked advance in design as to ring employment; relative to the difficulties occasioned with ring-groove wear. As is known, a series of loose rings makes an admirable pump. It is only recently that any real thought has been given to the matter of ring equipment. When the alloy piston was adopted, usually the same rings were employed as had been used on the supplanted iron piston. The combination that generally resulted was not a particularly favorable one. From the data given in Table 1 on hardness and how hardness is affected by heat, it will be appreciated that, under operating conditions, in addition to an increase in groove width due to expansion, there is also a considerable decrease in the hardness of the material in this part of the piston. Rings used with iron pistons usually have had and still have generous sections; so, here was a relatively heavy mass, of greater hardness and constantly resisting translation, in effect an anvil, on which the piston land or groove sides were hammering several hundred times per minute. In course of time, this peening action produced the wear noted, it being progressively less from the top ring down and having a direct relation to the decrease in piston temperatures from ring to ring, and to the state of hardness also. While the value of an alloy not susceptible to high temperature is at once apparent, the general situation has been improved materially, since the condition was recognized, by the perfectly logical step of decreasing the ring mass as far as possible; consequently, decreasing whatever resistance there is to translation due to inertia by reducing the width of the ring face and increasing the ring wall-thickness or depth to as great an extent as possible. In other words, the destructive peening force has been reduced materially, and a very much larger area has been presented to withstand it.

So far as fundamentals are concerned, the considerations involved in correct piston design are exactly the same today as in 1915, at which time I developed and presented formulas for a standardized design in a paper on Aluminum Alloy Piston Design.[2] In the main the experience of the past 7 years has vindicated the correctness of the design then laid down. In discussing E. G. Gunn's paper on Aluminum-Piston Design,[3] I advocated the use of much narrower rings and more of them and then continued by saying:

> I am more confident than ever that the aluminum-alloy piston is a sure thing. We may still have some troubles with it, but some one will solve them. . . . . I think this piston will be very much along conventional lines, yet embracing some special feature that will permit close fitting and at the same time allow the piston to yield when it would ordinarily seize.

[The discussion of this paper is printed on p. 228.]

[2] See TRANSACTIONS, vol. 11, part 1, p. 238.
[3] See TRANSACTIONS, vol. 15, part 1, p. 214.

Piston-Rings and Ring Grooves

By C. R. Manes[1]

Detroit Section Paper *Illustrated with* Diagram

The author defines the purpose of a piston-ring for an internal-combustion engine as being primarily accomplished by such a design as will, with the aid of lubricating oil, seal the clearance between the piston and the cylinder wall in such a manner that the charge in the cylinder will be compressed and the full power of the explosion utilized to force the piston downward on its impulse stroke.

The leakage of gas and oil past the piston is discussed and a diagram is presented and explained in connection with the action of piston-rings throughout a cycle of engine operation. Desirable piston-ring factors are mentioned and the advantage of multiple-piece rings over those of plain one-piece type commented upon. A table giving width and depth dimensions for piston-ring grooves is included.

In a paper by H. H. Platt entitled Important Factors in Piston-Ring Design,[2] that was presented at a meeting of the Washington Section, the following sentence struck me as being a rather strange statement:

> The purpose of piston-rings in an internal-combustion engine is to reduce to a minimum the leakage of gas from, and the seepage oil into, the combustion chamber

I doubt that this is a proper definition of the purpose of a piston-ring in an internal-combustion engine. Would it not be better to say that, primarily, the piston-ring is designed so that, with the aid of the lubricating oil, it will seal the clearance between the piston and the cylinder wall in such a manner that the charge in the cylinder will be compressed and the full power of the explosion will be utilized to force the piston downward on its impulse stroke? That appears to me to be a correct definition. The radiation of heat and the prevention of gas seepage and oil-pumping, although extremely important and of about equal importance, are purely secondary functions of the piston-ring. We read many statements about "bending-moment stresses," "uniform radial expanding forces" and similar terms that are used by one-piece-ring advocates, but we note that, while they all admit that the most serious losses occur by seepage of gas from, and oil into, the combustion-chamber and that this seepage is back of the ring, they never even suggest a remedy for these serious difficulties.

Gas and Oil Leakage

Mr. Platt states also that

> There are three possible paths for the leakage of gas past a piston-ring. Putting the least important first, these are (*a*) through the gap, (*b*) around the back of the ring, and (*c*) past the face

Why does Mr. Platt consider the seepage back of the ring of less importance than the seepage past the face? That depends upon how well the ring is lapped-in and also upon how great a clearance has been worn by the reciprocation of the ring in the groove. It is a safe statement that the greater seepage is past the face in an engine that has been run less than 1000 miles, and it is just as safe a statement that the seepage is greater back of the rings in an engine that has been run a greater mileage. In the case of replacements of new plain rings in worn grooves, the seepage is always greater back of the rings; it cannot be otherwise. One would think that piston-rings alone were responsible for gas seepage. In fact, 75 per cent of the gas seepage is caused by leaky valves that necessitate the use of a heavy mixture to make all cylinders fire. Of course, a certain amount of gas is not vaporized and, consequently, not consumed; therefore, it naturally flows down the cylinder wall into the crankcase. Care for the valves, use good rings, adjust the carbureter correctly and be careful not to abuse the choke in starting, and gas seepage of any consequence will cease to be. In other words, stop the sucking of liquid gasoline into the cylinders and there will be none there to seep.

One-piece rings have bevel joints, step joints and many kinds of lock joints, but what does the joint matter when there is an open gap from 0.001 to 0.010 in. wide all around the ring? Hence, I relegate all one-piece rings to the same class, because not one of them even pretends to close the door to the gas seepage or oil-pumping that is made possible by the inevitable wear on rings and grooves.

Regarding oil leakage, Mr. Platt says:

> It is of great importance to reduce to the minimum the volume of the space in the groove not occupied by ring material. Any such space acts as a first-class oil-pump and tends to fill with oil at the bottom of the stroke, where the conditions are those of suction and abundance of oil, and to empty at the top of the stroke when the oil is driven from the space by high-pressure gas. This fact eliminates the eccentric ring from serious competition and makes it important that the concentric ring should fill the groove as completely as safety permits

Let us see how far this line of argument is applicable. Fig. 1 shows a piston of a four-cycle engine at the end of each stroke; the shaded area indicates oil which has been scraped from the cylinder wall and shows what becomes of it. Mr. Platt is right thus far but, while I am not an advocate of eccentric rings, his argument against them is wrong because the space back of the ring is of no significance whatever. The reciprocating space is all-important, because no more oil can pass the rings in this way than the reciprocating space will allow. By being greater than the ring, the depth of the groove will not cause leakage of gas, compression or oil, unless the clearance has increased. But the trouble with the eccentric ring is that it is deep in the center and shallow at each end; therefore, the wear will be unequal and the wear in the groove will be unequal also. Hence, it is plain that the deeper the groove is and the deeper the ring, the less the wear on both the ring and the groove will be and, consequently, the longer the satisfactory operation of an engine so equipped will continue.

Following the piston in the diagram at the left of Fig. 1 through the four strokes, the diagram shows what

[1] Detroit distributor for the Pressure-Proof Piston-Ring Co., Boston.

[2] See The Journal, September, 1921, p. 195.

has taken place on the impulse stroke. On the exhaust stroke the rings are driven to the bottom of the groove and the oil is driven around and back to the top of the ring; so, when the ring reaches the top and is ready to start on the intake stroke, the oil is there ready to be sucked to the head of the piston by the partial vacuum created, only to be driven into the combustion-chamber on the compression stroke; and this constitutes an oil-pumper. The positions of the rings on the down-stroke and the up-stroke, respectively, should be noted. All shaded spaces shown on the diagram in Fig. 1 indicate the reciprocating space and the space back of the rings in the grooves, exaggerated, of course.

A piston-ring should be made of soft gray-iron containing graphitic carbon and be designed so that it will fill the groove permanently, without reciprocation or sticking in the groove. It should maintain a light and uniform pressure against the cylinder wall, thereby giving a minimum amount of frictional heat and, consequently causing a minimum amount of cylinder wear. It is an undisputed fact that a one-piece ring cannot be given, permanently, the inherent wall-pressure that is necessary; therefore, since it is apparent that all of the qualifications demanded cannot be embodied in a one-piece ring, it is imperative that the multiple-piece ring be resorted to if efficiency is to be obtained for any considerable time. The contact section must be expanded against the cylinder wall to obtain wall pressure and it must be held firmly against one or both of the groove lands, because it must not reciprocate in the groove or we will not have improved over the one-piece ring. Further, this expanding and wedging must be done permanently, through ever-varying temperatures. The problems thus are multiplied, but we know that one or more companies in this Country have solved these problems. We should cease building engines that last only through the guarantee period, add somewhat to their production cost and turn out engines that will operate efficiently for at least long enough to cause the owner to feel that the money he must spend in upkeep is justified by the good service he receives.

Another very important function of the piston-ring is the part it plays in heat radiation. It is known that approximately 90 per cent of the heat that leaves the piston is conducted by the rings into the cylinder wall; therefore, is it not reasonable to say that the deeper the rings are, the wider the contact will be; and that the less reciprocation there is, the greater and more even will be the flow of heat? Hence, deep grooves are essential. Let me say to the advocate of the narrow-faced ring that my personal experience proves that excessive wear on cylinders has resulted whenever and wherever it has been tried. For instance, compare a ring with a 1/8-in. face that has a 10-lb. per sq. in. pressure against the cylinder wall with a 1/4-in. ring having the same cylinder-wall pressure. There will be no difference so far as frictional heat is concerned, but the same pressure will be distributed over a wider area of wall surface by the larger ring and the cylinder wear will be correspondingly less. Consequently, too much importance cannot be attached to the proper width and depth of the grooves.

I have been specializing in stopping gas seepage and oil-pumping in engines for more than 3 years, and can say that I have kept a lathe and one man fairly busy in truing-up worn grooves, deepening and widening them so that they would permit the use of an efficient ring. Many pistons have been brought to me during this period and, owing to the lack of stock back of the rings and in the groove lands, it has been futile even to try to cure their defects; but, in cases where groove depth and groove width have been ample or where widening and deepening have been possible, my success has been uniform, although there have been instances where new and stronger valve-springs have had to be used.

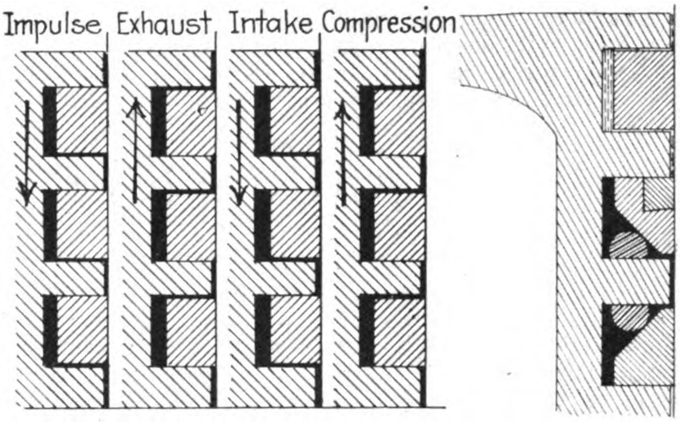


FIG. 1—AT THE LEFT IS A DIAGRAM SHOWING WHAT HAPPENS AS THE PISTON TRAVELS THROUGH THE FOUR STROKES OF THE ENGINE CYCLE AND AT THE RIGHT IS ANOTHER GIVING A COMPARISON OF A ONE-PIECE PLAIN, A THREE-PART PRESSURE-PROOF AND A TWO-PART PRESSURE-PROOF RING

There are many piston-ring manufacturers and while it is true that most of them use a plain one-piece ring owing to its simplicity, ease of installation and low cost, it is just as true that the man who buys a car equipped with rings of this type and is troubled with gas seepage and over-oiling often replaces them with some multiple-piece rings that he believes will improve the operation of his car. To do this he often is obliged to widen and deepen the grooves at his own expense or, possibly to purchase a new set of pistons. Therefore, knowing that this will occur, why not give to the purchaser of a car a plain ring designed to give the best service for the longest period of time, and provide a groove depth and width that will enable him ultimately to employ a multiple-piece ring without unnecessary expense, renewing the life of the engine and eliminating the nuisance of fouled spark-plugs, smoke and the quick deterioration of the lubricating oil by gas seepage?

An efficient multiple-piece ring cannot be forced into the space provided for a shallow, narrow ring. In designing, one should always remember that the piston-rings do more work and withstand more strain and wear than any other part of the engine, that a plain one-piece ring is sure to be the first part of the engine to need replacing and that it is futile to replace a plain ring in a worn groove with another plain ring. If, on account of manufacturing costs, one cannot use a long-lived piston-ring, one at least can make provision for using rings for possibly a year after the car has been sold and the guarantee period has passed; one should not multiply the problems of the purchaser of piston-rings by making it impracticable and unnecessarily expensive for him to renew the power and efficiency of his engine. I have prepared Table 1 with this in view. It is a schedule of widths and depths of grooves that will be ample for all.

TABLE 1—WIDTH AND DEPTH SCHEDULE FOR PISTON-RING GROOVES

Diameter of Piston, in.	Groove Width, in.	Groove Depth, in.
3, and under	3/16	5/32
Over 3, to 3½, inclusive	1/4	11/64
Over 3½, to 4, inclusive	1/4	3/16
Over 4, to 5, inclusive	5/16	1/4
Over 5, to 6, inclusive	5/8	5/16

The diagram at the right of Fig. 1 shows a piston equipped with one plain piston-ring in the top groove, a three-part pressure-proof ring in the middle groove and a two-part pressure-proof junior ring in the lower groove. It illustrates the non-reciprocating features of both multiple-piece rings. The shaded portion shows the space back of the two lower rings.

[The discussion of this paper is printed on p. 228.]

CHRONICLE AND COMMENT

(*Continued from p.* 224)

will be written by men engaged in the study of motor-car production for the larger Detroit passenger-car builders. The Studebaker, Packard, Wills and Ford companies have assured the Society of their cooperation in the meeting. Other companies are favorably considering representation on the program. Special factory inspection trips will be conducted by production executives each afternoon to supplement the papers. The Production Dinner on Thursday evening, Oct. 26, will be the first gathering of the kind supported by production executives and factory men whose genius has given the United States its world-dominant position in the economical production of motor vehicles. The Production Meeting is intended to serve the producing arm of the industry as a forum for collective reasoning and the free interchange of experience on manufacturing methods and processes. If it matches the results in the engineering field as represented by the S.A.E. Standards and Fuel Research work, then the meeting will be a credit to the Society.

The Section Meetings

SECTIONS meetings are an important phase of the Society's work. The wide geographical distribution of our membership prevents a majority of the members from enjoying the privilege and benefits of attendance at the national meetings. The multiplicity and diversification of automotive engineering problems make it increasingly difficult to cover the art adequately in the limited time available at the national meetings. Only problems that are urgent and of greatest general interest can be treated. Section meetings fill the gap. They provide the advantages of group study and discussion for many members who could not enjoy them otherwise. Papers in specialized fields find their proper place on the Sections' programs. Detroit Section meetings are logical forums for passenger-car discussion, Dayton for airplane subjects, Minneapolis for power-farming and so on.

Sections meetings are by no means of lesser rank, value or interest than national meetings. The attendance is often as large as at the national technical sessions. Creditable Section papers have equal status with those presented at national meetings and receive the same consideration in THE JOURNAL. Discussion of them is often more productive of valuable information because of the presence of a large group of specialists and the longer discussion period that is generally available.

Attend the meetings of your Section. Maintain a more intimate relationship with the Society through your local organization and you are certain to place a higher value on your membership. Read the announcements on the Sections page of THE JOURNAL each month. You will find them on p. 292 of this issue.

Simplified Practice

TO Secretary Hoover, of the Department of Commerce, lessening of waste, estimated at from 25 to 40 per cent of our total industrial effort, is essential to the maintenance of our rightful position in foreign trade. Simplification, that is procedure as to sizes, dimensions and materials that does not involve unnecessary elaboration, complication or addition, is a chief means to this end. Standardization, in the sense of normal uniform sizes or amounts, or a definite degree of any quality, viewed as a prescribed object of endeavor or a measure of what is adequate for some purpose, is of course closely allied to such procedure.

In this connection, the Division of Simplified Practice of the Department of Commerce, of which W. A. Durgin is the chief, offers clearing-house facilities and the moral support of the Government. Mr. Durgin points to the fact that there has been too little fundamental investigation and economic development of basic lines; it has seemed much simpler to start new varieties of endless products. He urges that the best feasible practices be made effective so far as possible, industrial representatives working out in conferences held under Government auspices recommendations embodying the most promising solutions of vexing problems, and the Government in turn accepting these recommendations as its own. Mr. Durgin says: "The craftsman can create just as beautiful a Chippendale if he adheres to a length of 6 ft. 3 in. as if impulsively he adopts 5 ft. 11 in." In addition, that all industries had better guard against over-diversification. The only reason for dimensional standardization of assembly units is to secure interchangeability; and this should be secured except in rare cases where proper freedom in design work would be curtailed.

Committees appointed by the Society and by the National Automobile Chamber of Commerce, to work with the Department of Commerce in the matter of the approval of practices and of the consequent distribution of Government reports of the kind indicated in the automotive field, have recognized the need for greater simplification in connection with storage-batteries, ball bearings, spark-plugs and tires. Additional formal sessions of these committees will be held in Detroit next month.

At the last meeting of the Society, R. M. Hudson, of the Division of Simplified Practice, emphasized the importance of standards developed through many years of practical experience by men who have learned how to reduce the best of technical practice into sound commercial products.

The Department of Commerce, in contact with scores of industries through their trade organizations, has completed some "simplifications"; each industry having been asked what work of the kind it considered would be of greatest advantage to it and consumers. The department seeks to bring together the commercial and the technical groups interested with the purpose of securing unanimous recommendations. It is felt that through wide distribution of pamphlets incorporating the recommendations, and following-up with regard to their reduction to commercial practice, more support of the best industrial thought can be had.

The Department of Commerce is expecting the Society to extend its efforts beyond the technical and commercial phases into that of economics.

The Winslow Automotive Boiler

By CHARLES B. PAGE[1]

MINNEAPOLIS SECTION PAPER *Illustrated with* DRAWINGS AND PHOTOGRAPHS

WITHOUT dwelling upon the properties of high-pressure steam and the economies resulting from its use, the paper describes and illustrates the construction of the Winslow boiler. Efficiency curves are presented and data are given in support of the claim that, per ton-mile or other work done, steam power and gas power are about at a stand-off in weight of fuel consumed.

It is asserted that the preponderance of opinion is to the effect that any wide automotive application of steam power must come through the use of the water-tube type of boiler. The history of the Winslow boiler is outlined and its advantages are enumerated, illustrations and tabular data being presented in substantiation of the claims made regarding its satisfactory performance.

THERE are many evidences of a widespread and increasing interest in the application of steam power to automobiles, trucks, tractors, unit railroad cars and the like, especially where the duty is heavy and where breakdowns result in losses all out of proportion to the cost of repairs. This interest is not new. It is basic, indicating an instinctive feeling on the part of engineers, producers and users that, if rightly applied, steam will compete more and more actively with the internal-combustion engine in the automotive industry.

I believe it would be perfectly safe for any impartial investigator to state that the great buying public is sold on steam; that capital is friendly; that the manufacturer is open-minded; and that, given a suitable boiler, the engineer is ready to back steam power with all his reputation. Aside from the psychology of the thing, which may be due to the familiarity of several generations with steam power, to its almost exclusive use aside from water-power as a prime-mover, and to that feeling of power and reliability as we sense it in the movement of passengers and freight over our railroad system, there is the matter of lower operating cost, greater simplicity and lower maintenance cost as compared with gas power. And beyond that is the very pressing issue of the conservation of volatile fuels.

It is not purposed in this paper to dwell upon the properties of high-pressure steam and the economies from the use of it. This matter has been covered admirably by contributions to the technical press and the engineering societies and is pretty generally understood by engineers everywhere. Suffice it to say in review and based on theoretical considerations, the Rankine theory, that the efficiency increases rapidly with a rise in pressure. Fig. 1 shows the efficiency curve from 100 to 600 lb. per sq. in. absolute pressure. In practice, there are several losses that cut down these theoretical efficiencies, including especially the loss due to initial condensation. However, this loss can be and is substantially eliminated by adding a moderate amount of superheat to the steam. Authenticated laboratory performances of high-pressure steam-engines show a steam consumption ranging down to 12 lb. of steam per b. hp-hr. Allowing 10 per cent for driving auxiliaries, such as pumps, fans and the like, the

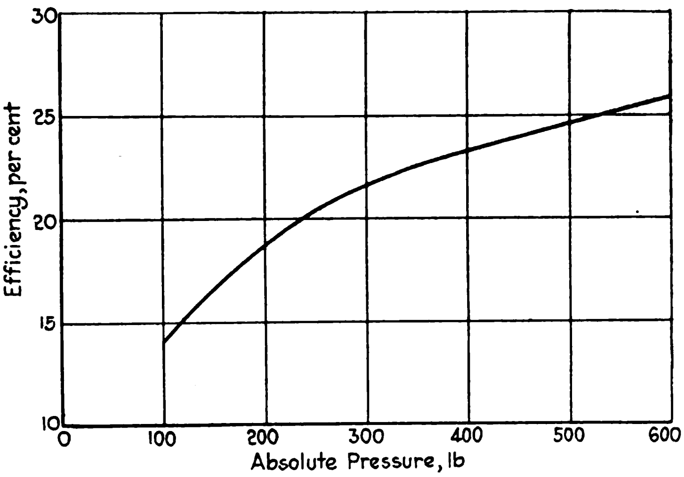


FIG. 1—CURVE SHOWING HOW THE EFFICIENCY OF A STEAM POWER-PLANT RISES AS THE PRESSURE IS INCREASED

actual steam consumption per effective horsepower delivered to the propeller-shaft would equal 13.2 lb. If we assume 80-per cent boiler efficiency and a heat value of the fuel of 19,900 B.t.u., the equivalent evaporation is 16.4 lb. of water per lb. of fuel. This figure, translated into terms of steam at a boiler pressure of 400 lb. per sq. in., and a superheat of 100 deg. fahr. evaporated from a feed-water temperature of 120 deg. fahr., equals 13.01 lb. of water per lb. of fuel. From these figures we arrive at an effective brake-horsepower per 1.01 lb. of fuel and an overall thermal efficiency of 12.6 per cent. Average field performance should be very close to these figures. The steam consumption stated is by the use of compound engines. The simple locomotive type of engine will use from 40 to 60 per cent more steam. The field performance of a gas-powered vehicle will average about the same fuel-consumption per effective horsepower; that is, 1 lb. per hp-hr. A check on this is found in the mileage record of a popular steam car weighing 3800 lb., general reports indicating from 10 to 14 miles per gal. of kerosene, in spite of the fact that the engine is of the simple locomotive type and therefore uneconomical as compared with the compound type of engine. Nevertheless, this mileage is comparable to that of gas cars of similar weight and power.

Therefore, it can be claimed safely that, per ton-mile or other work done, steam power and gas power are about at a stand-off in the weight of fuel consumed. Lower fuel-cost comes through the use of the heavier kinds of oil, any oil which will flow freely through a pipe, ranging, according to present-day prices, from 15 down to 6 cents per gal., depending upon the gravity and source of supply, compared with gasoline ranging from 20 cents upward. As to simplicity, one authority claims that a steam plant has about one-half the number of moving parts required in a gasoline-engine power-unit of the same horsepower. The number of working parts is a consideration, but to the layman the most important matter is the ease with which powerplant troubles can be

[1] M.S.A.E.—President, Page, Beck & White, Inc., Chicago.

diagnosed. In the main, one's troubles with a steam plant are due to leaks that make themselves evident immediately. It may be a leak in a stuffing-box or a union, something that can be fixed with a wrench at the journey's end. The peculiarities of both powers must be understood to assure commercial operation, but after 20 years of experience with gas power as engineer, salesman and manufacturer, and a much shorter experience with steam, I am inclined to the opinion that the relative simplicity of the two powers is about proportionate to the ratio of the number of their moving parts.

In the matter of long life and maintenance cost, there is this in the main to be said: Steam power employs an expansive medium applied without shock or jar, whereas the power from an internal-combustion engine is in harnessing a series of explosions. Broadly speaking, the steam-powered vehicle will run until it wears out, while the gas-powered vehicle fails through a breakdown, due to crystallization resulting from the explosions from which the power is derived. For proof, recall some of the old-timers on rail and water, in the factory and out in the woods, still wheezing away, not quite so efficient, perhaps, but still on the job, or, more to the point, note the year and the model of some of the ancient White and Stanley cars still in service.

What of the conservation of volatile fuels? Unless the rapidly increasing consumption of gasoline can be checked, the automotive industry will reach the point of saturation very shortly, simply because of the mounting price of that fuel. Save gasoline for those vehicles for which gas power is best adapted. Use the heavier liquid fuels for the heavier types of automobile, truck, tractor, motorbus, unit railroad-car and industrial locomotive. Make a wider use of waterpower for industrial and railroad purposes and conserve coal for cooking and heating.

This idea of the superiority of steam for automotive work persists in the mind of the public in spite of many failures during the last 10 years. Even capital does not seem to be discouraged, since today there is more expenditure for and more activity in steam power than ever before. Some of this work is being undertaken in a highly intelligent manner. Some of it is being done without any understanding of previous work and the present state of the art. Much of it is without any comprehension of the basic principles of steam engineering. The field is not closed to the inventor, whether or not he be technical, but the really constructive work must come from the trained engineer, as in every other line of engineering endeavor. All honor to the Stanley Company for having kept the steam idea alive during the last decade. Its success has been due to the fact that in an engineering sense it has kept its feet on the ground. More particularly, we might say that its success has been due to the Stanley boiler. It supplies steam in relatively large quantity and of good quality. It is constant in action because it has a stable water-level. In design it conforms to basic boiler principles and the laws of physics governing evaporation, circulation and separation. Of the automotive failures of the past 10 years, by far the greater majority are chargeable to the boiler used, and in every such case it will be found that the failure was due to departure from the laws mentioned.

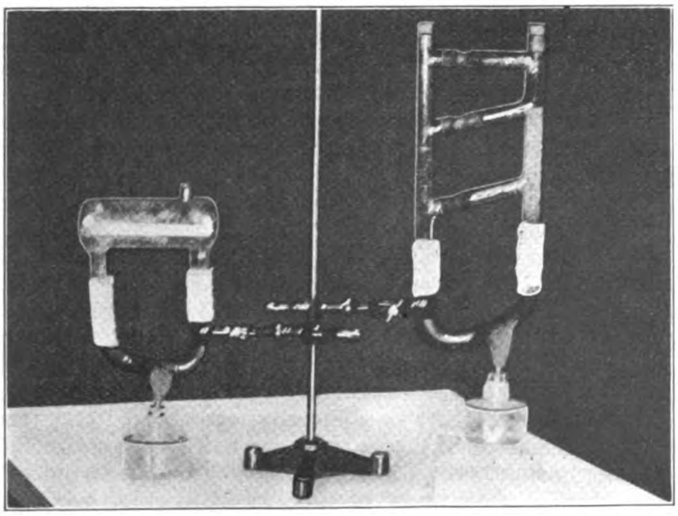


FIG. 2—LABORATORY EXPERIMENT SHOWING THE THEORY ON WHICH THE OPERATION OF THE WINSLOW BOILER IS BASED

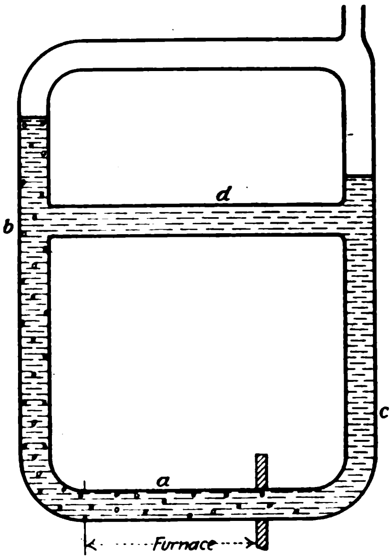


FIG. 3—THE THERMAL ACTION THAT TAKES PLACE IN THE EXPERIMENT ILLUSTRATED IN FIG. 2

POSSIBLE BOILER TYPES

There are three general types of boiler possibly adaptable to automotive work; fire-tube, water-tube and flash. I will not discuss the comparative merits of these. Capable and conscientious engineers are working in these several divisions. May they all be successful. Competition is the life of trade and we need it to put steam where it ought to be in the automotive industry. Nevertheless, as one gets around the Country and talks with engineers interested and versed in the subject, one is led by the large preponderance of opinion to conclude that any wide application of steam power must come through the use of the water-tube type of boiler.

For the better understanding of the problems confronting the steam automotive engineer, it is necessary to appreciate the limitations governing the design of the boiler. It must be efficient, quick-steaming, have an abundance of reserve power, be safe and of ample capacity to supply enough steam to do the work required. Its product, steam, must be uniform in pressure and temperature. It must be independent of the use of distilled water and be long-lived. It must be all these things within a space and weight almost unbelievable when one thinks in terms of stationary, marine and locomotive boilers. Not the least of the foregoing is the matter of water supply and incrustation from scale, mud or carbon, the last-named being a possible consequence of condensing the exhaust steam containing the cylinder oil from the engine and the delivery of a portion of this oil

into the boiler with the feed water. With reference to mud, the comment by the executive of a prominent tractor company is significant. He writes:

> I have seen a service-man take 21 wheelbarrow loads of sediment out of an old-type steam traction-engine after 30 days of use. Now suppose your boiler was run 100 days on that same kind of water.
>
> (Signed) President

Another remarks:

> One point worth considering, I think, with special reference to this section of the Country, is that much of the water found in many localities is not of the best, to say the least, for steam purposes.
>
> (Signed) Manager Tractor Department

Quick steaming and reserve power are diametrically opposed. The engineer must make the best compromise possible. Safety and restrictions as to weight and space spell the elimination of all drums. For capacity, an astonishing amount of heating surface must be packed under the hood and into a very few cubic feet of space. An unstable pressure would mean unsteady power or work done. An ungovernable temperature would destroy the cylinder lubrication. A stationary marine or locomotive engineer has no such problems to solve as has he who would design a boiler for automotive work and, while the latter's work must be governed by basic laws and principles, there is little in the former's experience to show him the way.

History of Development

In 1911 W. H. Winslow, of Chicago, set himself the task of developing a commercial automotive boiler. After ownership of several steam cars, he was no novice

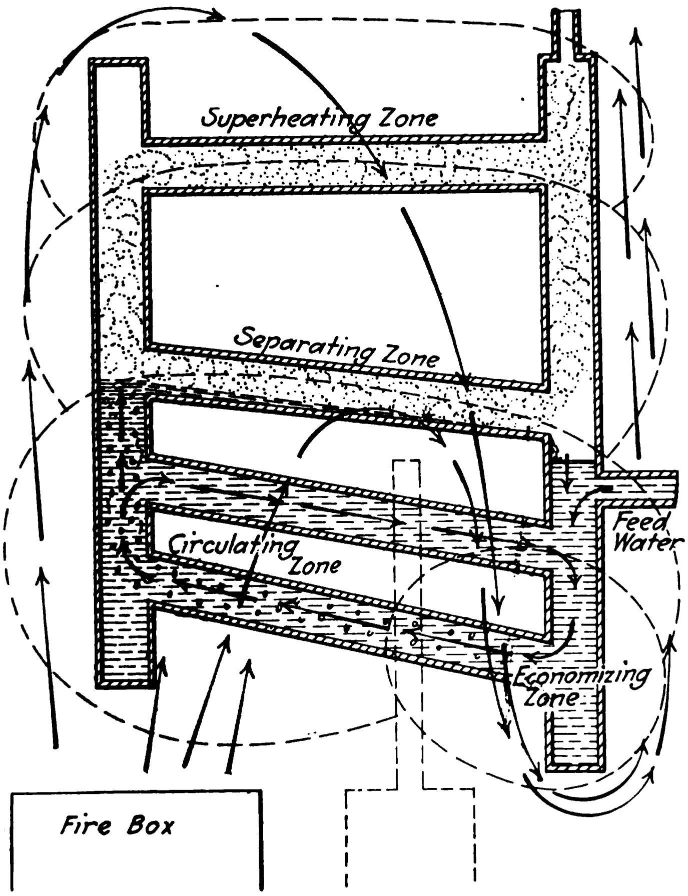


Fig. 4—Sketch Showing the Application of This Principle to a Boiler

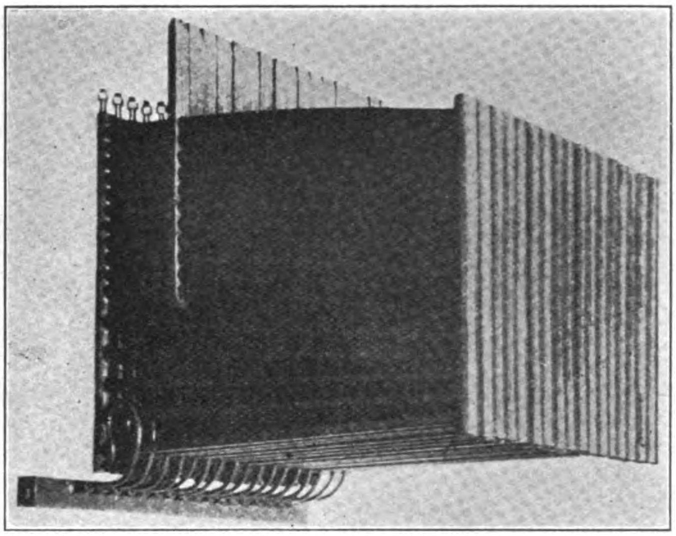


Fig. 5—The Original Winslow Automotive Boiler

in the state of the art and the history of its development. Viewing the requirements of a successful automotive boiler, the water-tube type was selected as the best means to the end sought. He approached this subject from the standpoint of the scientific investigator, having ample funds at his command and being supported by a thoroughly competent engineering staff. First came a series of laboratory tests and experiments. Mr. Winslow early perceived the necessity of devising a method of joining or assembling the water-tubes or pipes in such a manner as to obviate the use of all screwed, rolled or expanded joints and fittings in the zone of the hot gases of combustion. How else could the required heating surface be nested in the space available? The mechanical means for accomplishing this had just come to hand, autogenous welding. The outcome was the unitary system made up of tubes welded into headers and presenting the largest amount of heating surface that could be combined into drumless form.

Coincident with this development, in order to guarantee safety and further economies in weight and space, Mr. Winslow conceived the idea of generating steam in a drumless form of boiler made up of unitary sections, each section a complete boiler in itself in which the thermal action of generating dry saturated steam, involving evaporation, circulation and separation, can be carried on regularly and continuously. One of the laboratory experiments used in developing this idea is illustrated in Fig. 2. Heat applied as shown sets up a rapid ebullition and circulation. Separation was observed in the upper cross-tubes. Fig. 3 records the thermal action that took place. The ebullition in *a* causes a higher water-level in *b* than in *c* and therefore a return flow through *d*, causing a rapid circulation through the circuit. In drafted form, this system is illustrated by Fig. 4.

The first experimental boiler was a small one of some five or six sections on which a number of block-tests were made to prove the principle of operation. The tests with the six sections were so satisfactory that a complete boiler was built and installed in a Stanley limousine, displacing its so-called 23-in. fire-tube boiler. This car ran continuously $2\frac{1}{2}$ to 3 years, averaging 1200 to 1500 miles per month. Mr. Winslow personally ran the boiler dry 27 times with the fire full-on. When the boiler was red-hot, he at once used the hand-pump to charge the boiler with water while the fire was still burning. Some of the

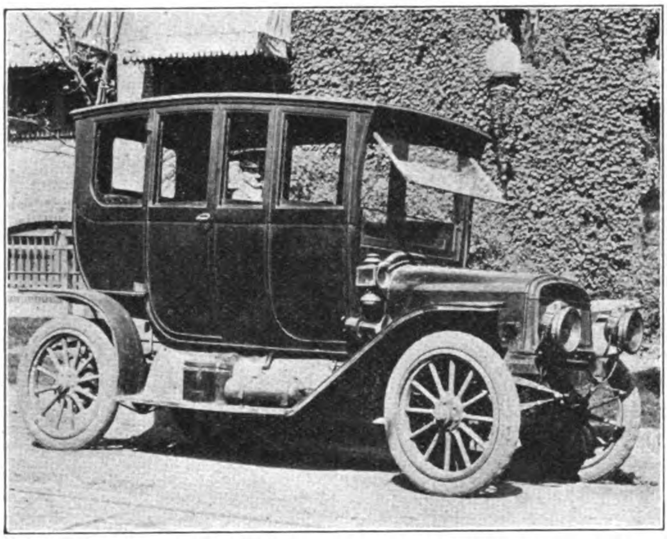


Fig. 6—The Car in Which the First Winslow Boiler Was Installed

tubes warped a trifle but otherwise there was no damage. From the chauffeur's record, the car was driven 46,000 miles. The boiler was then removed and sawed into sections for examination, from which it appeared that it should have been good for 100,000 miles additional.

The boiler and the car referred to are shown in Figs. 5 and 6. Meanwhile, other boilers of various sizes and types were built and tested in the laboratory and on the road, until in 1913 Mr. Winslow and his associated engineers turned their attention to the possibilities of this type of boiler for stationary and marine work. A 75-hp. stationary boiler was built as shown in Fig. 7, followed by a 500-hp. boiler of the same general type. These boilers were used for experimental and commercial purposes, yielding a vast amount of very valuable engineering data applicable in part to the automotive type and demonstrating the practicability and economy of high-pressure steam-boilers for central-station work and on shipboard.

In 1919 a 5-ton truck, illustrated in Fig. 8, was constructed. This truck was put into commercial operation and gave an exceedingly satisfactory account of itself except for some engine trouble. Also in 1919 the automotive boiler illustrated in Fig. 9 was built. This boiler has been subjected to a series of exhaustive laboratory tests. Tests Nos. 10, 22 and 24, being averages of the series, are quoted in condensed form in Table 1. The boiler, economizer, superheater, burner and case occupy a space 24 in. wide, 37 in. high and 31 in. long. The total weight is 961 lb. and 100 lb. is deducted as being the weight of the burner. The heating surface of the horizontal water-tubes only is 89.6 sq. ft., and the water capacity is 13 gal.

These tests were especially important as establishing the capacity, evaporation and efficiency of the boiler when used in combination with a blast type of burner. While there has been some variation in detail during this 10-year period of development, the fundamental ideas advanced by Mr. Winslow in the early days have proved to be sound in every respect. The unitary welded section is substantial and yet possesses sufficient flexibility to take care of any strains due to unequal expansion. The sections combine advantageously into any capacity and dimensions. Thermal action proceeds uniformly in every section and in perfect balance as between the sections. All sections deliver steam to a common equalizing header and receive water from an equalizing feed-pipe or mud-drum. The desired water-level is maintained readily and the product of each section is dry saturated steam at a uniform pressure and temperature. Superheating can be accomplished in the section itself by providing additional tubes above the separating zone but, on account of the limited height available for automotive work, is preferably added outside of the unitary section in the form of coils looped between the sections. An examination of Fig. 10 will help to fix in one's mind the thermal action described.

TABLE 1—RECORD OF TESTS OF WINSLOW BOILER, COMMERCIAL LABORATORY AUTOMOTIVE TYPE, KEROSENE FUEL, BLAST BURNER

Test No.	10	22	24
Date of Test,	Nov. 19, 1919	Nov. 25, 1919	Nov. 25, 1919
Duration of Test, min.	60	60	40
Pressure, lb. per sq. in.	599.2	587.0	578.0
Saturated Temperature, deg. fahr.	489.0	487.0	485.0
Superheated Temperature, deg. fahr.	530.0	566.0	543.0
Feed-Water Temperature, deg. fahr.	59.0	57.0	57.5
Actual Evaporation, lb. per hr.	946.5	876.0	558.0
Equivalent Evaporation, lb. per hr.	1,212.0	1,130.0	708.0
Fuel Burned, lb. per hr.	66.75	62.25	41.25
Actual Evaporation, lb. of water per lb. of fuel	14.10	14.10	13.50
Factor of Evaporation	1.26	1.29	1.27
Equivalent Evaporation, lb. of water per lb. of fuel	17.7	18.1	17.1
Boiler Efficiency, per cent	86.0	88.0	83.0
Boiler Horsepower	35.0	32.0	20.5

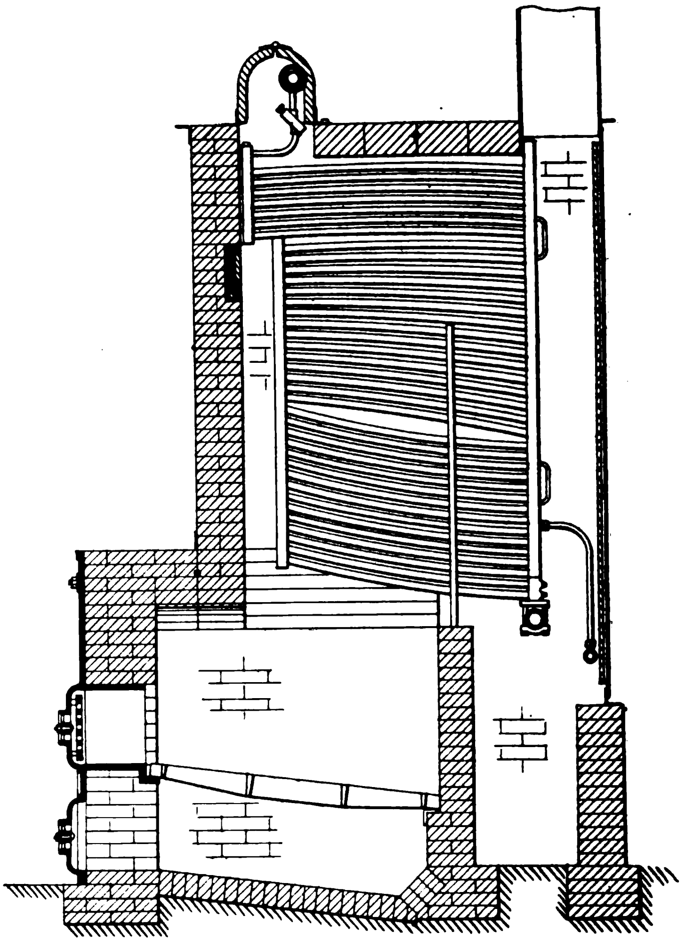


Fig. 7—Cross-Section of a 75-Hp. Stationary Boiler Embodying the Same Principles of Construction

The larger or cold-end header is located well outside of the furnace, while the smaller or hot-end header is well inside the path of the ascending gases of combustion. Therefore, when heat is applied to the boiler, its first action is to expand the water contained in that part of the section in front of the cold-end header. Expansion and ebullition cause the water to flow rapidly toward and upward in the hot-end header. The difference in water-levels thus created causes a gravitational return-flow to the cold-end header. The hotter the fire is, the more return-tubes are brought into action; therefore, circulation proceeds in perfect accord under all working conditions. The water in the tubes and headers and the hot furnace gases that surround them move at high velocities, thus increasing the heat transference greatly. The formation of steam bubbles further tends to raise the water-level in the front-end header, increasing the gravitational difference between the two headers, tending to increase the circulation further. Mixed steam and water return to the cold-end header by gravitation through the tubes in the zone of separation. A large water-surface is thereby developed from which the process of separation is accomplished in an ideal manner without the use of drums. The result is dry saturated steam, produced in a combination of tubes and headers of relatively small diameter devoid of all bolted, threaded, riveted, expanded or beaded joints in the fire area. The regular and convenient control of the feed-water pumps is possible because of the presence of a constant water-level. This stability contributes substantially to the maintenance of constant pressures and temperatures.

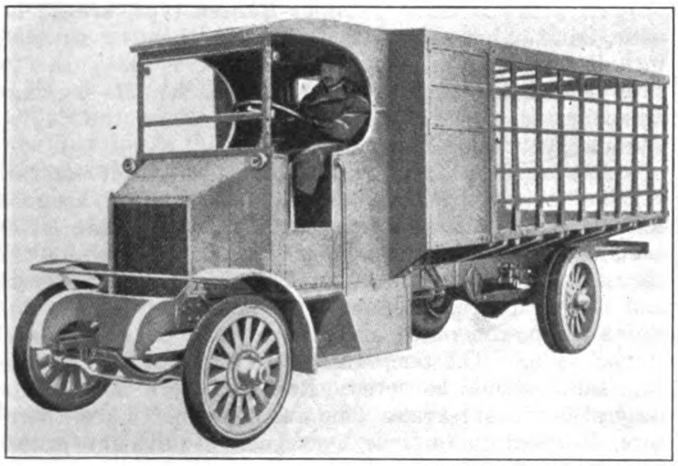


Fig. 8—A 5-Ton Truck Built in 1919 That Used the Winslow Boiler

The circulation within the horizontal tubes and hot-end header is so rapid as to prevent the deposit of scale. Further reference to Fig. 10 shows that a continuation of the downward path of the water in circulation in the cold-end header leads directly into the mud-drum. Particles that are heavier than water tend to settle out directly into this drum, whence they can be blown out at stated or convenient intervals. The use of soft water free from mud is desirable; nevertheless, due to the excellent provisions for the elimination of water impurities, the practical use of the Winslow boiler is not limited anywhere because of the water available. There is a gain also from the oil pumped into the boiler with the feed water. All automotive steam-plants are now condensing. The cylinder oil passes through the condenser with the exhaust steam into the hot-well or water-tank. Most of this oil floats to the top and is flushed off with each filling of the tank. What mixes with the feed-water and goes into the boiler serves the admirable purpose of preventing corrosion and the adhesion of scale to the tubes and, with the impurities in the water, forms a sludge that separates out into the mud-drum as described. This is not to say that, with the use of bad water, cleaning in the course of time may not be desirable. When required, a section can be dismounted readily and laid on a convenient block to tap the tubes and headers lightly with a copper hammer or wooden mallet. This action loosens any deposits, which can then be washed or shaken out, or a scale-dissolving compound pumped into the boiler and allowed to stand for a day or two, when the boiler can be washed clear.

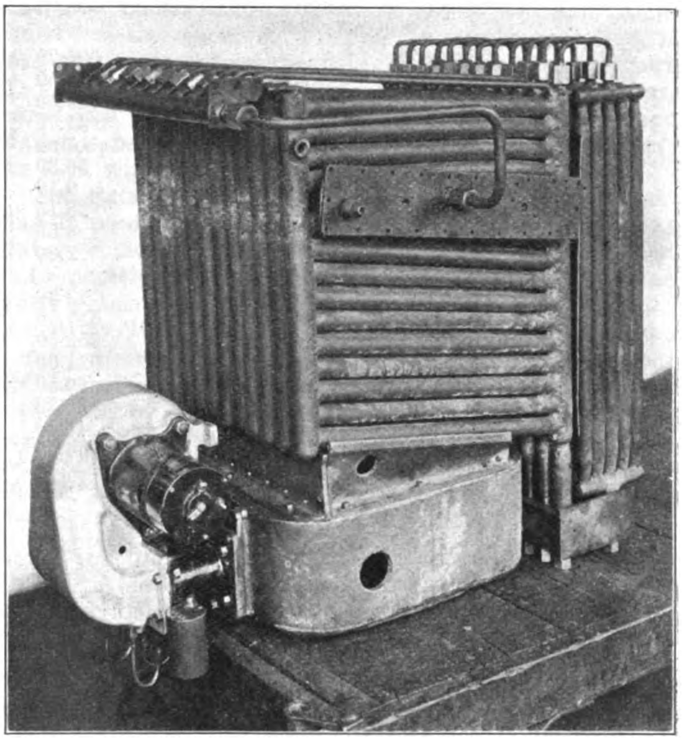


Fig. 9—The Present Form of Winslow Automotive Boiler

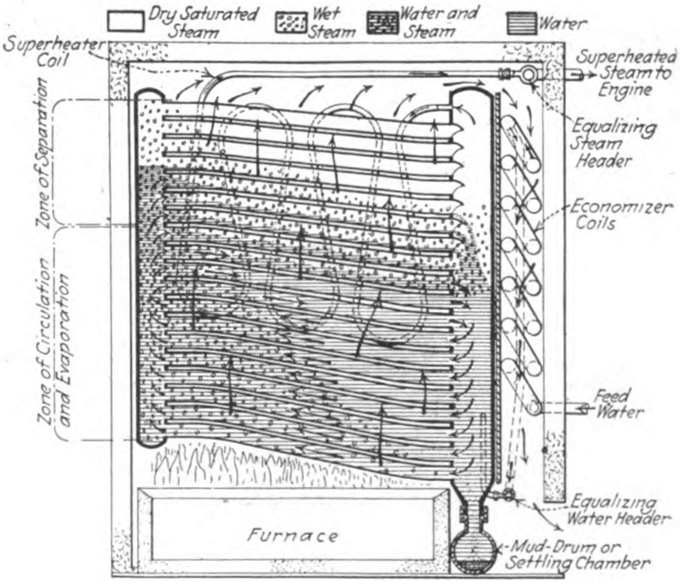


Fig. 10—Cross-Section of the Winslow Boiler as Now Constructed

TABLE 2—SUMMARY OF ARMOUR INSTITUTE OF TECHNOLOGY TESTS

Water Used, Total, lb.	695.20
Water Used, Total, gal.	83.50
Water Used, per Hr., lb.	347.60
Water Used, per Hr., gal.	41.75
Kind of Fuel Used	Gasoline
Fuel Used, Total, lb.	56.30
Fuel Used, Total, gal.	9.10
Fuel Used, per Hr., lb.	28.15
Fuel Used, per Hr., gal.	4.55
Duration of Test, hr.	2.00
Equivalent Evaporation, lb. of water per lb. of fuel	12.36
Average Steam-Pressure, lb. per sq. in.	517.00
Temperature of Steam, deg. fahr.	652.50
Temperature of Flue Gas, deg. fahr.	621.00
Temperature of Feed-Water, deg. fahr.	60.50
Barometric Pressure in inches of Mercury	29.22
Temperature of Room, deg. fahr.	46.30
Specific Gravity of Fuel, deg. Baumé	56.25

MAJOR ADVANTAGES

Given an otherwise perfect steam-generator for automotive use, failure would nevertheless be the final verdict unless the boiler were substantially self-cleaning. Experience with Winslow boilers built and used during the last 10 years proves conclusively that they are self-cleaning; that the oil contained in the feed-water and the rapidity of circulation prevent scaling in the tubes; and that the impurities in the water or sludge separate out and down into the mud-drum. Any section can be isolated in the field by uncoupling two unions, inserting blank flanges and recoupling; or a section can be withdrawn completely and sent to the nearest garage for repair by welding, blanks in the meantime being inserted in the unions. The work in hand can then be completed without material delay, although with a reduction in overload capacity; or, very properly, a spare section can be carried in stock so that replacement can be made immediately. Each boiler-section is made of the highest grade of seamless drawn-steel tubing welded into headers of the same material. Joints so made are stronger than the walls of the tubes themselves. There are no riveted laps or other joints whatever and no screw fittings subject to high temperature. Sections are tested to 4000 lb. per sq. in. Working pressures up to 1500 lb. per sq. in. are feasible. Even if a tube should rupture under pressure, the action would be equivalent to the popping of a safety-valve and involve no greater hazard. An explosion is out of the range of possibilities. The boiler can be run dry, heated to a bright red and cold water pumped in while in that heated condition, without damage. The boilers are not only safe but indestructible.

For 90 days, recently, I have been driving a test car

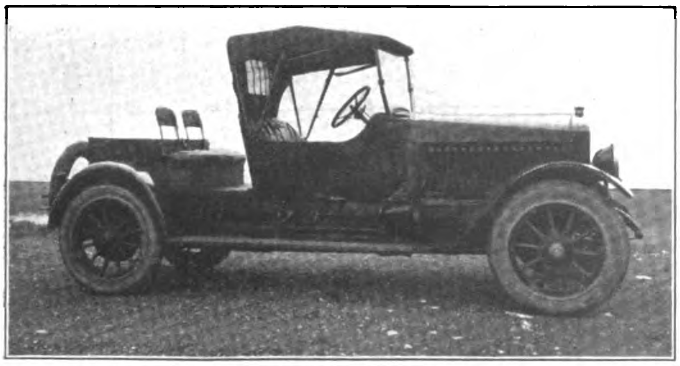


FIG. 11—A CAR EQUIPPED WITH THE LATEST DESIGN OF WINSLOW BOILER

such as is shown in Fig. 11. It has a Stanley chassis fitted with the latest model of Winslow boiler. The total distance traveled to date is something over 2200 miles, mostly 'cross-country, using water picked up from any convenient source. The performance of the boiler has been 100 per cent throughout. With sudden opening of the throttle, there is no priming or carrying over of slugs of water into the steam pipe. There have been no deposits of scale, mud or carbon, at least so far as might be indicated by some evaporation tests just completed at the Armour Institute of Technology. There is reserve capacity in abundance. Steam can be raised from all-cold in 7 min. and from a normal condition, that is, pilot-light left burning overnight, in 3 min. The boiler has been run dry purposely with the burner full-on on eight separate occasions, the lower tubes each time being heated to a bright red. Under this condition water was pumped into the boiler, resulting in a flash generation of steam, but absolutely without damage to the boiler. Table 2 is a summary of tests made Nov. 26, 1921, at the Armour Institute of Technology. In these tests the gasoline pilot-light was always burning; it consumed 0.37 lb. of gasoline per hr., which is not figured into any of the calculated values. Attention is called by G. F. Gebhardt to the fact that the fluctuations in steam pressure, temperature and quality and water-level may cause a considerable error in the results, and that the average observed and calculated values should be accepted only with this in mind.

The general dimensions of this boiler, including the burner, which is of the Stanley-Bunsen type, are 22 in. wide, 34 in. high and 34 in. long. The heating surface, including horizontal water-tubes only, is 93½ sq. ft. The temperature of saturated steam at the average boiler-pressure is 473 deg. fahr.; therefore, the degree of superheat was 179 deg.

Previous tests had indicated that the burner was set a trifle too close to the boiler tubes to afford as good combustion with kerosene as with gasoline. The latter fuel, therefore, was used in this test. For road work kerosene is used exclusively. Fluctuations in pressure and temperature were due to the "off" and "on" of the water-pump, controlled automatically by the water-level in the boiler. The temperature of the feed water, 60½ deg. fahr., should be noted. Readings were taken regularly at 5-min. intervals. The averages quoted are, therefore, believed to be true averages. Equivalent evaporation corrected to include fuel burned in the pilot-light is 16.32 lb. Figuring the heat value of the gasoline at 20,500 B.t.u., the boiler efficiency is 78.5 per cent. Note that the temperatures of the flue gases, 621 deg. fahr., are given. If this boiler were fitted with an economizer, which might be expected to cut down this temperature to about 450 deg. fahr., the overall efficiency might be raised to approximately 82.5 per cent, as high a percentage as could be conceded in view of the best performance of stationary water-tube boilers.

TABLE 3—FLUE-GAS ANALYSES

Test No.	1	2	3
CO_2, per cent	11.0	11.3	12.0
O_2, per cent	4.7	4.2	3.6
CO, per cent	0.0	0.3	0.4

Flue-gas analyses of a previous test run under the same conditions, except with a water-level 2 in. lower, are given in Table 3. The type of burner used accounts for the lower capacity as compared with that shown by the tests

quoted in the earlier parts of this paper. Otherwise the results are in line and each checks with the other.

Burners of both the vaporizing and the atomizing type are available. Pumps and other accessories such as fans are standardized as to form, dimensions and manner of use. Reliable and economical types of steam engine have proved themselves through years of service. Therefore, the work of the steam automotive engineer may henceforth be said to be that of combining and adapting demonstrated units into the required powerplant. Thus may automotive steam-power become the vogue, at least for heavy-duty work.

THE DISCUSSION

CHAIRMAN W. G. CLARK:—Regarding the thermal action of the boiler, which way are the sections placed in an automotive vehicle, fore-and-aft or across?

CHARLES B. PAGE:—They may be of the fore-and-aft or crosswise type. In the locomotive illustrated the tubes are placed crosswise of the vehicle, there being two sets, right and left-hand. As a one-piece boiler, the gases of combustion pass straight up through the tubes and out of the stack. The smaller or hot-end headers are inside of the cold-end headers. The former practically insulate the cold-end headers, thereby tending to maintain circulation under all conditions and without regard to the slope of the tubes due to hill-climbing.

CHAIRMAN CLARK:—The question was prompted by the fact that the circulation of the water in the boiler, as I understand it, is due to the temperature difference between the hot and the cold-end headers. If you were going downhill, the slope of the tubes would be neutralized.

MR. PAGE:—Although circulation is facilitated by a moderate inclination given to the tubes, it is fundamentally due to the difference in the head and the density of the water in the two headers. Due to this difference, which is increased considerably with increased evaporation, circulation will be maintained from the cold-end header through the tubes to the hot-end header even though the tubes are brought into a position almost reversed from normal in hill-climbing.

CHAIRMAN CLARK:—How do you prevent priming when the throttle is opened suddenly?

MR. PAGE:—Due to the construction, we do not seem to have priming. Many a time, driving down Michigan Avenue, in Chicago, I have found myself held up at a street crossing, in front of two or three gasoline-driven cars. Under such circumstances one cannot resist the temptation to show what steam can do in the way of quick acceleration. With a long cut-off and a well-opened throttle, I have been almost halfway down the block before the gasoline-driven cars get across the cross street; all this without boiler priming. Dry steam under all service conditions is a fundamental characteristic of this boiler.

CHAIRMAN CLARK:—Are the upper tubes of greater cubic capacity than the lower?

MR. PAGE:—No, the upper tubes tend to smaller diameters than the lower tubes, but they are still of equivalent cross-section; that is, they give sufficient free surface of water for the separation of the steam.

E. R. GREER:—Let me suggest an imaginary case of a farm tractor equipped with such a boiler. Take a case where the tractor is to operate 30 min. on a hill sloping in one direction. Whether the boiler is set fore-and-aft or crosswise, the tubes are exactly in the opposite position to the slant desired and there is a continuous hard pulling condition. Even if going downhill, the pull of the plow means that most of the power will still be required of the machine. Further, in winter operation, if the tractor is out in the field, perhaps a mile or more from water, and it must be left standing over Sunday, what can be done? Must one drain the water out and carry it back there on Monday? Water cannot be left in it in winter.

MR. PAGE:—If there is a restriction as to width making it necessary to place the sections fore-and-aft, the tubes would be inclined sufficiently that on the steepest hills practicable for tractor plowing there would be no interference with the circulation. Under the conditions stated by Mr. Greer, the water would have to be drawn out of the water-tank, to say the least, and probably out of the boiler, although it would be impossible to rupture either the tubes or the headers if the water should freeze in the boiler.

A. R. SANDT:—In the car that you operate, what is the capacity of the water-tank and how long does the water last?

MR. PAGE:—The water-tank capacity is 25 gal. Stanley salesmen and agents claim an average of 100 miles to a tank of water. They do better at times and much worse on occasions. We have had to refill the water-tank after 25 miles and we have gone as high as 12 miles to the gal. or at the rate of 300 miles per tank of 25 gal. The amount of water used depends upon the temperature of the air, the speed of the car and the character of the road. For instance, if the car is on a concrete road and traveling at from 30 to 35 m.p.h., the condensation is complete. If the car is pulling through deep sand, the loss of water is very considerable. The Stanley car is not fitted with a fan. If a fan were installed behind the condenser, the water loss would be very much less because the fan would tend to compensate for the reduced speed where the going was difficult. With an improved design we should get the water-consumption down to an amount not in excess of the fuel burned.

W. H. PETERSON:—I presume it would be impossible to condense all of the steam with an air-cooled condenser so as to make the steam plant fully-condensing.

MR. PAGE:—It could be done, but it would mean a large condenser and would hardly be worthwhile unless the car were going into a locality where water is very expensive. In the road locomotive we figure the condenser capacity at normal temperature as somewhat in excess of the normal rating of the boiler. There would be some loss in steam when working at an overload, but that loss would depend entirely on the temperature and the amount of the overload.

MR. PETERSON:—Are you not troubled with oil?

MR. PAGE:—No. We want some oil in the boiler because it smears the interior of the tubes with a thin coating and, in addition, forms a sludge with the impurities in the feed water that otherwise would form into scale. The sludge settles into the mud-drum whence it is blown off at regular intervals. The boiler that Mr. Winslow operated for 46,000 miles and which he estimated could have been run for 100,000 miles in addition, was not equipped with a condenser and, therefore, he did not have the benefit of such oil as we now pump back into the boiler.

MR. PETERSON:—Did he not contemplate running with a condenser at one time to get the benefit of a vacuum?

MR. PAGE:—In the early days I think that was not considered, but today no one would think of building a steam-powered vehicle without putting on a condenser.

MR. PETERSON:—You have said very little about the engine.

MR. PAGE:—There are a number of very well-developed engines and it is simply a matter for the engineer to pick out the type that he thinks will best suit the job. Beyond a doubt, the best practice in England and in this Country is in favor of the compound type of engine. The uniflow type has been highly developed for full-condensing installations; but it has hardly been commercialized, at least in this Country, for automotive applications.

A. H. LOOMIS:—The company that I represent has been building track-laying-type log-pullers for a number of years. There seems to be a demand for a machine like the one we have been building that would not require so much water. This is why I look on this boiler with favor; the strong point is the saving of water.

A. W. SCARRATT:—Concerning the application of the steam boiler to tractors, Mr. Page stated with reference to the car he is driving that the water loss is excessive when heavy going is encountered. I presume one would not expect more than 100 miles to 25 gal. of water under good average conditions, but the conditions encountered with automobiles are mild when compared with those met by tractors in the field. Vast areas in this and other countries require plowing in very hot and dry seasons and many sections are usually in need of water at that time. Water is at a premium in some localities. Making, we will say, 100 miles in an automobile would be the equivalent of about a 3-hr. usage of a tractor employing a boiler and engine of the same capacity. At that rate, a 10-hr. day would require the refilling of the tank approximately three times, or about 3 bbl. of water per day, a physical impossibility in some of those districts.

MR. LOOMIS:—I should explain that we have never been interested in the agricultural field. Our interest is in the log-hauling field where the going is up and down grades and there are off-and-on conditions similar to those of an automobile.

MR. PAGE:—Regardless of the application of steam power, be it to automobile, truck or tractor, the problem resolves itself into determining the proper size of the condenser and fan and means for driving the fan to get the desired results. To meet the working conditions described by Mr. Scarratt, I see no reason why it would not be perfectly feasible to install a condenser and a fan of such capacity as to keep down the water loss to the equivalent of the amount of fuel burned; that is, if the tractor were burning 1 bbl. of fuel oil per day, the water loss would not exceed 1 bbl. per day.

A. R. SANDT:—What would be the size of the condenser compared with that of the boiler in a tractor layout of that kind?

MR. PAGE:—That would depend upon the size of the tractor. We have made some layouts that meet Mr. Scarratt's specifications. The proportions as to dimensions and weight are attractive and practical.

MR. SANDT:—What is the comparative cost of production of steam-driven and gasoline-driven tractors?

MR. PAGE:—The production cost would be high at first. One correspondent said that he believed that when volume came up to a reasonable figure production costs would be less than for the gasoline-powered vehicle. Some companies are now endeavoring to bring out a $1,000 steam car. I do not know that they will succeed in producing a marketable car to sell at that price. We have made some studies of production costs of boilers for cars of that size and type and believe that by proper manufacturing methods the cost can be cut to a competitive figure.

MR. PETERSON:—It has seemed to me for a long time that there are great possibilities in developing steam plants for tractors and automobiles. The question of condensation depends largely upon the point to which the steam is expanded in the engine; if it is exhausted at a very high pressure, the temperature is high and the condenser would have to be of disproportionate size. I feel that the engine has not yet been developed which will work to the best advantage on tractors and automobiles. On some steam turbines now in operation, the steam is expanded down to a temperature of about 80 deg. fahr. In the Minneapolis flour-mills one can hold one's hand on a low-pressure turbine and it will seem to be cold. There is a great opportunity to develop something of this kind that will be of great value in the automotive industry.

On the subject of economy, I feel that Mr. Page is very conservative. In Europe, where fuel has been high for many years, they had developed long before the war steam engines that showed greater economy than this plant of which he speaks. There are powerplants of the Locomobile type that will develop 1 b.hp-hr. from 1 to 1¼ lb. of comparatively low-grade coal; that is, coal having a thermal value of perhaps 12,000 or 13,000 B.t.u. Mr. Page has mentioned fuel that runs from 19,000 to 20,000 B.t.u.

MR. GREER:—Do you have a pump in the feed-water system?

MR. PAGE:—Yes.

MR. GREER:—It seems to me that all such extras introduce many complexities.

MR. PAGE:—The feed-water pump is an extremely simple mechanism. There is a plunger working in a cylinder and a stuffing-box to keep the water from leaking out; and there are two check-valves, one suction and one discharge. If the pump fails to work, it is easy to locate the trouble. The rest of the steam plant is simple. There are, I believe, more Stanley steamers in the Chicago district than in any other section of the Country. I have found, to my surprise, that the number of physicians there owning steam cars is greatly out of proportion to the other professions represented. The average physician is a poor mechanic. The fact that so many physicians own Stanley cars is a tribute to the simplicity of the steam powerplant.

E. B. MCCARTNEY:—I understand that the burners of a certain steam-driven car cause much trouble by clogging up.

MR. PAGE:—Certain things have to be done to get satisfactory service from the steam powerplant. One of these is proper and periodical cleaning of the burner and pilot light.

CHAIRMAN CLARK:—Is not the trouble with burners due to the kind of fuel used?

MR. PAGE:—With the vaporizing type such as the Stanley car uses, the fuel burned is kerosene. The blast type of burner must be used to burn the heavier grades of fuel. This type is in service commercially all over the world. Referring again to the practical every-day utility of steam-powered vehicles, speaking broadly, I repeat that commercial operation depends upon regular inspection and service by the owner. This is not laborious, technical or expensive; it is just routine. It will assure satisfactory performance.

MR. SCARRATT:—If the thermal efficiency of the complete powerplant using a high-grade engine and this boiler is 12.6 per cent, is the average steam-consumption per brake-horsepower delivered in the neighborhood of 15 lb. per hr.?

(*Concluded on p.* 274)

Piston-Rings

By JOHN MAGEE[1] DETROIT SECTION PAPER

THE author believes the piston-ring problem to be an engineering one worthy of serious study and that it should be possible to standardize types and sizes in a way that will go far toward eliminating present difficulties.

It is stated that cast iron is the only satisfactory metal suitable for use in the internal-combustion engine and that the foundry offers the greatest opportunity for improvement, in the elimination of poor castings. The superiority of individually cast rings is averred and a formula for their composition is given.

Leakage and oil-pumping are discussed, followed by comment upon the width and form most desirable for piston-rings; and scme of the difficulties of their manufacture are enumerated, together with suggested improvements, inclusive of inspection and testing methods.

SINCE many piston-rings have failed in the performance of their function either through faulty engineering or careless manufacture, piston-rings have developed into a problem. It seems that the problem is logically an engineering one, that piston-rings should offer a real opportunity for study and that it should be possible to standardize types and sizes in a way that will go far toward eliminating present difficulties. With this in mind, any discussion covering practices of today should assist in formulating a consensus of opinion as to the best practice.

While bronze, Swedish iron and even malleable iron and steel have been tried, it will be conceded that, so far, cast iron is the only satisfactory metal suitable for piston-ring usage in the internal-combustion engine. The density, the resiliency and the small cross-sectional area each being an important factor, it is evident at once that the foundry offers the greatest opportunity for improvement toward piston-ring perfection. Manifestly, with poor castings at the start, very little better than poor results can be expected at the finish. There is little question as to the superiority of the individually cast over the pot-cast piston-ring. Table 1 gives a mixture formula for individually cast rings.

Extreme care in the selection of materials, combined with frequent physical tests, will be necessary to maintain the standard. A required property of a test-bar ½ in. square is a Shore hardness of 35 to 40 or a Brinell hardness of 200 to 230.

TABLE 1—FORMULA FOR INDIVIDUALLY CAST PISTON-RINGS

Substance	Per Cent
Silicon	2.50 to 3.00
Sulphur, maximum	.... .. 0.70
Phosphorus	0.30 to 0.50
Manganese	0.45 to 0.70
Combined Carbon	0.50 to 0.60
Graphitic Carbon	2.75 to 2.65

LEAKAGE AND OIL-PUMPING

Many of the large number of piston-ring designs include some special joint. Because the periphery of a piston-ring is broken only at the joint, the impression seems to have prevailed that most leaks could be located there. Some of the devices for effectually sealing the joint are real tributes to inventive genius. As a matter of fact, the joint occupies such a relatively small percentage of the circumference that its effect on the whole is rather insignificant. In other words, it would be possible to construct a perfectly sealed ring-joint and yet have about 98 per cent of the remainder of the circumference leaking gas and causing loss of compression-pressure.

R. E. Lawrence, professor of mechanical engineering at the University of Detroit, recently calculated the leakage using a formula for the flow of gas through an orifice. The dimensions taken were those of the Ford-engine cylinder, with 0.002-in. clearance between the piston and the cylinder wall. Both tangs of a step-cut piston-ring were removed, and 3/16 in. on the circumference of the ring was open. If the piston were then held still during a period equal to the duration of a stroke at a 15-m.p.h. road speed, a pressure of 450 lb. per sq. in. would allow 0.0006 cu. ft. of gas to pass through the opening. This represents, by this calculation, the maximum leakage through a joint under the most "favorable" conditions. Such conditions never actually exist in the cylinder. The explosion rarely generates a pressure of 450 lb. per sq. in. and in any event the pressure always drops very quickly during the power stroke. Then, if there be three rings instead of one, with a film of lubricating oil helping to impede the progress of the escaping pressure, it will be seen readily that the actual amount of gas which could pass through the joint of a piston-ring under ordinary conditions would be expressed in a fraction much smaller than 0.0006 cu. ft.

In regard to so-called oil-pumping, it seems advisable to keep the ring periphery as nearly continuous as possible. The step-cut provides a joint that accomplishes this well and makes it the most desirable for all purposes.

WIDTH AND FORM

The proper width of piston-rings is another subject productive of much discussion. The reduction in weight of all reciprocating parts is certainly to be desired. When it is taken into consideration that any added weight on a piston-ring must be multiplied by a factor of 6 to determine its equivalent inertia effect for each cylinder, the advisability of narrow widths is evident immediately. Theoretically a knife-edge in contact with the cylinder wall will produce the proper result. It remains only to establish the added width necessary to allow for practical production, always considering that the minimum width is desired. Present methods of manufacture indicate that ⅛ in. is the proper width, all things considered. The thickness of the ring or the depth of the groove is subject to the same consideration in all ways.

The merits of both eccentric and concentric forms have been discussed from time to time. No doubt, the eccentric ring is more correct for theoretical uniform wall-pressure. However, if the pattern for the casting is designed for the ring at its full opening, and the natural surface density of the inside of the ring is left undisturbed in machining, a proper foundry-mixture will

[1] Detroit Piston Ring Co., Detroit.

produce a concentric ring with a wall pressure that is so nearly uniform in actual operation that its many other advantages make it preferable. It should be remembered also that the theoretical eccentric ring tapers down from its heaviest section, opposite the joint, to a knife-edge at the joint. Any design for an eccentric ring must modify this form to a certain extent, to avoid the thin wall at the joint.

Manufacturing Difficulties

It is not my intention to enlarge upon the proper methods of manufacture of piston-rings. It must be admitted, however, that more failures in piston-rings are due to faulty manufacturing than to faulty engineering principles. After all, from an engineering standpoint, the function of a piston-ring is a simple one. Unfortunately, the manufacturing problem is not so simple as is supposed sometimes. Some study of the manufacture of piston-rings will facilitate developing specifications that are likely to be met.

All piston-ring manufacturers prefer to make rings with the finished surfaces ground to size. It is easier to hold accurate dimensions on a grinding machine than it is on a lathe. However, hard castings and hard spots in castings are machined without difficulty or detection by grinding. For this reason piston-rings with a *turned* finish on the diameter dimension should be specified. Neither the scleroscope nor the Brinell-hardness test will disclose hard spots in castings. A ground finish may cover up many hard spots or a hard scale. The production of a turned surface is a somewhat slower process for the manufacturer, but it guarantees a uniform soft wearing surface.

Flatness is very essential, since a serpentine condition will allow leakage around the back of the ring through the ring groove. If all of the internal stresses are not removed during the process of machining, the ring is as apt to warp sidewise as it is to become elliptical. Therefore, width dimensions should be inspected with a light gage instead of with a micrometer. With a ring lying on a perfectly flat surface, a side-warp will cause it to register oversize. With measurements at intervals, the two points of a micrometer might indicate a parallel width on a ring considerably warped.

The most common "defect" in the manufacture of piston-rings is the elliptical ring, generally termed out-of-round. The specified out-of-round tolerance may vary according to the amount that the ring is likely to be worn-in on the block. At best it is desirable to keep the variation within very low limits, say 0.00025 to 0.00050 in. If the tolerance is expressed in light-gage terms, it is much more simple for inspection, because the light-gage is at present the most practical inspection device for locating out-of-round rings.

No accurate data are available for determining the poundage or wall pressure of piston-rings to accomplish definite purposes. Actual experiments, conducted at different times for different purposes, seem to indicate that a poundage in excess of 4 lb. per sq. in. of bearing surface is needed to prevent collapse under pressure. Therefore, a poundage of 5 lb. per sq. in. has commonly been specified for all purposes.

There seems to be a diversity of opinion as to the proper gap-opening or expansion allowance. This probably is due to the difference in the estimated temperatures to which a piston-ring is subjected. Using 0.0000056 per in. per deg. as the coefficient of cast-iron expansion, a minimum opening-allowance for maximum expansion is obtained by multiplying this coefficient by the circumference of the ring.

The foregoing comments are made with a view to the standardization of the best engineering and manufacturing practice, rather than to attempt to dictate the ultimate design for piston-rings. The ever-increasing number of sizes and designs, some of which are freakish, is certainly an indication of the dire need for such a standard.

[The discussion of this paper is printed on p. 228.]

WINSLOW AUTOMOTIVE BOILER

(*Concluded from p.* 272)

Mr. Page:—It is calculated at 13.2 lb. per hr. at the actual boiler-pressure and temperature of the steam, including the auxiliaries.

Mr. Scarratt:—That steam is exhausted from the engine at a very high temperature. From a tractor standpoint it seems that a large condenser would be necessary to handle the volume of steam and convert it back into feed-water at something like 120 deg. fahr., and that a considerable portion of the available energy of the powerplant would be used for driving the fans that produce the air-blasts for condensing purposes. At present about 5 to 7 per cent of the total energy developed is put into fan energy simply to reduce the water-temperature on an average approximately 12 to 15 deg. fahr. That is the difference in the temperature of the water between the time it leaves the cylinder-head of the engine and enters the water-jacket, and represents in heat units approximately 32 per cent of the total thermal value of the fuel.

Mr. Page:—The type of fan used on a gasoline engine is not efficient for handling large volumes of air at a low pressure. That is one item of difference. Of course, the amount of horsepower used to drive a fan on a steam powerplant is greater than on a gasoline powerplant of the same horsepower but, whatever the horsepower requirements may be, they are figured against the gross horsepower developed.

J. L. Mowry:—I believe we will come back to steam application on the tractor. However, I am not entirely convinced that our present steam engines will fill the bill. I have thought for a long time that the so-called automobile type of high-pressure boiler would do the work. With reference to the points that have been made in regard to the angle of the tubes in ascending and descending grades with a tractor, is there a limit to the pitch that can be given them without obstructing the steam separation?

Mr. Mowry:—Can that be done without getting too much pitch for a heavy pull uphill? More power is needed then than when going down.

Mr. Page:—There one needs the overload capacity. A heavy tractor would probably call for all the overload capacity of the boiler, as well as that of the engine.

Mr. Mowry:—But if one does not get steam separation, the boiler will prime.

Mr. Page:—The construction of the section is such that separation takes place as the mixture of steam and water flows from the front to the rear header, even if the tubes are unusually inclined.

The Gasoline-Driven Motor-Coach for Railroad Service

By Charles O. Guernsey[1]

Indiana Section Paper *Illustrated with* Photographs

EFFORTS to operate railroad rolling stock by gasoline engines have met with little success in comparison with the results achieved in other lines. To bring out the reasons and to show what the field is for this class of equipment is the purpose of the present article. The various attempts to adapt gasoline engines to railroad work are reviewed from the days of the Strang and McKeen cars. To demonstrate that the real field of the gasoline railcar lies in the middle-ground between the rail bus and the steam train, the author enumerates the advantages of both kinds of service and shows that, to develop a satisfactory car, each side of the controversy must make concessions to the other. The early difficulties having been largely due to the use of too heavy cars, the weight could be decreased materially by the application of approved features of automotive practice. Other requirements should include four-wheel pivoted trucks, front and rear, full speed in either direction, air-brakes and safety appliances. The motor-coach should combine the light weight of the motor truck with the safety, steadiness, comfort and convenience of the steam coach.

SINCE the days of the first automobile, various attempts have been made to design railroad equipment operated by gasoline engines. It seems rather remarkable that so little progress has been made, when one considers the place that the gasoline engine has in the marine, aeronautic, automotive, stationary and, in fact, practically every power field except the railroad. I am of the opinion that there are definite reasons why progress has been slow up to the present time. It is the purpose of this article to bring out these reasons and to show what the present field is for this class of equipment. The motor-coach will fill a decided need but it must not be considered as a cure-all. It has limitations.

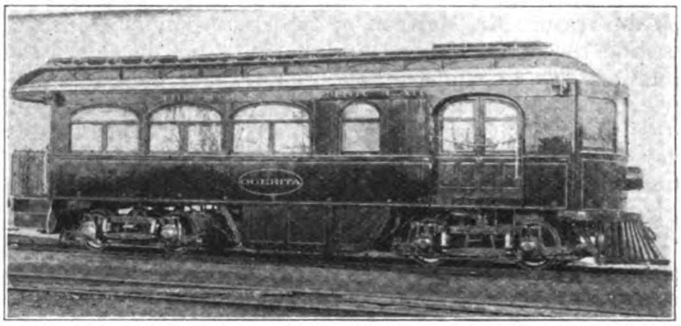


Fig. 1—An Early Motor Rail-Coach, the Strang Gas-Electric Car That Was Built in 1905

One of the early attempts was that of Strang, which occurred about 1905; it is illustrated in Fig. 1. Previous to this, various efforts had been made to provide a single-unit car with steam. These cars were unsuccessful primarily on account of boiler limitations, the light fire-engine type of boiler being too expensive to maintain and the locomotive type too heavy. They had no particu-

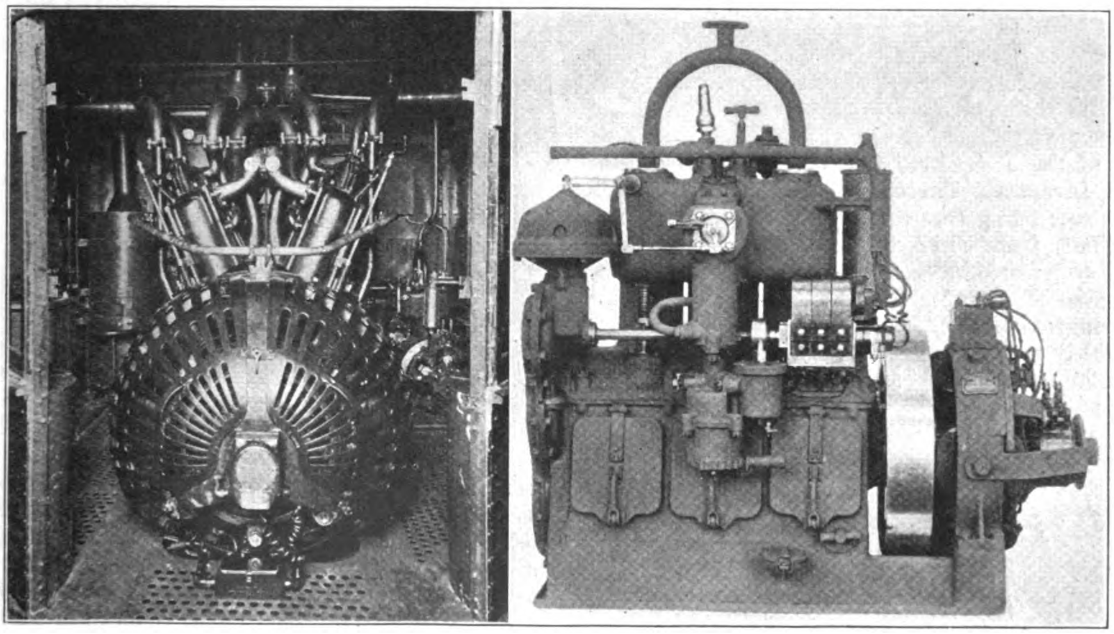


Fig. 2—End and Side Views of the Gasoline-Engine Generating Set Developed by the General Electric Co. in 1910

[1] M.S.A.E.—Manager, railroad division, Service Motor Truck Co., Wabash, Ind.

lar advantage over the ordinary type of steam locomotive since they did not eliminate roundhouse supervision and the like. About 1910 the General Electric Co. became interested in the problem and spent considerable time and effort in developing cars to be propelled by electric motors, the current for which was to be supplied by a 175-hp. gasoline-engine, connected to an electric generator as shown in Fig. 2. Notwithstanding that a number of these cars, such as the one shown in Fig. 3, were built, and some of them, in fact, are still in service. Generally speaking, they were not successful because the great weight, complication and maintenance expense made their operating cost almost as high as that of a steam train. Mr. McKeen, of the Union Pacific Railroad, seeing the need of something of this kind, developed the McKeen car, of which probably more have gone into service than any other up to the present time. Here again, the great weight necessitated an engine considerably larger than was commercially practicable, so that the total cost of operating the car was only slightly less than that of a steam train. Another reason why these early cars were not popular was that the builders failed to give due consideration to the economics of the situation. They attempted to have the same capacity and speed in a gasoline car as in a steam train consisting of a locomotive and two cars. Wherever such capacity is required, the steam train can ordinarily be operated at a profit. The real field for the gasoline car is in service where the steam train has more capacity than is required.

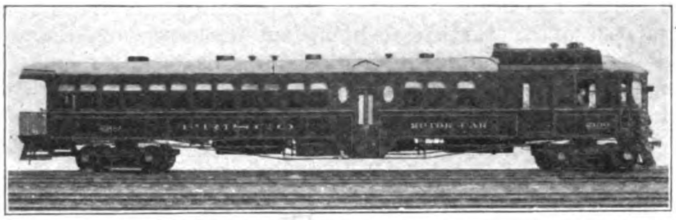


Fig. 3—The General Electric Motor Rail-Coach

The heavy gasoline-cars referred to above ran too near to the steam train in capacity, speed and operating cost. They did not fit the field.

About 15 years ago the car shown in Fig. 4 was built at the plant of the J. G. Brill Co., Philadelphia, on contract for the inventor. The engine was mounted in the center of the car, being connected by a silent-chain drive to a countershaft from which the drive was through propeller-shafts to two bevel-geared axles, one on either truck, as shown in Fig. 5. Other cars have been built similarly, including those of Hall-Scott and Sargent. Cars that have failed did so because the designers failed to see the peculiar field of the gasoline engine.

It is generally conceded that for continuous heavy-duty work, engines having cylinders with a bore larger than

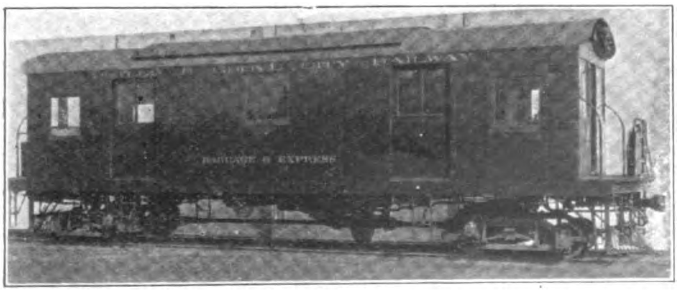


Fig. 4—Another Early Motor Rail-Coach in Which a Silent-Chain Drive and Propeller-Shaft Were Used

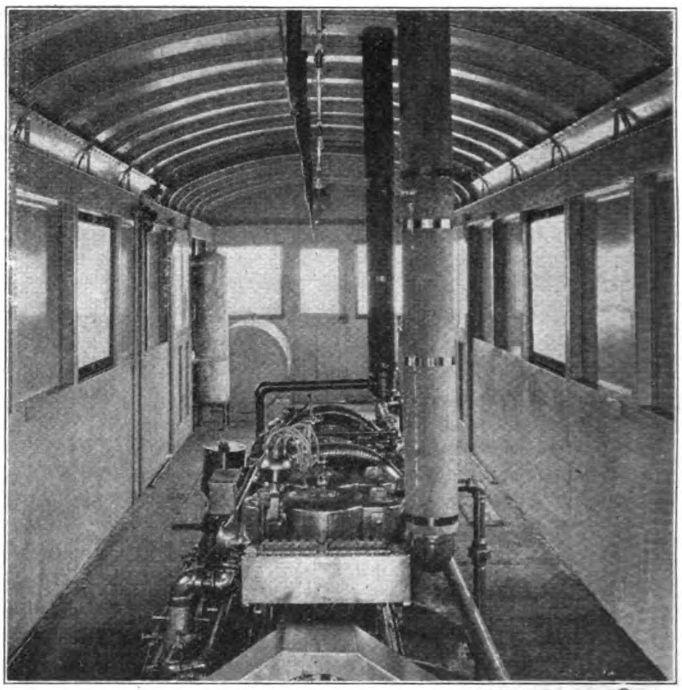


Fig. 5—Powerplant of the Car Illustrated in Fig. 4

5 in. are not commercially successful. The troubles due to warpage, lubrication difficulties, the heating of valves and piston heads and the like become too great to handle. Apparently, the failure of these cars can be traced directly to the failure to appreciate the limitations of the gasoline engine.

On the other hand, builders of motor trucks for several years have equipped chassis ranging from 3/4 to 5-tons capacity with flanged driving-wheels and with other means to adapt them to operation on rails, as shown in Fig. 6. These cars, in general, have been suc-

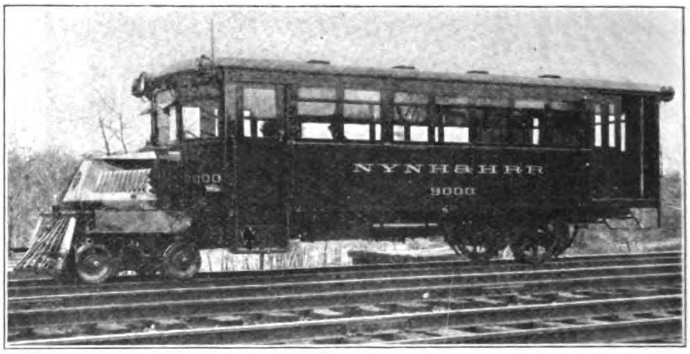


Fig. 6—A Motor-Truck Chassis Equipped with Flanged Driving-Wheels for Operation on Rails

cessful. Due to light weight, low rolling-resistance, the small engines required, and to the fact that in some installations they can be handled by one man, the operating cost has been exceedingly low. These cars, however, being unduly limited in capacity and speed, fill only a limited demand. There still remains a middle-ground between the rail motor-bus or the rail motor-car, as we have known them in the past, and the proper field of the steam train. It is in this middle-ground that the motor-coach can make a place for itself.

A motor-coach is defined as a passenger-carrying gasoline-driven railroad motor-car, designed specifically for operation on rails. It is in no sense a converted motor-

truck, but represents a combination of automotive and railroad practice. Previous attempts have been based almost entirely on one to the exclusion of the other. To illustrate, some of the earlier cars developed by the railroad men weighed more than 50 tons and required 300 hp., although seating capacity was provided for only some 50 to 60 passengers. They did not give due consideration to the automotive side of the design. Instead of building a gasoline car, they attempted to use a gasoline engine in a car of steam-train design.

By careful design and the use of alloy steel, anti-friction bearings and other approved features of automotive practice, the weight can be reduced materially. The weight must be held to a minimum to keep the motor-coach requirements within the capacity of proved gasoline-engines. It is also undesirable to forget the railroad point of view entirely. Many features of railroad design are the result of almost a century of development. The designer must weigh his problem carefully, choosing from railroad practice those features that fit this new type of equipment. The converted motor-truck does not, according to experienced railroad officials, meet the requirements. Due to the use of two-wheel driving-trucks and other practices, it does not ride as steadily

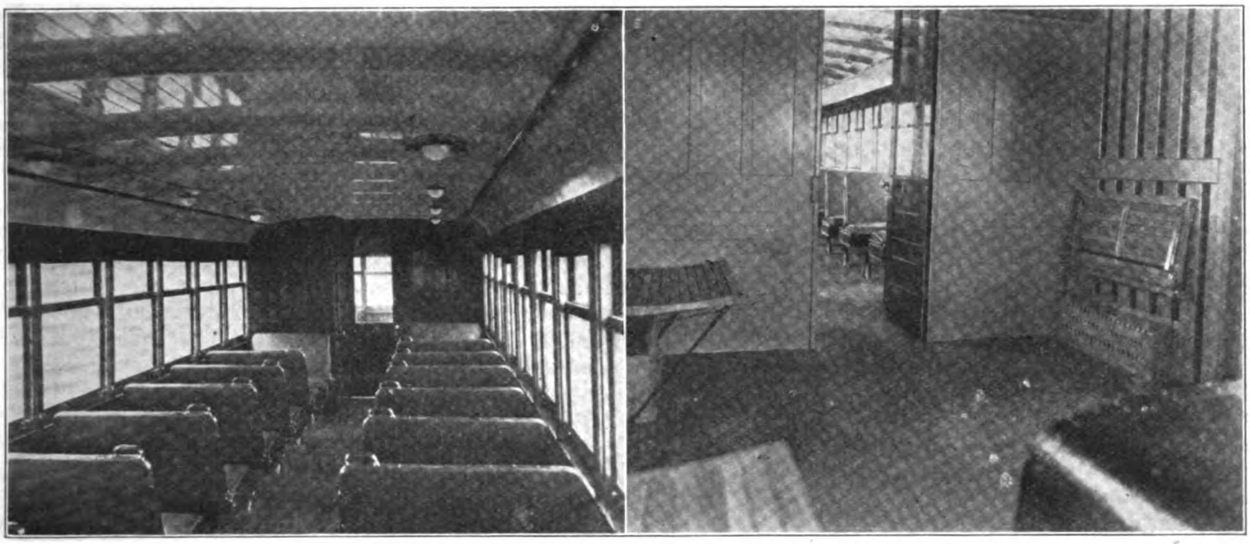


Fig. 8—Views of the Passenger and Baggage Compartments of the Car Shown in Fig. 7

or as safely as the usual railroad car, which has a four-wheel truck under either end of the car.

There is an insistent demand for several features not ordinarily included. Some of the more important of these are four-wheel pivoted trucks, front and rear, full speed in either direction, air-brakes and safety appliances. The motor-coach should combine the light weight of the motor truck with the safety, steadiness, comfort and convenience of the steam coach. Fig. 7 illustrates a motor-coach, having an overall length of about 44 ft. and a seating capacity of 38, in addition to drop-seats for eight passengers in the baggage-room, making a total seating capacity of 46, as shown in Fig. 8. The baggage space is 70 sq. ft. The car is provided with standard vestibule-doors for entrance, a saloon, comfortable seats, electric lights and other features commonly associated with modern railroad design.

The total weight of this car is only 13 tons. This reduction of weight to less than one-third that of old-time motor-cars of the same capacity, makes it possible to use a 68-hp. engine, as against the 200-hp. engine required by other types. At the same time a speed of 48 m.p.h.

Fig. 7—A 44-Ft. Motor-Coach Having a Total Seating Capacity of 46 Passengers

has been attained and a speed of 35 m.p.h. can be maintained indefinitely, without damage to the mechanism. Due to the light weight and the correspondingly small amount of power required, a car of this type will show exceptional economy. The fuel-consumption is light, the car running between 5 and 7 miles per gal. of gasoline. Due, also, to the light weight, the car has very good acceleration, reaching a speed of 25 m.p.h. in 30 sec. from a standing start. This car is arranged with two four-wheel pivoted trucks. The drive is from the unit powerplant, located forward, through an auxiliary transmission, contained in the bolster of the front truck, shown in Fig. 9, to the two axles of the front truck. The

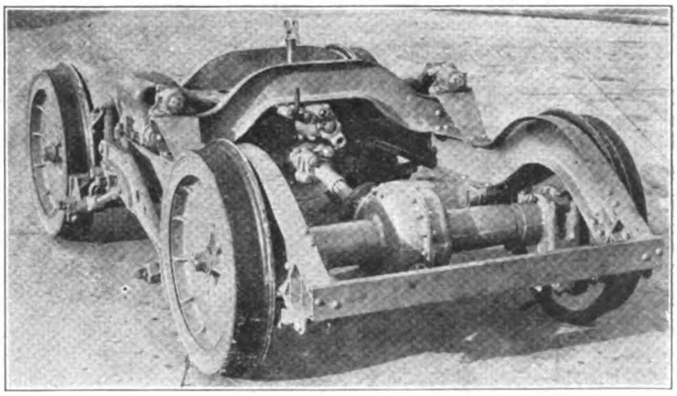


Fig. 9—Front Truck of the Car Illustrated in Fig. 7 in Which Power Is Delivered to Both Axles by an Auxiliary Transmission Contained in the Truck Bolster

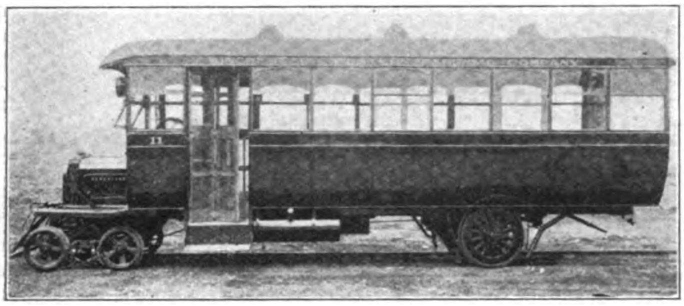


FIG. 10—A 30-PASSENGER GASOLINE MOTOR-COACH THAT IS IN SERVICE ON A BRANCH LINE IN THE BLUE RIDGE MOUNTAINS

auxiliary transmission is arranged so that either of two pairs of gears can be used for transmitting the drive, thus, in effect, giving two high-gears. One of these gears is proportioned for the ruling grade on the particular railroad on which the car is to be used, and the other is proportioned to give a maximum speed in straightaway operation.

SERVICE POSSIBILITIES

The success of the motor-coach, after all, hinges primarily on the engine. The car must be designed with this thought always uppermost. The engine must be one that will stand up under the severest service. It must be capable of operating continually at high speed and under wide-open throttle with the minimum of vibration. Everything must be accessible and so arranged that repairs can be made quickly.

The gasoline-driven railroad motor-coach will enable many branch lines and short lines that are now operating at an enormous loss to be converted to a money-making basis on account of the low cost of operation, maintenance and the like. A 13-ton car can be operated for 25 to 35 cents per car-mile, as against a cost of $1 to $2 for steam operation. The initial cost is low.

Frequent service could be given where it is not justified now by a steam train. A freight-carrying unit might be installed in conjunction with a passenger unit at an initial cost much below that of a passenger unit. The gasoline motor-coach as a unit possesses many advantages over steam and electrified service; namely, speed, frequent service, low cost of operation, a crew of one or two men, the elimination of the usual terminal facilities

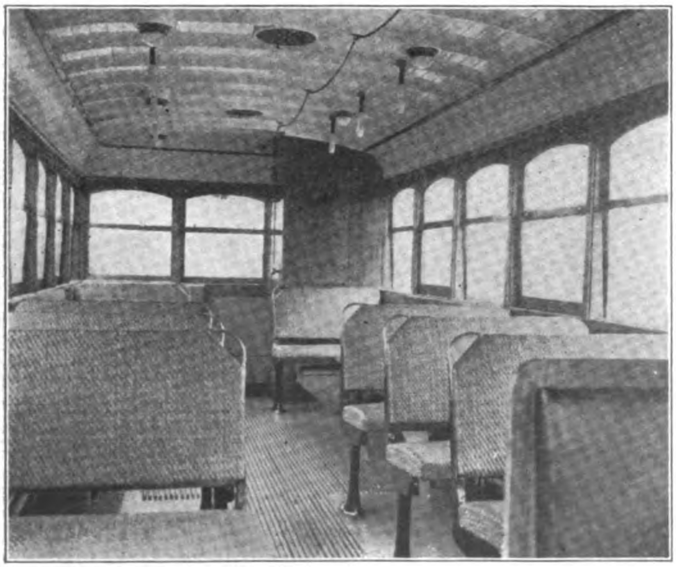


FIG. 11—INTERIOR OF THE CAR SHOWN IN FIG. 10

and a great reduction in the initial cost. In conclusion, the railroad motor-coach is primarily designed for service where steam operation is too expensive and frequent service is desired.

There is a small branch-line in the Blue Ridge Mountains that was getting only about $30 per day in passenger revenue while the operating cost with a steam train was $46 per day. A 30-passenger gasoline motor-coach, illustrated in Figs. 10 and 11, was installed and is running for about $10 per day or about 11 cents per mile; this is the total operating cost. Coal docks, water tanks, cinder pits, hostlers and the like are not necessary. That branch line, running from "nowhere to nowhere," is now making $20 per day profit. This sort of operation is a live issue with the railroads. As a result of this showing this road has put on a second car, and it is doubling its service voluntarily because it pays. The larger cars will vary in operating cost according to their capacity and speed. A car of this type can be operated for 20 to 30 cents per mile, depending upon the track, speed, number of stops and other conditions.

THE DISCUSSION

C. CHANDLER:—Mr. Hall, vice-president and general manager of the New Orleans & Lower Coast Railroad, states that the operation of its gasoline railroad motor-car is costing 15 cents per mile, without depreciation. I understand that a trailer, to be installed in the near future, has been ordered. The Illinois Central Railroad is, of course, very much interested in anything that will reduce operating expenses. With this idea in mind the management has started an investigation covering the use of motor cars on branch lines. At the present time we do not know just what is desired in the way of a railroad motor-car. On account of the race question in the Southern territory, it is possible that a car which would be satisfactory on the Northern lines would not answer at all in the Southern districts. The investigation so far indicates that the handling of freight by motor car is not desirable.

CHARLES O. GUERNSEY:—I think that about 75 hp. would be the outside limit of engine capacity. Where the roads are fairly flat, it is possible to haul a light trailer behind one of the cars. Almost every case has to be considered alone. I would say that when hauling passengers and no express on an ordinary road having less than 1-per cent grades, from 50 to 70 people could be handled. The time schedule is a governing factor.

A MEMBER:—I notice that in riding there seems to be a perpetual jar. The operation of the engine jars everything loose. The windows rattle and you think there is a hail-storm. It must be due to the constant rattling of the engine. I am wondering if that has been overcome since I rode on a gasoline coach, operated by the Ann Arbor Railroad and the Pere Marquette Railway several years ago.

MR. GUERNSEY:—That car on the Ann Arbor line weighs 54 tons. This means that they must have about 200 hp.; that they must go beyond the commercial limits of the gasoline engine to get sufficient power; and that the big heavy engine vibrates to such an extent that it becomes uncomfortable for passengers. The thing we need to do is to make the car light enough to conform to the size of engine that has been proved. With a light car it is possible to get better acceleration with 60 hp. available than they got with 200 hp. Another point is that those heavy cars, because of the enormous weight and power required to move them, cost almost as much to operate as a steam train. The operating cost varies.

We get a different report everywhere we go. Conditions vary, of course.

MR. CHANDLER:—They are operating at a cost of about 40 cents per mile, with a passenger revenue of about 55 cents per mile, not including revenue from milk, express or mail service. That is the only car of that kind that I know of.

L. G. PLANT:—Light local-passenger service, which involves the operation of a steam locomotive and not more than three coaches, is undoubtedly the most expensive service performed by the railroads in proportion to the revenue received. Testifying before the Interstate Commerce Commission in regard to railroad problems and the efficiency of railroad management, Mr. Willard, president of the Baltimore & Ohio Railroad, recently made this significant statement regarding railroad passenger service:

> The expensive feature of the passenger business is not the ordinary running of the heavy through-trains. My observation is that usually they pay. The expensive part of the business is the running of thousands of miles of unprofitable passenger service on light branches, or light portions of main lines where the people demand the service. I have not made figures recently, but I recall that a few years ago some 30 per cent of the total passenger traffic carried by the Baltimore & Ohio Railroad earned less than the actual out-of-pocket cost. We earned less money than the actual wages paid to the trainmen and enginemen, for the coal burned and the oil used, and there was not a single train either that we could take off. They were established and have been run in response to the public demand; and I am not in position to say that the public is not entitled to the service. Usually the service consisted of two passenger trains in each direction over branch lines. The business did not justify running the trains, but the people had no other way to travel.

While Mr. Willard did not refer to the steps taken by his railroad toward reducing the cost of this service, it is understood that the Baltimore & Ohio Railroad is actively interested in the possibilities of the gasoline motor-car. *The Railway Review* also is interested in the possibilities of the gasoline rail motor-car and from the time the first article describing this class of equipment appeared in this paper some months ago, no one subject has created greater interest. It is our belief that in the present and prospective designs we have the solution to the problem outlined by Mr. Willard. In fact, my experience some years ago in the purchase of a small railroad in a mountainous section of Virginia convinced me of the value of the gasoline rail motor-car as a means for handling light local-passenger traffic. This road had been a losing venture for many years and, in planning for the future operation of the road, I conceived the idea of purchasing a motorbus, such as I had seen operating on the streets of Birmingham, one of the first Southern cities to adopt bus transportation extensively, and equipping it with flanged wheels. I have since been a firm believer in the gasoline railroad-car as a practical means of reducing the cost and improving the character of local-passenger service, provided the elements that have contributed to the success of the motor truck, such as simplicity of operation and maintenance, light weight and low first cost, are not ignored. It is interesting to observe that a recent investigation into the use of gasoline railcars on short-line railroads revealed the fact that in every instance the substitution of gasoline service for the steam locomotive had converted an annual deficit into a surplus, although in many instances gasoline equipment of the most primitive type was being used.

In view of the interest on the part of railroad officials, it is perhaps puzzling to the builders of gasoline cars to find so much hesitancy on the part of the railroads toward the actual purchase of the equipment, even where the capacity of the railcar has been demonstrated and the operating economies are obvious. If a motor-truck builder can convince a local coal-dealer that he can save $1000 per year through the substitution of a motor truck for team delivery and the dealer can finance the purchase, the sale of the truck is ordinarily assured, but it should be borne in mind that in this case the profit from the investment will accrue directly to the dealer, whereas the meager profit resulting from railroad operation is paid to stockholders whose connection with the actual details of the service is usually remote. I do not wish to indicate that railroad managers and employes are not making a sincere effort to improve the efficiency of railroad operation in every possible way, but their less responsive attitude can be ascribed chiefly to a lack of business initiative that is more or less characteristic of all large institutions.

While railroad officials are undoubtedly interested in the development of the gasoline railcar, they are inclined to see the objections to this equipment rather than its possibilities and to stress the importance of certain changes in the details of the construction of the cars instead of the lower operating cost of and small investment required for the equipment in its present form. Railroad officials also are inclined to want equipment that will duplicate the existing steam service. In the situation described by Mr. Willard, the question is not necessarily that of duplicating the performance of the two passenger-trains that operate daily in each direction over branch lines, but of expanding the service and of building up additional traffic at less cost than is possible with steam operation. It is conceivable that on local service radiating from a shopping center a gasoline railcar could be maintained in continuous operation throughout the day and handle a greater number of passengers at less cost than is possible with a steam train. It is reasonable to assume that more frequent service would stimulate a greater volume of traffic. I have in mind a typical situation on a short branch-line where there is but one train into town early in the morning and one train returning late in the evening. The installation of a frequent gasoline railcar service throughout the day would not only attract more travel, as in the case of the electric line, but enable the railroad to handle a greater number of passengers per day at a lower cost. Notwithstanding these obvious possibilities, we find that one railroad is not interested in gasoline railcars unless the cars are designed to seat 72 people, while another road insists that the equipment must be capable of pulling four or five box-cars if necessary. Still another railroad expects to use its old coaches weighing some 60,000 lb. as trailers rather than charge this equipment off its books, this being a case where the railroad has failed to make a sufficient depreciation allowance for the coaches and is carrying them at a higher book-value than they are actually worth.

On the other hand, there are certain requirements of the railroad, particularly those that relate to safety, the importance of which may not be fully appreciated by manufacturers, but which designers, in their desire to make the equipment as light as possible, should not ignore. Builders of gasoline railcar equipment cannot yet be said to have a full understanding of the problems involved and an appreciation of the fact that it is not what the car is, but what it will do, that interests the practical railroad man. Generally speaking, the manufacturers

have not differentiated between the problem involved in marketing trucks and the problem of selling to the railroads. It will necessitate an organization that understands the railroad problem and the full significance of departmental relationship that is foreign to the ordinary commercial organization. I believe that the arrangement to market gasoline railcars through local agencies that receive a large commission would not appeal to the railroads, and that the service feature so essential to the truck business would not meet with favor on the average railroad. It will be necessary in a majority of cases not only to demonstrate that the car will operate, but to show the railroads where they can use the cars to advantage; and this involves a large amount of educational work.

CHAIRMAN LON R. SMITH:—Nobody would claim that the gasoline railroad-car would take care of the service that demands a steam outfit. The branch-line service, which is necessary on a steam line, makes the advantages of these gasoline-propelled cars of interest. Most of the railroad people seemingly are interested in the subject. The question is, what type of car will best meet their needs.

J. D. RISTINE:—The railroad motor-coach is of vital interest. It can be used successfully where the cost of steam operation is too great and frequent service is required, at a cost approximately one-fifth to one-third that of a steam train. The gasoline engine is, first of all, the all-important thing to be considered. We must not fail, however, to consider its limitations and construct a coach accordingly. For the present we are limited to engines of about 75 hp. as a maximum, for the reason that larger engines have not been perfected to the point where they are commercially successful. Coaches should be designed so that minor changes can be made to meet conditions, but in all cases the total weight must be given careful consideration.

MARK A. SMITH:—The car that Mr. Guernsey has described was driven from Wabash, Ind., today, carrying a group of railroad men as passengers. The running time from Peru, 75 miles distant, was 2 hr. and 35 min. The time on the steam train would have been 13 min. longer. The maximum speed attained in this run was 48 m.p.h. The coach took the grades at from 25 to 30 m.p.h. The gasoline consumption from Wabash, 91 miles, was 15 gal., including idling; or 6 miles plus per gal.

A. L. NELSON:—That is very good economy.

ALBERT KING:—I am a road foreman of engines on the Wabash Railroad. The car mentioned is a very simply constructed affair. I ran it myself the first time I was ever on it. I own an automobile but I do not consider myself an expert driver. The car has many features that could be used to advantage by railroads. I take it that they all have the same conditions that we have. A steam locomotive never operates at its maximum efficiency except at or near its maximum capacity. If it had sufficient tractive power to haul 2000 tons over a 100-mile division and the coal consumption were, say, 6 tons, under the best and most economical management it would not take one-half that load over the same division with one-half that amount of fuel. That is why the operation of branch lines is so expensive. The only thing that I could suggest to Mr. Guernsey and his fellow-workers is to perfect a car that will meet all the requirements of branch-line or local-passenger service. The cars that are being designed are a little too small. A 35, 40 or 50-passenger car is hardly large enough, and the trailer feature is not, to my mind, desirable. The men engaged in building railroad motor-cars have a broad field.

GEORGE A. WEIDELY:—Why did Mr. Guernsey specify the limit at 75 hp.? Engines used in the fire-pump service develop 100 to 120 hp. continuously and economically under severe conditions, making runs of 30 to 50 hr. It seems to me that engines of that kind would be very satisfactory in railroad service. I am also wondering what Mr. Guernsey's idea is as to the number of cylinders most suitable for that service.

MR. GUERNSEY:—I may have been a little unfortunate in the manner in which I stated the horsepower limit. We do not say that we have the ultimate thing. I see no reason why ultimately it should not be possible to have a larger car with more power but with an increased number of cylinders. Of course, we must remember that increasing the capacity of the car increases the fuel cost and the operating cost. It is in the haul in which you get 40 people sometimes, but usually 6 or 8, where these things really belong. I think there are great possibilities for the future development of these cars.

IRA C. KOEHNE:—A few weeks ago I took the train from New Haven toward Waterbury and rode about 40 min. on a gasoline-driven railcar. That ride was a punishment on account of the excessively short jerky vibrations, not only from the engine, which is in the front of the car, but from the rear axle as well. I think it would be a menace to the health of passengers to ride on a car of that construction. I can see that the construction pointed out by Mr. Guernsey has features that cushion the vibrations. Economy of operation is a very desirable consideration, but to make the thing a success the comfort of the passengers must be conserved.

E. O. MANSUR:—I have had the benefit of all the grief that came with the introduction of the gasoline rail motor-cars, having operated them since 1914. I am operating them on the Akron, Canton & Youngstown Railway at present.

The gasoline-electric motor-car with which I am most familiar has an eight-cylinder engine, 8-in. bore and 10-in. stroke; it runs at 550 r.p.m. and gives 175 hp. This engine is direct-connected to a 600-volt series-wound generator that furnishes the current for two 100-hp. motors on the front truck. The most trouble was encountered because the steam engineers did not understand the gasoline engine and did not know what to do in emergencies. A short time ago an engineer called me up from the road, reporting that a valve had broken and dropped into the cylinder. He wanted to know what to do. I told him to shut down the engine, but it was too late. The damage was done. These engines have too many moving parts and are too complicated for an inexperienced man to watch.

The electrical equipment has, and has not, given trouble, depending upon where the car is taken care of. In some cases the men do not realize the necessity of keeping the oil away from the motors, and I know of one case where an armature was saturated with oil. When the oil is kept out of the motors, no trouble occurs. I am of the opinion that these cars could still be made to operate efficiently if an automotive engineer would study them and make a few changes in ignition and carburetion.

While these cars weigh 35 tons and 60 per cent of this weight is on the front truck, I believe we can still go ahead with the large type of car and reduce some of the weight by using a high-speed engine. With the past types of construction, the railroad men have condemned the gasoline engine for railroad service because of the failures and delays on the road that are due to the break-

(*Concluded on p.* 283)

Coming Meetings of the Society

THE ABERDEEN MEETING

MEMBERS of the Society of Automotive Engineers, the American Society of Mechanical Engineers and the Army Ordnance Association have been invited by the Ordnance Department of the United States Army to witness a program of test-firings and demonstrations of post-war ordnance materiel at the Aberdeen Proving Ground, Md., Friday, Oct. 6, 1922. The arrangements for this year's meeting will be similar to those for the meeting in October of last year. There are no accommodations permitting visitors to remain overnight at the reservation, thus necessitating the limitation of the meeting to a single day. The program will start about 9:00 a. m. allowing time for the arrival of the regular through trains on the Baltimore & Ohio and the Pennsylvania Railroads. Definite railroad schedules will be included in the October issue of THE JOURNAL.

Special badges admitting Society members to the Proving Ground on the day of the firings will be issued to those requesting them from the Commanding Officer on the blank at the bottom of this page. The application blank must be accompanied by check for $3 to cover the cost of luncheon and dinner to be served on the reservation. Reservations for meals are irrevocable since all food and facilities for this service are transported to the Grounds by a Baltimore caterer and it is necessary that he have definite advance notice of the number to be served. No provision will be made to serve meals to those who fail to secure tickets in advance of the meeting date. Breakfast should be eaten on the trains before arrival since luncheon will be the first meal served on the Proving Grounds.

The Ordnance Department makes particular mention of the fact that, due to limited accommodations at the Proving Ground, it will not be practical to permit ladies to witness the firings. Absolutely no exception to this rule will be made and members are requested to avoid an unpleasant situation by complying with this request.

Arrangements will be made to have the early-morning Pennsylvania and Baltimore & Ohio through trains stop at the reservation. Pullman reservations from New York City may be made through the Society offices at 29 West 39th Street, New York City. The acknowledgment of the reservation will state the time of departure and the cost of the ticket and Pullman. Payment may then be made by check and the tickets will be forwarded to the member ordering them.

The details of the firings and demonstrations have not been announced at this time but a representative conception of the program may be had from a reading of the notice of the 1921 Aberdeen Meeting on p. 217 of THE JOURNAL for October, 1921.

ANNUAL MEETING PAPERS

The Meetings Committee is now selecting authors and papers for the technical sessions at the Annual Meeting in New York City, Jan. 9 to 12. If you wish to be included in the program, submit an outline of your proposed paper at once. Be sure that its subject is general in interest and one of the matters demanding close attention at this time. The limited period available for presentation and discussion necessitates the concentration of the papers on problems that are general in their appeal and whose solution is urgent. It is doubtful if any request for inclusion in the program can be granted after Oct. 1.

THE DETROIT PRODUCTION MEETING

THE program and arrangements for the national automotive Production Meeting of the Society in Detroit, Oct. 26 and 27 are well advanced at this time. The General Motors Corporation has very kindly provided quarters for the meetings in the massive General Motors Building on the Grand Boulevard. The building is easily reached from all sections of the city by surface car, motorbus or automobile, and is in the geographical center of the manufacturing district. It is one of the largest structures of its kind and has nearly all of its space occupied by automotive interests.

ABERDEEN ORDNANCE MEETING

OCT. 6, 1922

APPLICATION FOR TICKET OF ADMISSION AND RESERVATION FOR MEALS

Important

1. Only citizens of the United States can attend this meeting.
2. No one engaged in the manufacture of arms or munitions for a foreign government which is at war is permitted on the Proving Ground.
3. No ladies will be permitted to attend this meeting.
4. Make all checks payable to the Commanding Officer, Aberdeen Proving Ground.

(over)

The two-day program of the Production Meeting will include two morning sessions for the reading of papers, two series of factory visits in the afternoons and a Production Dinner on the first evening. The meeting is to be national in scope and it is expected to attract a large number of production executives to Detroit.

A Novel Meeting Plan

The committee charged with the selection of papers and conduct of the meetings has laid out a program that is unusual in plan and promises to be of exceptional interest and value. The committee is composed of Karl Herrmann, chairman, C. Harold Wills, E. F. Roberts, T. J. Litle, W. J. Alles and F. A. Whitten. At the first meeting of the committee, Mr. Wills suggested that the production meetings be devoted to a symposium on production problems, several passenger-car producers being invited to contribute their experiences. The several papers are to be read by each of the contributors in advance of the meeting and their contents coordinated so that the thought expressed will be fairly representative of the entire group. Mr. Wills' suggestion met with unanimous approval and the plan is being carried out at present. The following companies have members of their production staff preparing material for the symposium: Dodge, Ford, Packard, Studebaker and Wills. The cooperation of other companies is expected so that a total of 10 organizations will be represented by their production officials on the program.

The subject matter of the papers may be subdivided into three groups. First, manufacturing problems that are common to all producers will be enumerated in order that the thought of production engineers will be centered on the solution of them. Second, those producers who have made a particular study of any of these problems will contribute the results of their research for the benefit of the industry. Third, each producer will describe the outstanding developments in machine tools or methods in his factory that he believes to be of general interest to production men. A number of problems have come to light under the first heading; the cutting of accurate gear teeth, the manufacture of silent-running gearing, the cutting of accurate threads and the production of perfect cylinder bores. Authorities on machine-tool design will be invited to contribute discussion on certain of the problems at the meeting after the formal presentation of the papers. It is not the intention to preprint either the papers or discussion for general circulation in advance of the meetings.

Factory Inspection Visits Planned

K. K. Hoagg, vice-chairman of the Detroit Section, is heading a committee that will arrange the series of factory inspection trips each afternoon of the Production Meeting. Immediately following the morning meetings, luncheon will be served in the General Motors Building convenient to the meeting room. This scheme will enable the members and guests to remain together for a short social period when informal personal discussions will be in order. Following luncheon, the members will be taken by automobile to the larger Detroit factories for inspection trips which will be conducted in each case by the production executives in charge of that factory. Several plants will be visited simultaneously necessitating a member choosing the particular ones of the greatest interest to him. The inspection trips will differ from those generally taken on such occasons. The routes through the shops will be selected to enable the production men to inspect only those departments whose equipment, layout or methods are unusual. A period will be provided at the end of each visit to accommodate members who wish to see a particular department of interest to them.

The Production Dinner

The Production Dinner will be held on Thursday evening, Oct. 26, the location to be announced later. The dinner will be strictly informal and evening clothes conspicuous by their absence. The dinner talks will be short and to the point. The speakers will be representative executives of the automotive industry and their remarks will be confined to the industry's business. The purpose of the dinner is to promote intimate friendships among the men who build motor vehicles, to provide an enjoyable evening's entertainment and to inspire all factory men to keep automotive quality foremost in their minds that industry may endure.

Non-Member Production Men Welcome

It is recognized that a large number of production men who are not members of the Society will want to attend the meetings, inspections and the dinner. They should be assured by the members that their participation in all three of these activities is not only welcome but invited. This meeting is a producers' meeting. It is intended for shop executives, and their presence in large numbers is essential to its success. Each member should cooperate to the extent of personally inviting the men in his factory to participate in the Production Meeting on Oct. 26 and 27.

THE COMMANDING OFFICER,

**ABERDEEN PROVING GROUND,
ABERDEEN, MD.**

As a member of the Society of Automotive Engineers, I accept the invitation to visit the Aberdeen Proving Ground on Friday, Oct. 6, 1922. Please provide luncheon and dinner for me that day, for which I enclose remittance for $3.

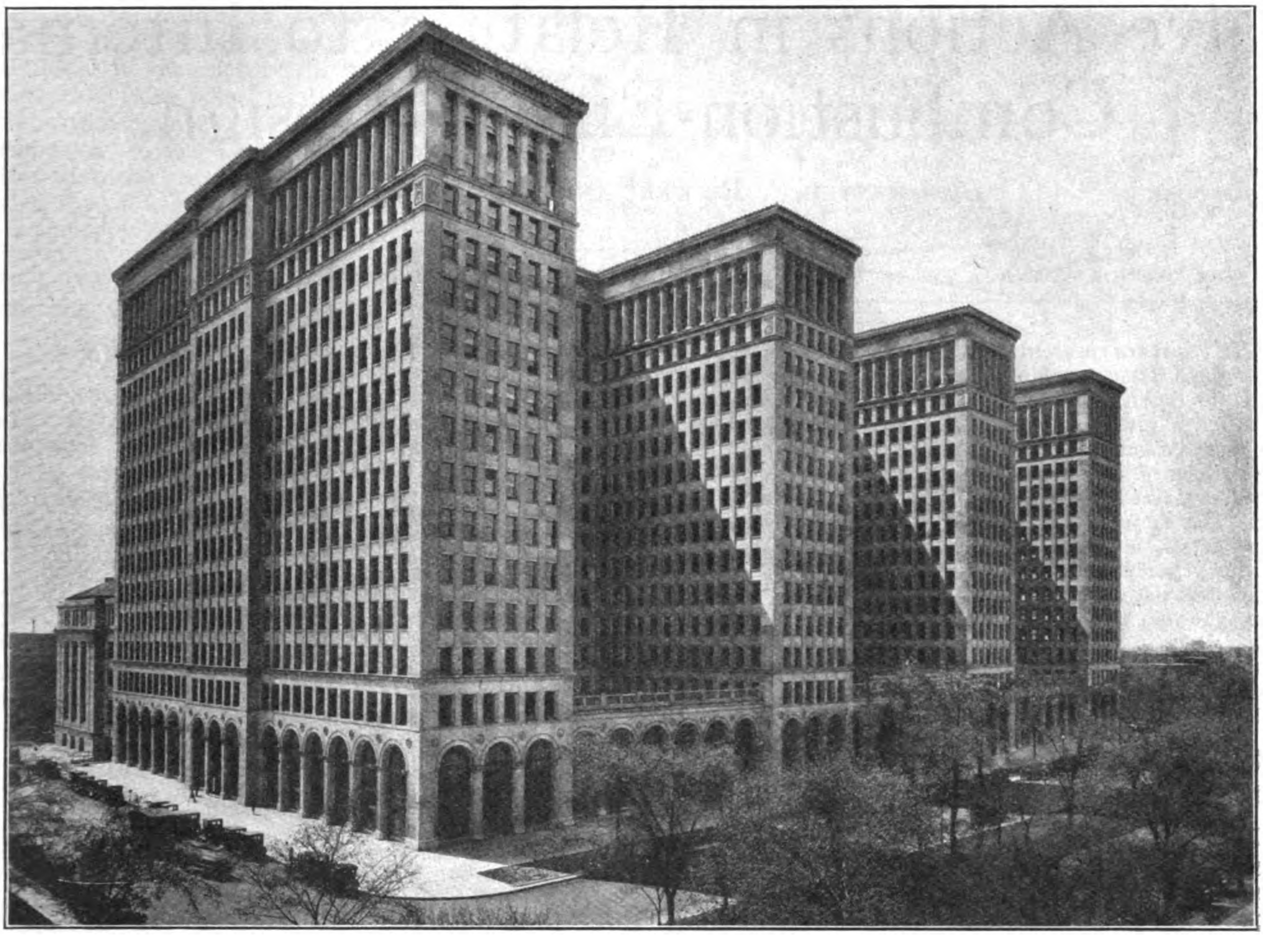
I hereby certify that I am a citizen of the United States and am not connected with any company manufacturing munitions for any foreign government.

Please forward identification badge and luncheon and dinner tickets to me at

. . . .
Signature

. . . .
Address

. . . .
City



OFFICE BUILDING OF THE GENERAL MOTORS CORPORATION, DETROIT, IN WHICH THE PRODUCTION MEETING OF THE SOCIETY WILL BE HELD, OCT. 26 AND 27

GASOLINE RAIL MOTOR-COACH

(*Concluded from p.* 280)

ing of small parts and to having them operated by inexperienced men.

W. H. BUDERUS:—I am interested in lubrication. Our greatest trouble in the earlier cars was on account of the oil in the crankcase becoming diluted so quickly. Those crankcases held 26 gal. The lighter type of engine, which Mr. Guernsey has explained, holds from 2 to 3 gal., a tremendous saving. In regard to the engine of lower horsepower, I happen to know that a car is now under construction which will use a 100-hp. engine. I understand the Baltimore & Ohio Railroad estimates that it could place 50 gasoline railcars on its system alone; that is, there is an opportunity for this number of cars to replace unprofitable steam-train service.

MR. GUERNSEY:—It is well within the possibilities to build a car for way-freight service and for use where it might be required to handle one or two loaded cars. In fact, the operation of these cars can be pretty well illustrated by what is being done on the electric interurban lines. They haul way freight in carload lots and by a trailer. These cases come up on branch lines, where the speed need not be more than 20 or 30 m.p.h. A car-and-trailer arrangement can be worked out, provided the speed conditions and grades are not too severe. We run about 700 miles per gal. of oil. As to the noise from the engine, and the vibration and rough riding due to the reaction from the driving axle being objectionable to the passengers, these are things we have set out to overcome. They are one of the reasons we are using two four-wheel trucks. The point was raised regarding bus competition. A car running on the rails can starve a bus man to death. A 30-passenger bus operating on the highway costs about 30 cents per mile to operate, while one running on rails will cost about 12 cents per mile. These cars, because of their light weight, require very little expenditure for maintenance.

Valve Actions in Relation to Internal-Combustion-Engine Design

By Chester S. Ricker[1] and John C. Moore[2]

Mid-West Section Paper — *Illustrated with* Diagrams and Charts

THE authors present and discuss the results obtained from combined road and laboratory tests made to determine the amount of power required to maintain a given car speed. The specifications of the car and its engine are stated and the variable-ratio rocker-arm of the engine is illustrated and its advantages explained, together with those of the valve-timing. The subject of manifold gas-velocity is treated in some detail, inclusive of a diagram showing the hot-spot or vaporizing device that was used.

The test data are reduced to curve form, eight charts being shown. The curves include those for brake horsepower, indicated horsepower, comparative performance, performance at different throttle-openings and at different loads, fuel consumption and indicated thermal efficiencies.

THE first tests made were to determine the road resistance offered by a car running at a constant speed over a ½-mile course. This was the graveled Milton-Pike road, north of Connersville, Ind. The average outdoor temperature was 45 deg. fahr. and the barometer read 29.4 in. To determine the same points in the laboratory later with the engine and carbureter, graduated cards were fastened on the carbureter and on the ignition control so that we knew the exact point at which each of these instruments was set at each speed at which the car was run. The purpose of these cards was to enable us to duplicate the speed test on the dynamometer in the laboratory and determine what horsepower was required to maintain the car at each given speed. Another purpose of the tests was to make the data which we obtained comparable with the data A. L. Nelson obtained in 1920. The curves shown were plotted from the data obtained during these tests. The Lexington car with which the tests were conducted and the Ansted engine used in it had the specifications shown in Tables 1 to 5.

The type of rocker-arm shown in Fig. 1 is interesting; it has a variable ratio. It opens the valve slowly and then, as the valve continues to open, the ratio on the rocker-arm changes and, when the valve is wide-open, the end-point gives the rocker-arm nearly three times the ratio that it has at the start. In other words, the ratio

TABLE 1—SPECIFICATIONS OF LEXINGTON MODEL ST CAR

Weight with Driver, Spare Wheel, Tools and Tanks Full, lb.	3,405
Weight, Less That of 159-lb. Driver, lb.	3,246
Allweather-Tread Cord Tire, Size, in.	32 x 4
Average Roll of Rear Wheel, in.	101
Number of Wheel Revolutions per Mile	628
Number of Engine Revolutions per Mile	2,908
Exhaust Cut-Out	Open
Rear-Axle Gear Ratio	4⅝ to 1

TABLE 2—GENERAL SPECIFICATIONS OF THE ENGINE USED

Type	Valve-in-head
Number of Cylinders	6
Bore, in.	3¼
Stroke, in.	4½
Piston Displacement, cu. in.	224
Number of Crankshaft Bearings	3
Length of Front Main Bearing, in.	2½
Diameter of Front Main Bearing, in.	1¾
Length of Center Main Bearing, in.	2⅝
Diameter of Center Main Bearing, in.	2¼
Length of Rear Main Bearing, in.	3⅜
Diameter of Rear Main Bearing, in.	2⅝
Length of Crankpin, in.	1½
Diameter of Crankpin, in.	2¼
Lubrication System	Full Pressure
Lubrication Regulation	Vacuum
Cooling Regulation	Rayfield Thermostat
Carbureter Type	Rayfield Model M Horizontal
Carbureter Size, in.	1½

TABLE 3—AVERAGE WEIGHT OF 10 SAMPLES EACH OF MOVING PARTS TAKEN AT RANDOM

	Oz.
Cast-Iron Piston	24.32
Piston-Ring	0.85
Piston-Rings per Piston	1.70
Piston-Pin	3.44
Connecting-Rod Assembly, Small End	7.90
Connecting-Rod Assembly, Total Weight	45.47
Valve Alone	3.31
Valve Push-Rod with End	3.29
Valve Lifter, Roller and Pin	6.23
Valve Rocker-Arm Complete	4.34
Valve-Spring Retainer and Key	0.82

TABLE 4—PISTON AND CONNECTING-ROD SPECIFICATIONS

Piston Material	Cast Iron
Length of Piston, in.	3 17/32
Number of Piston-Rings	2
Width of Piston-Rings, in.	3/16
Length of Piston-Pin, in.	2⅞
Outside Diameter of Piston-Pin, in.	⅞
Inside Diameter of Piston-Pin, in.	11/16
Connecting-Rod Section	I-Beam
Length of Connecting-Rod, in.	8½
Number of Connecting-Rod Cap-Bolts	2
Diameter of Connecting-Rod Cap-Bolts, in.	7/16

TABLE 5—VALVE-GEAR SPECIFICATIONS

Diameter of Valves, in.	1.7500
Clear Opening of Valve Port, in.	1.6250
Diameter of Valve-Stem, in.	0.4120
Length of Valve, in.	5.6870
Combined Pressure of Dual Valve-Springs, lb. per sq. in.	75
Valve Rocker-Arms	Moore Type
Valve Rocker-Arm Ratio, at Opening	1.265 to 1
Valve Rocker-Arm Ratio, Full Open	3.047 to 1
Diameter of Valve Push-Rods, in.	0.2500
Diameter of Camshaft, in.	1.1250
Radius of Cam Base-Circle, in.	0.6250
Radius of Cam Point, in.	0.8437
Lift of Cam, in.	0.2187
Diameter of Valve-Lifter Rollers, in.	1.1250
Intake-Valve opens	10 deg. late
Intake-Valve closes	56 deg. late
Exhaust-Valve opens	50 deg. early
Exhaust-Valve closes	10 deg. late

[1] M.S.A.E.—Consulting engineer, Indianapolis.
[2] M.S.A.E.—Chief engineer, Lexington Motor Co., Connersville, Ind.

is approximately 1¼ to 1 when the valve is cracked open, but when the valve is wide-open, the ratio is almost 3 1/16 to 1. This brings up an interesting point in connection with the rocker-arm for the exhaust-valve, because this valve opens against the full pressure in the combustion-chamber. This gives the maximum leverage or about a 1¼ to 1 ratio when the valve is cracked open; after that, the pressure being equal on both sides of the valve, the valve-gear has less work to do and this becomes less important.

Another point in connection with this variable-ratio rocker-arm is that a rocker-arm on the top of an engine rises and falls with the expansion of the cylinder-block. The push-rod may expand the same amount as the cylinder-block does or it may not. If the push-rod does not expand the same amount as the cylinder-block and the valves are adjusted when the engine is hot so that they have very little clearance, there is the possibility of holding the valves open when the engine is cool, due to the variation in the push-rod and the cylinder shrinkage. That is clear to anyone who has had anything to do with an aluminum-cylinder engine. In fact, it has been found necessary to allow from 0.013 to 0.015-in. clearance between the rocker-arms and the valve-stems on an aluminum-cylinder engine when they were hot, to insure that the valves close when the engine is cold. Here we have a very small ratio, practically a 1 to 1 ratio on the valve-gear at the closed point; hence, this design is not so susceptible to variations in the length of the cylinder-block or the length of the push-rod, due to manufacturing difficulties or to temperature. The subject of the velocity of opening and closing the valve is likewise noteworthy. The rocker-arm starts off with the 1¼ to 1 ratio and ends with a wide-open valve having a 3 1/16 to 1 ratio. That undoubtedly affects the valve action. In fact, we know it affects the valve very noticeably in regard to quietness.

The valve timing of the Ansted engine is such that the exhaust closes and the intake-valve opens at 10 deg. past top dead-center; the intake closes 56 deg. late, while the exhaust-valve opens 50 deg. early.

The intake-valve of the engine mentioned by A. L. Nelson in his paper entitled Fuel Problems in Relation to Engineering Viewpoint[3] opens at 4 deg. past top dead-center and closes 60 deg. late, or only 4 deg. later than the intake-valve of the Ansted engine; hence, so far as the intake-valve timing is concerned, both engines are practically the same. The exhaust-valve of Mr. Nelson's engine opens 52 deg. before bottom dead-center.

For convenience in comparing the speed of the car and the revolutions per minute of the engine, all the charts are made on both a miles-per-hour and a revolutions-per-minute basis, as follows:

Car Speed, m.p.h.	Engine Speed, r.p.m.
10	484
20	968
30	1,454
40	1,936
50	2,420
60	2,908

For each 10-m.p.h. increase of the car speed, there is about a 500-r.p.m. increase in the engine speed, and this is an easy way to judge the curves.

Manifold Gas Velocity

In computing the velocities of gas in the manifolds of a high-speed engine, some problems develop. A sectional view through two manifolds is shown in Fig. 2; the upper one represents the manifolds that A. L. Nelson uses in his engine and the lower one a cross-section of the Ansted-engine manifold. In both engines the carbureter opening is given as a and the area of the longitudinal manifold in the cylinder-head as b. The feature in the comparison of the two engines and their performance is that the area at a in Mr. Nelson's manifold is large enough to maintain an average gas-velocity of 181 ft. per sec. when the engine is running at 3000 r.p.m. Apparently, he assumes in his design that only one-half of the gas comes over in this manifold, and therefore he gets a velocity of 175 ft. per sec. for one-half of the manifold; as the section b is reduced to 1½ in. from the section a which is 2 3/16 in. in diameter. These figures are computed for an engine of 295-cu. in. displacement. One difference exists between the two engines; the performance must prove whether it is for better or worse. In the Ansted engine, the manifold is 1¾ in. in diameter. The assumption made in its design was that more or less gas is handled over the entire manifold; so, instead of reducing this to maintain a uniform velocity for one-half of the gas, as Mr. Nelson assumes, it has been maintained at the same size. This difference is a manufacturing proposition. We question whether it was scrutinized very closely when the original design was made, but it worked so well that it has never been changed. With this 224-cu. in. engine we find by computation on the same basis that Mr. Nelson used that the gas velocities are very different. We find that the gas velocity at section b in this manifold is 97 ft. per sec., although the gas velocity in the manifold is 194 ft. per sec. at the same speed.

This is not so much a matter of comparison as to induce thought along the lines of what size of manifold

[3] See The Journal, February, 1921, p. 101.

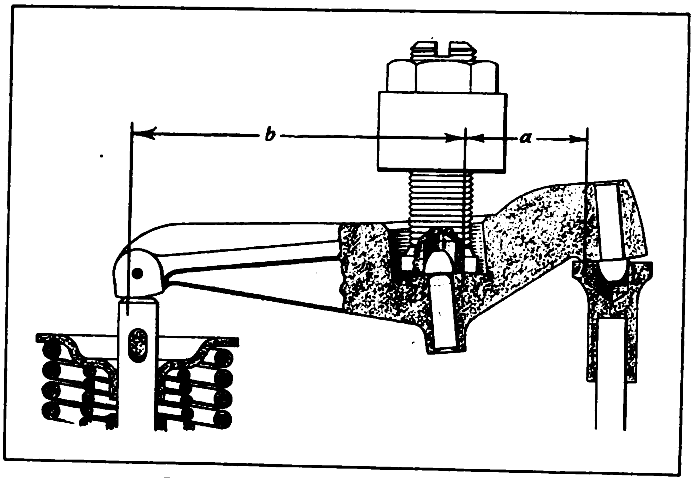


Fig. 1—Variable-Ratio Rocker-Arm
The Ratio of a to b Is 1:3 1/16

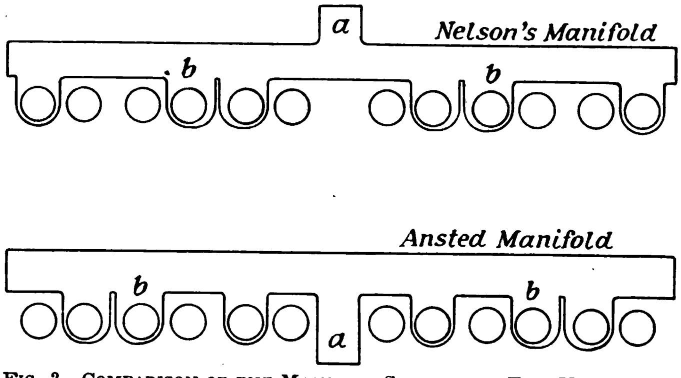


Fig. 2—Comparison of the Manifold Sections of Two High-Speed Engines, Showing the Difference in the Disposition of the Carbureter and Valve Ports

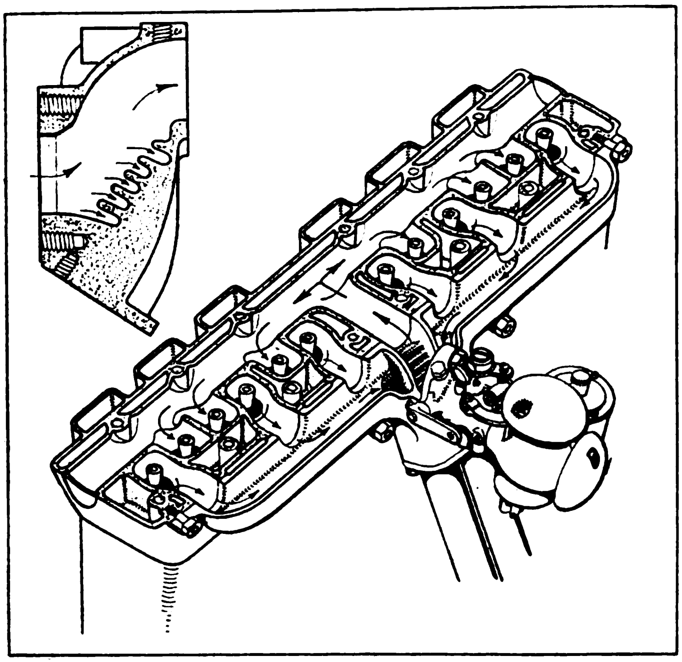


FIG. 3—CROSS-SECTIONS OF THE CYLINDER-HEAD AND THE HOT-SPOT GRID
The Exhaust Gases Pass along Either Side of the Hot Spot

should be used in a six-cylinder head of this design. Some of the carbureter men may have some comments to make in regard to those figures. The gas-velocity figures are for an engine speed of 3000 r.p.m.; at 400

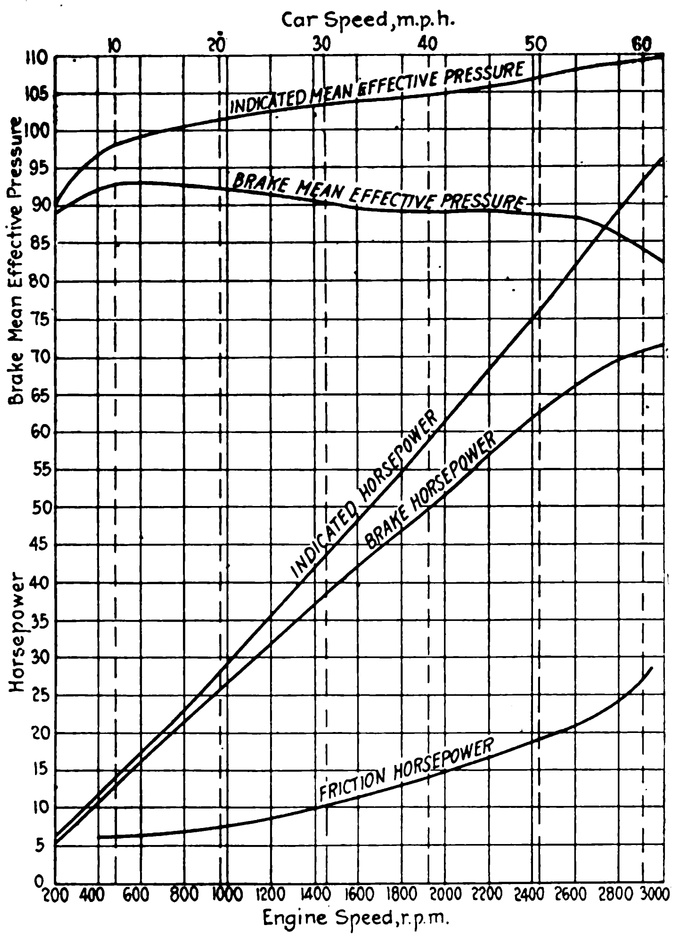


FIG. 4—CURVES SHOWING THE BRAKE HORSEPOWER AND MEAN EFFECTIVE PRESSURE DEVELOPED

r.p.m. in the low range, the results are very startling. For example, on the Nelson engine, the gas velocity is reduced to 47.9 ft. per sec. through section *a* at 400 r.p.m., with 51 ft. per sec. at section *b*, while on the Ansted engine the velocity in section *b* of the manifold drops to 25 ft. per sec. where the gas-velocity in Mr. Nelson's engine is maintained at 46 ft. per sec. A drop of 25 ft. per sec. ought to throw the unvaporized gasoline down into this manifold and cause loading, but apparently it does not. We believe that this is due to the character of hot-spot or vaporizing device that is used at section *a*, a diagram being shown in Fig. 3.

INTERPRETATION OF TEST DATA

Fig. 4 shows the brake horsepower obtained with the Ansted 224-cu. in. engine. It will be noticed that the brake mean effective pressure on this engine, which was calculated from this brake-horsepower curve, is unusually flat for an engine of this type. It starts at above 90 lb. per sq. in. mean effective pressure and holds it out to about 2000 r.p.m., or about 40 m.p.h. The indicated mean effective pressure is taken from the indicated-horsepower curve that was obtained by the addition of the friction-horsepower values to the brake horsepower. We get an indicated mean effective pressure of about 100 to 109 lb. per sq. in. from the indicated-horsepower curve. This brake-horsepower curve was made from an experimental engine that was thoroughly limbered up, but just after we finished the test we had the misfortune to break a valve; this necessitated the use of a new unlimbered engine taken out of ordinary production stock to obtain the friction-horsepower curve that is shown. Hence this friction horsepower is probably much higher than the friction horsepower on a well limbered engine, but we gave the freer engine the higher-friction curve to be sure that the engine received no credit for something that it did not deserve.

In the second series of curves shown in Fig. 5, we tried to interpret the data we obtained from the brake-horsepower curve so that it would be comparable with the data in Mr. Nelson's paper. He gave the performance of a 295.2-cu. in. engine with a 4¼ to 1 compression-ratio and then the results obtained from the same engine with a 5 to 1 compression-ratio. The results, plotted with revolutions per minute as the abscissas and the horsepower per cubic inch of piston displacement as the ordinates, give the three curves shown in Fig. 5. The reason they are brought to a horsepower cubic-inch-displacement basis is to make a direct comparison between the 295.2 and the 224.0-cu. in. engines. In the 5 to 1 compression-ratio engine, the performance falls below that which we obtained. This was surprising to us.

The comparison in percentage between the three curves is shown at the top of Fig. 5. Taking the Ansted-engine curve as 100 per cent, we find that the Nelson engine with a 5 to 1 compression-ratio, at from 900 to 1900 r.p.m., has about 2 per cent more power per cubic inch of displacement. We find that the old 4¼ to 1-ratio engine varied from 80 up to 92, and down again to 69 per cent of the Ansted-engine power, at speeds varying from 400 to 2800 r.p.m., the point at which the Nelson curve ends.

Fig. 6 is interesting as giving a general idea of how a standard engine performs, its torque curve being practically flat and running from 120 down to 105 lb-ft. at 3000 r.p.m. and down to 100 lb-ft. at 300 r.p.m. The mechanical efficiency of this engine fell off more rapidly from its low speed where it had about 98 per cent efficiency to 74 per cent at 3000 r.p.m.

Fig. 7 shows some surprising features. We used a

small quadrant in which the graduations of the movement of the carbureter throttle were divided into 20 divisions on the throttle arc. The throttle is absolutely tight when closed. On the Model-M horizontal Rayfield carbureter there is a small opening that allows the engine to idle when the throttle is closed. For that reason we were able to allow the engine to idle and start out with a zero reading for the throttle-opening on all of the tests. The quadrant placed on the car while it was being run on the road to determine the throttle-opening for each car speed, was graduated in 10 equal divisions. In Fig. 7 we have merely stretched them out in a straight line instead of showing them on a curve. The impressive thing is that, for the range up to 60 m.p.h. on a country road, only one-half throttle was required and, when the throttle was opened wide, the car speed only increased 10 m.p.h. The revolutions per minute of the engine correspond to the car speed and the throttle-

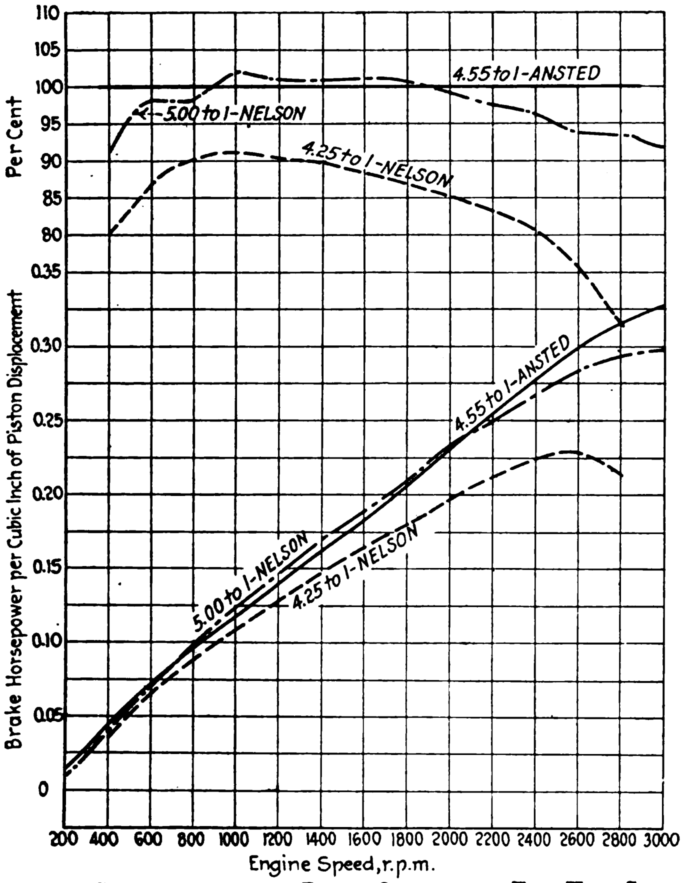


Fig. 5—Comparison of the Power Obtained in Two High-Speed Engines

opening. With this same dial on the dynamometer, adjusted exactly as when on the road, these same speeds were maintained with the same throttle-opening by varying the number of pounds on the dynamometer. Thus, we determined how much power was being developed at that speed and throttle-opening. At the same time, another set of tests was made that was equally significant. In trying to determine what one-quarter, one-half and three-quarter throttle-openings are, we found that nobody who had made previous tests had stated specifically how they had found what each such opening was with a butterfly throttle; so, we made four runs, taking one-quarter, one-half and three-quarters of the dynamometer load for three runs and then made a wide-open throttle run. That gave constant loads, not throttle-openings, that varied in proportion to the wide-open-throttle load.

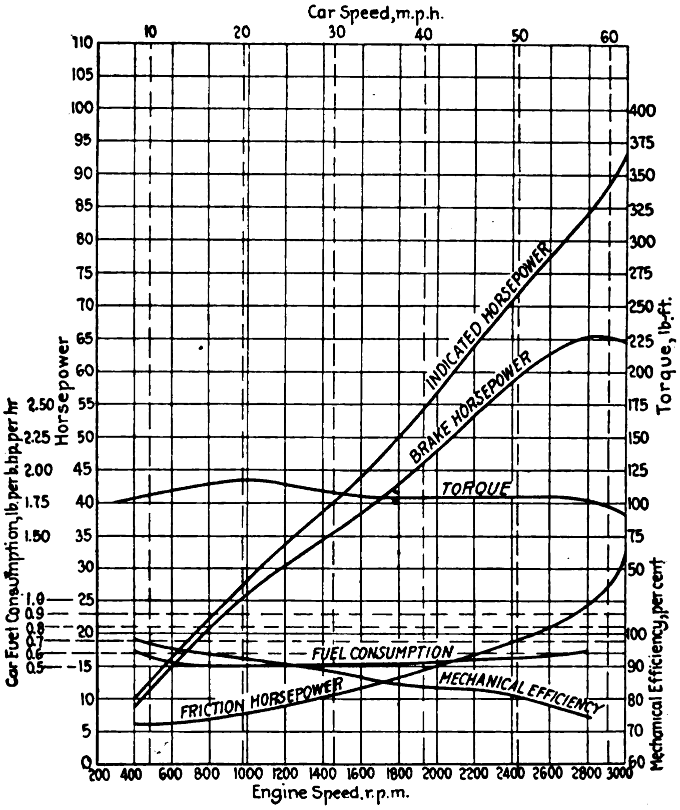


Fig. 6—Performance Characteristics of the Ansted Engine

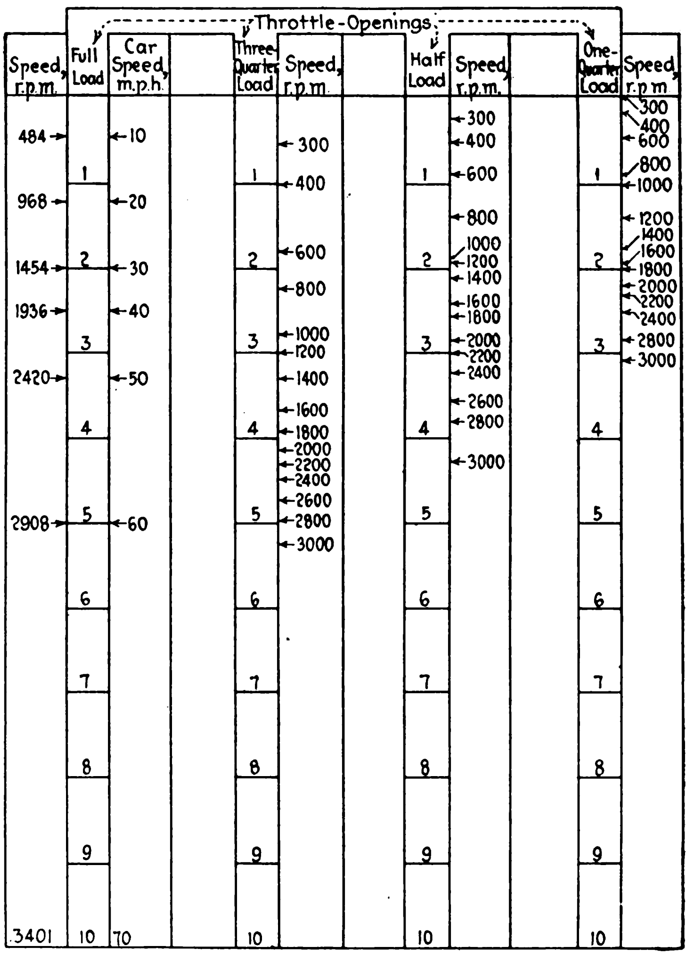


Fig. 7—Diagram Showing the Throttle-Openings at Various Loads

We merely ran the power curve for the maximum brake-horsepower, and then ran a series of other horsepower curves at three-quarters, one-half and one-quarter of whatever the ordinate was. We then determined the throttle-opening at those particular speeds during that run. With one-quarter load, the throttle is never opened more than one-third; with one-half load, it is scarcely open more than four-tenths; and, at three-quarters load, it is only a little more than half-open. One can almost draw a straight line through the same speed-points and throttle-openings.

After the road test, we came in, set the throttle as shown in Fig. 7, brought the revolutions per minute up to the same value and then read the pounds of torque shown by the dynamometer. From that result, we obtained the curve for road horsepower shown in Fig. 8. The abscissas are in terms of hundreds of revolutions and the miles per hour that the car travels. This is determined easily because we used a fixed gear-ratio. About 3 hp. is required at 10 m.p.h. to keep a car rolling steadily on a gravel road; about 7 hp. at 20 m.p.h.; 13 hp. at 30 m.p.h.; 20 hp. at 40 m.p.h.; 30 hp. at 50 m.p.h.; and 43 hp. at 60 m.p.h. Because we could not run the dynamometer at a speed of more than 3200 r.p.m., we did not determine what horsepower was necessary at 3400 r.p.m., which is the speed at which the combined road and wind resistances apparently balance the horsepower of the engine.

The dotted curve in Fig. 8 is the one obtained by Mr. Nelson on the Indianapolis Motor Speedway with a car weighing about 1000 lb. more than the car we used. It is particularly interesting because it shows that either our results were inaccurate or that there is greater resistance on a hard dirt road than that shown on the brick-paved Speedway. This is the first comparison made between the power required on the Speedway and on a dirt road. It is unfortunate that the same car was not used in both instances.

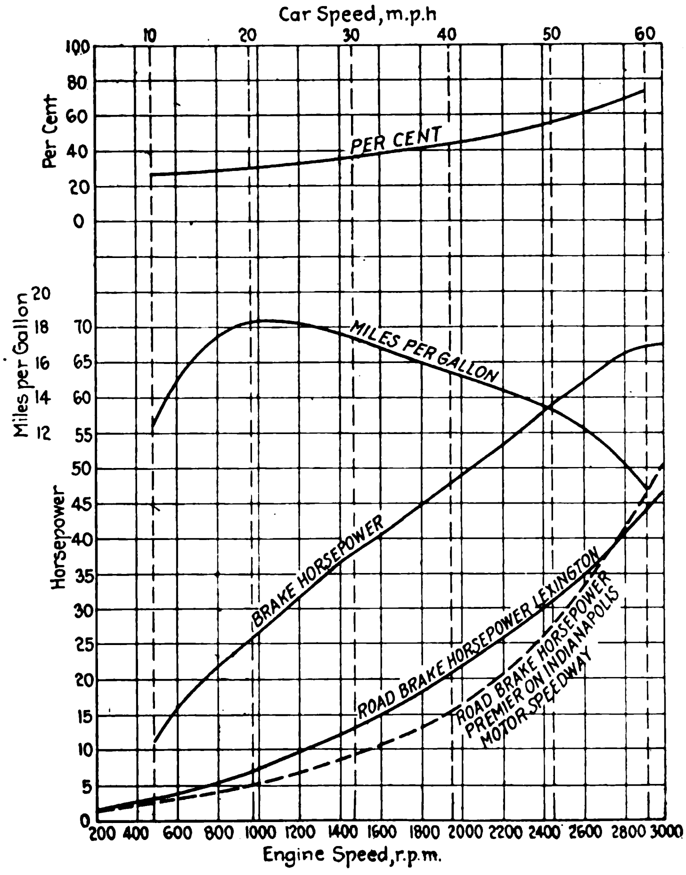


Fig. 8—Road Characteristics of a Car Equipped with the Ansted Engine

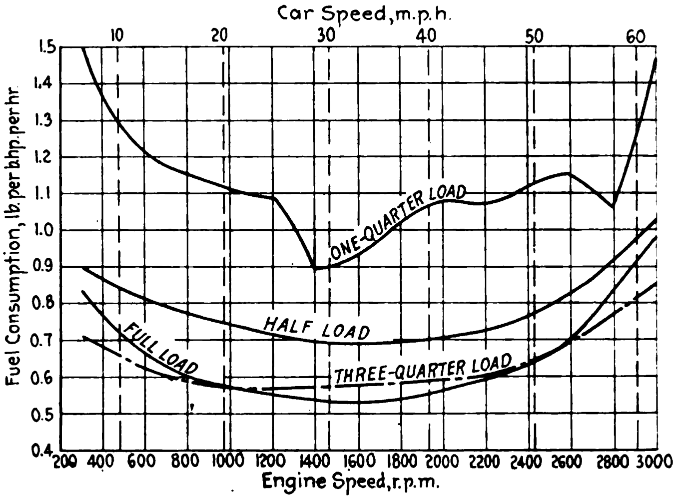


Fig. 9—Fuel-Consumption Curves at Various Engine Loads

The maximum brake-horsepower curve of the Ansted engine is shown in Fig. 8. It is evident that the power available and that required to run on the road are very different. The difference in power is the amount that is available for climbing hills and for acceleration. The percentage curve at the top of Fig. 8 indicates what it is. It is the ratio between the power required on the road and the brake horsepower available. At 10 m.p.h., about 22 per cent of the power is used to drive the car, but at 60 m.p.h. about 64 per cent is required.

An interesting miles-per-gallon curve is shown in this diagram. The values were obtained from the pounds of fuel used per brake horsepower when running to obtain the road curve shown below it. From these computations, we find the values to be 18½ miles per gal. at 20 m.p.h.; about 13 miles per gal. at 10 m.p.h.; about 17 miles per gal. at 30 m.p.h.; 15 miles per gal. at 40 m.p.h.; 13 miles per gal. at 50 m.p.h.; and 8½ miles per gal. at 60 m.p.h. These values are a close check on the average runs made by this car on the road. Driving between 20 and 40 m.p.h. day in and day out, we obtained somewhere between 16 and 17 miles per gal. This is an average that would be indicated by the curve.

Some of the results obtained from the laboratory test when running the engine at full, three-quarters, one-half and one-quarter load are incorporated in Fig. 9. These curves show the pounds of fuel per brake horsepower used at various engine speeds or at equivalent car speeds. The lowest curve shows the fuel consumption obtained with full-throttle. The other three curves are significant. The one-quarter-throttle curve is of such a character that we do not wish to interpret it at this time. We followed the points precisely as we found them on that curve and the other curves fell very close to the points as they were obtained during the test. The one-quarter-load curve shows some very unusual characteristics, indicating that something irregular happens in the carbureter and in the manifold at low-throttle.

The indicated thermal efficiencies are plotted on both the indicated-horsepower and the brake-thermal-efficiency curves in Fig. 10. The former are in full lines and the latter are dotted. They are particularly absorbing to power engineers. We do not realize how much the modern automobile-engine can do. On the indicated thermal efficiency this engine utilizes about 27½ per cent

of the fuel available when valuing the fuel at 19,500 B.t.u. per lb. of gasoline. We were using 59-deg. Baumé gasoline weighing about 6¼ lb. per gal. These curves were computed on that basis. The full-load curve for the thermal efficiency on the brake horsepower runs up to 24 per cent between 1800 and 2000 r.p.m. The curves that are below it are the thermal-efficiency curves at one-quarter, one-half and three-quarter loads. The three-quarter-load thermal-efficiency is almost as good as the full-load thermal-efficiency. The one-half-load curve begins to fall off rapidly and the one-quarter-load curve drops to as low as 14 per cent at the best point; but when we remember that a large powerplant runs at only about 25-per cent thermal-efficiency with everything in its favor, a small 70-hp. gas engine does well to show equally good thermal efficiencies.

For convenience in comparing the curves, percentage curves are also shown. Assuming that the full-load efficiency is 100 per cent, the three-quarter-efficiency curve at a certain point runs from 92 to 96 per cent of the full-load efficiency, and does not drop below 90 per cent under 50 m.p.h. At one-half load it runs down considerably more, from 72 per cent at low speeds up to 81 at 30 m.p.h. and then down to 70 per cent at 60 m.p.h. At one-quarter load, at which a car operates most of the time, this curve shows that, at 10 m.p.h., the operation is at 42½ per cent of full-load efficiency. At 30 to 40 m.p.h. we may obtain as high as 57 per cent of the full-load efficiency, but it drops off rapidly as the speed increases.

Fig. 11 shows a set of curves plotted from readings taken as the experiments progressed. A tube runs into the intake-manifold and is connected to a manometer indicating the inches of mercury depression in the intake-manifold at full load with wide-open throttle. At 3000 r.p.m. there is only a 2-in. depression, and less than ½-in. depression at 300 r.p.m. The unusual characteristics are

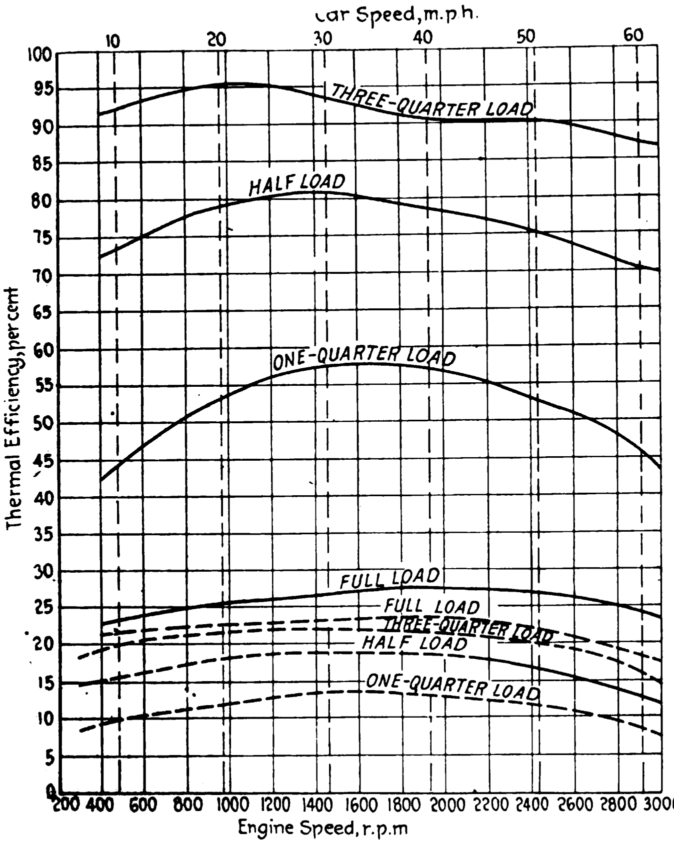


FIG. 10—COMPARISON OF BRAKE-THERMAL EFFICIENCIES AT VARIOUS LOADS

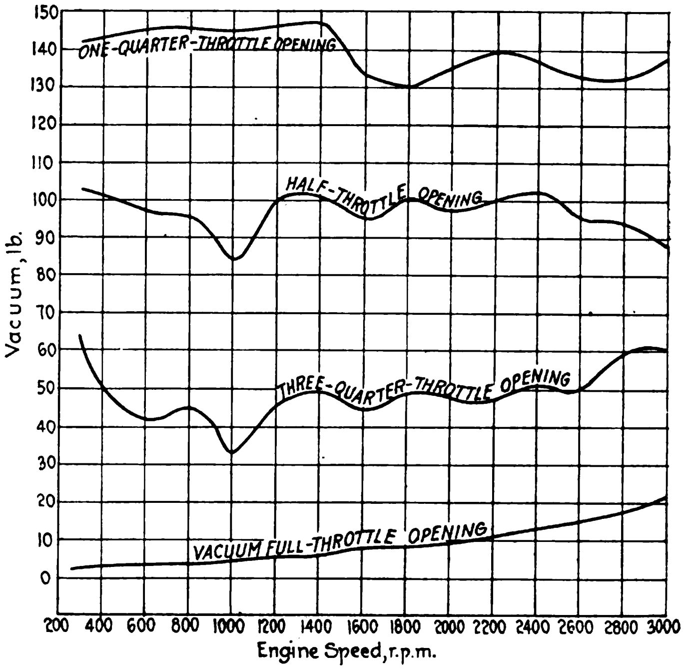


FIG. 11—INTAKE-MANIFOLD DEPRESSIONS AT VARIOUS LOADS

found in three-quarter, one-half and one-quarter-load intake-manifold pressures. The only reason that we followed the points exactly as we obtained them was that there seemed to be this general characteristic of depression in the curve at 1000 and again at 1600 r.p.m.

THE DISCUSSION

ROBERT W. CARINGTON:—What were the temperatures of the intake and exhaust-manifolds?

CHESTER S. RICKER:—We did not take their temperature readings.

G. U. SMITH:—What was the amount of compression in pounds per square inch?

MR. RICKER:—At present, it reaches 80 lb. per sq. in.

PROF. DANIEL ROESCH:—Were readings taken at different speeds so that an idea can be given of what constitutes low, medium and high speed?

MR. RICKER:—No readings were taken at any speed except that at which the starter cranks the engine; that is about 150 r.p.m.

C. H. KIRBY:—What is the peculiarity shown in that depression or vacuum curve?

FREDERICK PURDY:—I am unable to explain the curious curves shown in Fig. 11. What we call a fractional opening of the throttle is not a fractional division of the number of degrees; it is a certain metrical function of the cycle. I believe one-quarter of that, then another quarter and finally four-quarters would give the full opening, and that would give also the uniform area. That would account also for the half-open throttle at 60 m.p.h. and all the remainder of the way, because the increase of area is not very considerable.

When I first saw those curves, I thought that the velocity effect on the exploring tubes must have been the cause of an erratic reading; for, in the readings as they were taken, with a tube extending into the manifold and with the opening at right angles to the axis of the manifold at that point, the static or the true suction-head and also the effect of the velocity or the dynamic head would be shown; that is, if one were taking the minus pressure as a suction value. But on further consideration this seems not to be tenable. As I understand it, there was

no variation in the throttle-opening or in its position, and the only variation was a load on the dynamometer to change the speed; so, it seems that we could not have any such curious shapes as that to atone for. I thought also that the drop in the suction value might have been at a point where the air-valve opened, but this seems to be so pronounced and so uniform that it is a characteristic. There seem to be two drops. With this point somewhat further along and nearer the slow side, it might possibly be accounted for by the point at which the valve began to open. There would be a gradual rise in the suction value as the engine speed increased, and then a sudden drop as the valve broke its position. If that curve were laid out in terms of energy to speed and were a superimposed curve of the suction, there might be some sort of a correspondence there, but the curious shape is a mystery to me.

MR. RICKER:—Concerning air-valves, with a hot-spot "frying grid" such as is used in the manifold on this engine as shown in the cross-section of the head in Fig. 3, the fuel is fed into that hot-spot and vaporized. It is thrown in there by the inertia due to the sudden change in the direction of the gas. The thing to be noted in connection with that type of manifold is that an almost dry gas enters the cylinders under all conditions. The only thing to watch out for is a sudden starving of the engine when the accelerator pedal is pushed down all the way in attempting to get a quick start from low speed.

The next thing that had to be considered was enough port-opening to permit the high engine-speeds that are obtainable. It was essential to get a range of from 300 to 3000 r.p.m. in a stock engine. Without an air-valve type of carbureter it seems to be impossible to obtain a speed range of from 300 to 3000 r.p.m. Because the air column is so much lighter than the fuel column issuing from the jet, the former goes into the engine and leaves the fuel behind when the throttle is kicked wide-open at low speeds. Unless something is done to dampen or slow up that air-column movement for the instant when the throttle is kicked open, one cannot obtain anything like a uniform mixture. That accounts for the good acceleration with this engine from such low speeds as 2 m.p.h. without a flat spot in the curve, and the fact that the carbureter can be adjusted for economy while still giving the fuel-efficiency curves shown, without any readjustment during the tests. I am not saying that this carbureter is the only one that will give such performance but, from my experience with this and other engines, I feel that the air-valve type of carbureter merits very serious consideration for the maximum speed-range type of engine.

ROY E. BERG:—Were gas temperatures recorded during the runs?

MR. RICKER:—No.

MR. KIRBY:—How did you make the determination of gas speeds?

MR. RICKER:—The gas speed is the number of cubic inches of displacement per minute divided by the area of the manifold section in square inches and divided by 12 to reduce it to feet per second. The gas velocities given in Mr. Nelson's paper were figured for one-half the engine displacement for each side of his longitudinal manifold. Using those values here, we obtain about one-half the velocity Mr. Nelson reports.

A MEMBER:—Were these tests made with the radiator in place and the cooling system operating just as on a stock car?

MR. RICKER:—Not on the dynamometer test. The fan was the only part of the system not in operation. We had practically the same resistance as in the radiator. The engine is maintained at a constant temperature by thermostat; so, irrespective of whether it was on the car or on the dynamometer, the temperature of operation was maintained constant.

A MEMBER:—Was the thermostat connected to a pressure system or to a tank?

MR. RICKER:—To a tank.

A MEMBER:—Were those manifold-pressure curves taken at a fixed throttle-opening or at a fixed torque?

MR. RICKER:—The torque varied; the throttle-opening was constant.

DENT PARRETT:—Without offering any criticism of the carbureter, it occurs to me that possibly there is a point in this changing load where a leaner mixture is fed to the engine. For that reason, to pass this horsepower equivalent to a basis of 1000 r.p.m., the manifold pressure would be reduced; in other words, the suction would increase so as to provide a sufficient volume of fuel. It simply means that the carbureter was taking in a little more air in proportion to the fuel, to overcome that point.

MR. SMITH:—What were the ignition curves corresponding to the throttle-openings?

MR. RICKER:—We recorded the data but they have not yet been charted. They varied, however, and we maintained them the same on both the road tests and the dynamometer tests, the spark being advanced the maximum amount permissible without causing "pinging" in the cylinders during the road tests.

JOHN W. STACK:—How were those indicated-horsepower curves plotted? Are they the result of actual observations?

MR. RICKER:—No, the indicated horsepower was taken as the sum of the brake horsepower and the friction horsepower on two different engines, but taken at the same engine speeds.

F. G. SHOEMAKER:—In connection with the curves in Fig. 11, I have observed two things in dynamometer testing. For instance, in taking vacuum readings at partial throttle-openings with and without gasoline, that is, setting to a fixed throttle-opening and turning the mixture on, the pressure in the manifold changes considerably when gasoline is used and when there is no gasoline. For that reason, changes in mixture proportions might account for some of the changes in the curves. Another thing is that we set up resonant periods in intake-manifolds at certain speeds. Might not this be accounted for by resonance in the intake pipe? I have noticed that this has a strong effect in the exhaust-manifold. We had a long 5-in. pipe on the exhaust in our dynamometer equipment that extended about 10 ft. and the horsepower curve dropped off about 2 hp.

MR. PARRETT:—Would not the fact that the dip in the curve occurs at 1000 r.p.m. in both cases be an argument in favor of the idea that this resonance accounts for the effect?

GEORGE E. MARTIN:—Several late designs of engine have shown greater speed and horsepower than earlier engines of the same type. What is the limiting factor of the engine speed and horsepower that can be developed in an engine of a given size? Is it a matter of the amount of fuel that can be put into the engine and burned, or the method of introducing the fuel, or a mechanical characteristic of the engine?

MR. RICKER:—That can be answered by relating some experiences we have had with this engine. Experimental work shows that an increase in valve size or lift gives no further increase in the power. We are trying to determine how much a greater manifold size will do toward

increasing the power. The limitations seem to be almost entirely with the breathing apparatus. If the breathing apparatus were large enough, we might get all the power desired from the cylinders by allowing the engine to turn faster and faster. We may encounter balance and inertia effects that would require the use of some other type of piston, probably a lighter reciprocating part if the speeds were increased beyond this, but we have every reason to believe that if more power is desired there is a possibility of obtaining it by increasing the size of the breathing apparatus without otherwise changing the inherent design of the engine.

Mr. Martin:—Horsepower efficiency is based on the British thermal units in the fuel in steam-engine practice. If a point could be reached where 100 per cent of the British thermal units in the fuel were utilized, would not that be the limiting factor in what might be developed in a gas engine?

Mr. Ricker:—Judging from Diesel-engine practice, we know that the theoretically perfect engine, operating on the constant-pressure, assuming air-standard efficiency, cycle, develops mechanical power that represents only 57 per cent of the available British thermal units in the fuel. Therefore, the Diesel engine, having an efficiency of say 40 per cent or better, is really utilizing 80 per cent or more of the available heat units in the fuel. On that basis, if 27-per cent efficiency is being obtained as shown by the indicated-horsepower curve, we are really obtaining about 55 per cent of the available power in the fuel, and this is far from being an ideal engine performance.

Mr. Purdy:—The very fact that the valley of the curve, not the peak, occurs at a fixed engine speed rather than at a suction value, indicates that resonance is an effect that is responsive to speed and not to suction. The carbureter cannot take cognizance of the engine speed; it only takes cognizance of the suction value. So, if it were due to something within the carbureter or to some change that takes place inside the carbureter, that characteristic drop would not be at the same engine speed.

Mr. Shoemaker:—Perhaps that resonance is in the air-valve of the carbureter, and we can attribute the effect to the carbureter.

Mr. Purdy:—Undoubtedly, but that resonance effect would be responsive to suction rather than to engine speed because, with some particular carbureter, such as the one used in this instance, it is so well damped that resonance in the mechanical part of the carbureter is rather out of the question. Resonance in the moving column of air would be, of course, the same in one type of carbureter as in another, assuming that it was not modified by the resonance or reaction of the mechanical moving parts.

Mr. Kirby:—Were the manometer readings steady at the time?

Mr. Ricker:—They were practically steady.

Mr. Kirby:—Were they uniform throughout the range?

Mr. Ricker:—Yes.

Professor Roesch:—That would cure the resonance effect. Would not the same effect occur at a speed of 2000 r.p.m.?

Mr. Ricker:—There is a slight effect at 1600 r.p.m.

Mr. Parrett:—It may be due to the effect of the pulsations in the suction of the engine combined with the action of the air-valve. At some certain speed the valve would fluctuate more and, if the valve were closed part of the time, that would generate suction and cause this drop in the curve.

Professor Roesch:—Why is that manifold constructed with square ends, as shown?

Mr. Ricker:—It has a hole straight through with the ends closed by core plugs. It is done as a matter of manufacturing convenience to support a core at each end.

Professor Roesch:—Is that the best design and was it determined experimentally?

Mr. Ricker:—It is a matter of manufacturing convenience, but I believe it has considerable merit.

Professor Roesch:—In the application of the friction-horsepower curve in Fig. 6 to the brake-horsepower curve in Fig. 4, was that friction-horsepower curve taken on a stiffer engine?

Mr. Ricker:—Yes, it was taken on a very much stiffer engine.

Professor Roesch:—Then the indicated-horsepower curve would be too sigh in Fig. 4.

Mr. Ricker:—Yes.

Professor Roesch:—In connection with measuring the throttle-openings shown in Fig. 7, were the 10 equal divisions measured on the steering-post?

Mr. Ricker:—No. The measurements were made directly on an indicator attached to the throttle-valve arm, and the sheet was accurately fixed on the carbureter itself. A sheet-metal support was made for it, so there was no question as to its positioning with respect to the center of the throttle-arm. The radius of the points was about 5½ in.; so, there was a fairly small movement of the throttle-valve itself and a very large indication on the dial. The range is slightly more than 90 deg., without any lost motion.

Professor Roesch:—In connection with those small throttle-openings required to drive a car at from 400 to 800 r.p.m. of the engine, the observation of the intake-manifold suction might be checked, because it extends over a rather wide range and is measured more easily than smaller dial ranges.

Activities of the Sections

Secretaries of the Sections

(See facing page for photographs)

BUFFALO SECTION
A. J. Fitzgibbons, 168 Claremont Avenue, Buffalo
CLEVELAND SECTION
E. W. Weaver, 5103 Euclid Avenue, Cleveland
DAYTON SECTION
R. B. May, Dayton Engineering Laboratories, Dayton
DETROIT SECTION
Thomas J. Lítle, Jr., 733 Seyburn Avenue, Detroit
INDIANA SECTION
B. F. Kelly, Weidely Motors Co., Indianapolis
METROPOLITAN SECTION
R. E. Plimpton, 129 East 45th Street, New York City
MID-WEST SECTION
H. O. K. Meister, Hyatt Roller Bearing Co., 2715 South Michigan Avenue, Chicago
MINNEAPOLIS SECTION
Phil N. Overman, 10 South 10th Street, Minneapolis
NEW ENGLAND SECTION
V. A. Nielsen, 701 Beacon Street, Boston
PENNSYLVANIA SECTION
Edward L. Clark, Hunting Park and Rising Sun Avenues, Philadelphia
WASHINGTON SECTION
Benjamin R. Newcomb, 211 Victor Building, City of Washington

A FEW of the Sections have scheduled meetings during September but the active season will not be ushered in until October when vacations have departed along with the warm weather. Reports reaching the Society offices show that the Sections are aiming high in their respective meeting plans for the coming winter. Officers are appreciative of the fact that successful meetings are largely the result of adequate preparation. Speakers are being selected well in advance and given plenty of time to prepare their papers. Authorities are being invited to discuss certain specific points in each paper so that discussion will be premeditated rather than extemporaneous. Meetings are to be appropriately advertised and given prominence in the press before hand. In many cases the papers are to be printed and circulated in advance of the meeting date. All of these steps guarantee better Section meetings and the officers responsible for them are to be congratulated on their good judgment.

SECTION PAPERS

The author who accepts the invitation of a Section to present a paper at one of its meetings has undertaken more than an ordinary obligation. The task of preparing a paper should not be considered lightly and postponed until the last minute. It is not a difficult undertaking if given proper study and sufficient time is devoted to it. Papers written and presented creditably establish an engineer's position in his profession as few other means can, but loosely written or extemporaneous matter undermines one's reputation and leaves an impression of incompetence.

In preparing a paper be sure that you have something to say. This prerequisite calls for familiarity with your subject. Try to present new material, the results of research, unless your audience has little knowledge of the topic you are to treat. Give facts substantiated by tests and avoid qualitative statements. Write an outline of your paper as the first step, arrange the items in proper sequence and then write your thoughts around each item in rough form. Expansion and revision of this first draft will result from several readings of it. Make these revisions as they occur to you and have the manuscript typed. Submit copies of it to men whose experience enables them to offer constructive criticism and suggestion. The Society staff is always glad to assist in this way. Incorporate these suggestions in your draft and undertake the final revision.

Do not be satisfied until your paper is brief and every paragraph pertinent to the subject. Avoid reiteration and needless description. The two criticisms most commonly voiced against engineering papers are that the thought wanders and that they are too long and complex. The scheme of writing an outline corrects these in major part. Brevity can be attained by repeated reading and revision of the manuscript, and the elimination of all but the most important thought.

Schedule of Sections Meetings

SEPTEMBER

16—METROPOLITAN SECTION — Outing to West Point

21—METROPOLITAN SECTION — Effects of Oil-Pumping—G. A. Round—Engine Lubrication Systems—Finley R. Porter

22—NEW ENGLAND SECTION — Electric Starting and Lighting Equipment — Louis B. Ehrlich

29—DETROIT SECTION—Oil Consumption—A. A. Bull

DETROIT SECTION

The first fall meeting of the Detroit Section will be held on the evening of Sept. 29 at the Board of Commerce Building. The meeting will be preceded by the customary supper at 6:30 o'clock. A. A. Bull will review the paper given by him at the White Sulphur Springs meeting on Oil Consumption and arrangements have been made to have a number of engineers who are well versed on the subject discuss this paper with Mr. Bull.

METROPOLITAN SECTION

The Metropolitan Section has selected engine lubrication as the topic for discussion at its meeting on Sept. 21. This meeting will be held at the Automobile Club of America, 247


PHIL OVERMAN


E. W. WEAVER


EDWARD L. CLARK


BENJAMIN H. NEWCOMB


R. B. MAY


THOMAS J. LITLE, JR.


A. J. FITZGIBBONS


B. F. KELLY


H. O. K. MEISTER


V. A. NIELSEN


R. E. PLIMPTON

West 54th Street, New York City, at 8 o'clock. The technical program will be preceded by an informal dinner at 6:30 p. m. George A. Round of the automotive engineering department of the Vacuum Oil Co. will read a paper on the difficulties resulting from crankcase oil dilution. Oil-pumping, excessive carbonization and spark-plug fouling will be treated. Finley R. Porter will present a paper describing the types of engine lubrication systems in general use and summarizing the advantages and disadvantages of each. Other authorities are being asked to contribute their experience and an instructive meeting seems assured.

The Metropolitan Section has arranged a very attractive outing and visit to West Point on Saturday, Sept. 16. A boat has been chartered for the picturesque trip up the Hudson River. This boat leaves 132nd Street and the North River at 10 a. m., daylight saving time,, and it is planned to reach West Point at 1:30 p. m. Lunch will be served en route. The members will be accorded the privilege of inspecting the buildings and grounds of the United States Military Academy during the afternoon. Dinner will be eaten at the West Point Hotel and the return trip started at 7 p. m. Dancing and other entertainment features will add to the enjoyment of the boat trip. Reservations for this outing may be made through the Society office in New York City. Prompt action is suggested as the accommodations are limited.

New England Section

The first fall meeting of the New England Section will be held on the evening of Sept. 22, at the Engineers Club, Boston. Louis B. Ehrlich, chief engineer of Gray & Davis, Inc., will read a paper on Electric Starting and Lighting Equipment. The meeting starts promptly at 8 o'clock and everyone interested in the subject of the evening is invited to attend and take part in the discussion.

COUNCIL ACTIVITIES

THE meeting of the Council held on July 26 at New York City was attended by President Bachman, Vice-Presidents Brautigam and Clark, Councilors Crane and Scott and A. J. Scaife.

The financial report as of June 30 showed a net balance of assets over liabilities of $121,550.32, this being $17,714.58 less than the corresponding figure on the same day of 1921. The income of the Society for the first 9 months of the current fiscal year amounted to $121,891.89. The operating expense during the same period was $138,972.06.

Twenty-eight applications for individual membership and 2 for student enrollment were approved. The following transfers in grade of membership were approved: From Junior to Member, L. M. De Turk, R. L. Spragle; from Associate to Member, B. B. Webb.

It was reported that up to July 25, 1922, 468 applications for membership had been received, as compared with 504 received during the first 7 months of 1921, and 754 during the first 7 months of 1920. On June 30 there were 5543 names on the rolls of the Society, including affiliate member representatives and enrolled students, as compared with 5494 on the same day of 1921, and 4781 on the same day of 1920.

The following appointments to the Standards Committee were made:

Nomenclature Division—W. P. Culver

Truck and Transmission Divisions—Walter M. Petty.

H. E. Brunner was named as a representative of the Society on the Sectional Committee on Ball Bearings.

The September meeting of the Council has been scheduled tentatively to be held in New York City on the 19th.

OBITUARY

Horace Elmer Rice, assistant to the president of the American Bosch Magneto Corporation, Springfield, Mass., was killed Aug. 4, 1922, when his automobile was struck by a Pennsylvania Railroad train at a grade crossing in Oxford, Pa. At the time of the accident Mr. Rice, who was 39 years old, was on a trip that was to have included the principal cities of the West.

He was born June 4, 1883, at Philadelphia. After completing a high school course, he entered the service of the Consolidated Fire Alarm Co. at Philadelphia in 1898. In 1903 he accepted a position in the laboratory of the Philadelphia Electric Co. as a special designer and after remaining there for 3 years went with the Rowland Telegraphic Co., Baltimore, as its chief electrical draftsman. After remaining there for 2 years he became sales engineer for W. P. Dallet of Philadelphia. In 1910 Mr. Rice joined the forces of the Atwater Kent Mfg. Co. and remained there 3 years, when he accepted the position of sales manager with the Schoen-Jackson Co., Media, Pa. In 1914 he returned to the Atwater Kent organization as sales manager and a year later took charge of the advertising work in addition to his duties as sales manager. He remained with this company until last March when he went to Springfield, Mass., to become associated with the American Bosch Magneto Corporation.

Mr. Rice was elected to Associate Member grade in the Society Sept. 8, 1915.

Current Standardization Work

Tentative Fall Schedule of Division Meetings

Division	Cleveland	Chicago	Detroit	New York City	Rochester
Agricultural Power Equipment		Nov. 7[a]			
Axle and Wheels	Sept. 20[b]				
Ball and Roller Bearings	Sept. 20[b]				
Chain	Oct. 25[c]				
Electric Vehicle			Nov. 3		
Electrical Equipment					Oct. 2[d]
Engine		Nov. 6			
Frames			Sept. 25		
Iron and Steel				Oct. 26	
Lighting	Sept. 19				
Lubricants	Sept. 21				
Motorboat				Sept. 28	
Parts and Fittings			Oct. 30		
Passenger Car			Nov. 1		
Passenger-Car Body			Nov. 2		
Springs	Oct. 24				
Stationary Engine		Nov. 7[a]			
Storage Battery				Sept. 15	
Transmission			Oct. 31		
Truck			Sept. 22		

[a]Joint meeting of Agricultural Power and Stationary Engine Divisions.
[b]Joint meeting of Axle and Wheels and Ball and Roller Bearings Divisions.
[c]Joint meeting with Power Transmission Chain Committee of American Society of Mechanical Engineers.
[d]Joint meeting with Automotive Electric Association.

THE Truck Division held a meeting on July 24 to consider the important subject of motor-truck rating. Other subjects were discussed and acted upon as recorded below. The Electrical Equipment Subdivision on Spark-Plugs held a meeting on Aug. 18. The Subcommittee on Landing-Fields of the Sectional Committee on Aeronautical Safety Code held a meeting on July 28.

Two series of meetings, as outlined in the accompanying table, have been scheduled for the months preceding the Annual Standards Committee meeting in January. The dates are subject to change, but Division and Subdivision members should bear them in mind in connection with any standardization work they are carrying on. In all cases individual notices definitely specifying the time and place of each meeting will be mailed to committeemen from 10 days to 2 weeks in advance of the sessions. If meetings scheduled tentatively are cancelled or postponed, definite advices to this effect will be transmitted to them.

Aeronautic Safety Code

A meeting of the Subdivision on Landing-Fields of the Sectional Committee on Aeronautic Safety Code was held in New York City on July 28. The original draft of the code covering this section was revised and will be reissued shortly for comment.

Body Hold-Down Clamps

A discussion on the standardization of body hold-down clamps at the Motor-Truck Division meeting in July brought out the opinion that this subject is of more interest to body-builders than to truck producers inasmuch as the former usually make their own clamps. The following recommendation was, however, approved for adoption as S. A. E. Recommended Practice for the guidance of the smaller body builders:

> The top or bottom flange of motor-truck frames should not be drilled for body or hoist platform hold-down clamps. "U" clamps should be used with a wood-block filler between the frame flanges to prevent bending, similar to the construction shown in Fig. 1.

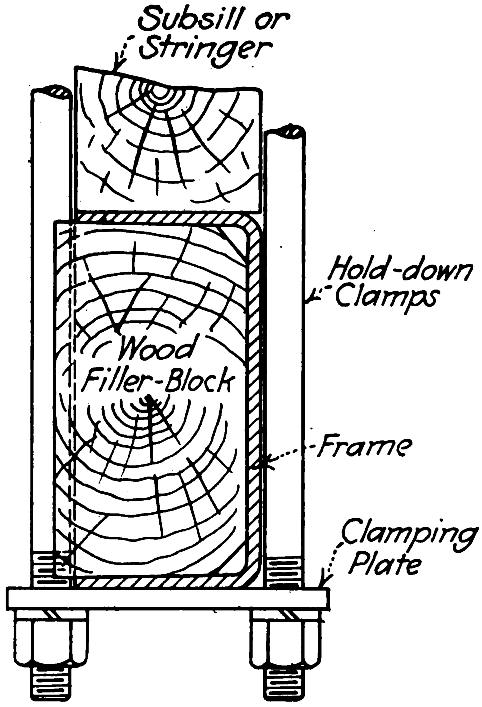


Fig. 1—Recommended Construction of Body Hold-Down Clamps

The use of too many hold-down clamps for securing the body to the frame should be guarded against, particularly in the mounting of very stiff bodies such as those for oil-tanks.

Gages and Gaging

The report of the Screw-Threads Division on Gages and Gaging was not approved at the meeting of the Standards

Committee on June 20 at White Sulphur Springs, as reported in the August issue of THE JOURNAL, because it was felt that the report did not deal sufficiently with gaging for lead error. Earle Buckingham, who formulated the original report, has submitted the following paragraphs covering gaging for lead error as an extension of the report. These paragraphs have been referred to the members of the Screw-Threads Division for approval.

> A screw-thread is comprised of several elements. First, the outside or major diameter; second, the pitch diameter; third, the core or minor diameter; fourth, the angle of the thread form; and fifth, the lead. There is a broad general principle in regard to limit gages that should always be kept in mind. Where compound tolerances are not involved, a "Go" gage with fixed measuring surfaces may check as many dimensions at one time as desired, and effective inspection will be secured. On the other hand, an effective "Not Go" gage can check only one dimension. By effective inspection is meant assurance that specified requirements in regard to size are not exceeded.
>
> The most difficult element of a screw-thread to gage is the lead. Lead-testing devices for checking tools and gages are available, but in general their operation is too slow for use as production inspection equipment. In addition, the lead is the most important element of a screw-thread as regards the nature of the contact between mating parts. Furthermore, an error in lead has almost double the effect of an equal error in diameter as regards interchangeability. Thus, for exacting threaded work, if the method of inspection of the parts produced does not effectively inspect for lead errors, the tools used to produce these parts must be carefully inspected for lead.

Gasoline Railroad Cars

The standardization of gasoline railroad cars was discussed at the July Truck Division Meeting in consequence of a suggestion that had been received by the Society that a new Division of the Standards Committee be established to work on the standardization of this type of vehicle. It was indicated that the conventional truck chassis will be the basis for the construction of gasoline railroad cars, rather than gasoline powerplant equipment being adapted to typical railroad-car construction.

There was a considerable discussion as to whether railroad or automotive engineers should carry on such standardization work, general opinion indicating that the automotive engineers should do so; also that, if the work is undertaken, a separate Division should be established.

Molybdenum Steels

The standardization of definite chemical analyses for molybdenum steels has been referred to the members of the Iron and Steel Division at the request of an extensive user of molybdenum steel. Definite action in regard to this matter will be taken at the next meeting of the Division. Up to the present time the Iron and Steel Division has recommended only that the numeral "4" be used as the index to the molybdenum steel specification numbers. No specific chemical compositions have been approved.

Motorbus Bodies

It was suggested at the last Truck Division meeting that the standardization of mounting dimensions for motorbus bodies be studied, but it was thought that it is too early in the development of this type of vehicle to undertake such standardization.

Motor-Truck Cabs

At the July meeting of the Truck Division H. B. Knap, of the Packard Motor Car Co., was appointed a Subdivision of one to prepare a report on the standardization of motor-truck cabs. Tabulated dimensions showing present practice as to cab construction was turned over to Mr. Knap for use in this connection.

Motor-Truck Rating

In response to demands from the highway authorities of the State of Connecticut, a meeting was held at Detroit on July 24 to discuss the formulation of a "yardstick" of gross carrying-capacity and safe operation of motor-trucks for administrative use by licensing and law-enforcement officials that can be definitely determined by the manufacturer, buyer and the law-enforcement official. This meeting was attended by members of the Truck Division, a representative of the National Automobile Chamber of Commerce and representatives of various truck builders.

It was felt that there is a definite need for a rating and that the essential elements to be considered are the strength and ability of steering-gears and of brakes and the strength of the axles. It is considered that these three factors are the important ones in determining the safety ability of a truck on the road and that all are equally important. A subdivision, consisting of A. K. Brumbaugh, chairman, D. C. Fenner and A. J. Scaife, was appointed to submit the suggestions of the meeting to the manufacturers for their consideration and comment.

Spark-Plugs

A meeting of the members of the Electrical Equipment Subdivision on Spark-Plugs was held in Detroit on Aug. 18 at the plant of the Packard Motor Car Co. The present S. A. E. Standards for Spark-Plugs, pp. A10, A11 and A12 of the S. A. E. HANDBOOK, were reviewed and revisions necessary to bring them into accord with the best spark-plug practice of today recommended. The recommendations will be submitted to engine manufacturers and users for comment before action is taken by the Electrical Equipment Division. The personnel of the Spark-Plug Subdivision is O. C. Rohde, chairman, Champion Spark Plug Co.; B. de Guichard, A. C. Spark Plug Co.; A. D. T. Libby, Splitdorf Electrical Co.; C. S. Price, Bethlehem Spark Plug Co.; D. L. Arnold, International Harvester Co.; L. M. Woolson, Packard Motor Car Co., M. J. Steele, Packard Motor Car Co., and S. F. Evelyn, Continental Motor Corporation

Spring Shackle-Bolts

The standardization work on spring shackle-bolts has been discontinued by the Parts and Fittings Division in view of the fact that the comments received from passenger-car builders indicate that the type of spring shackle-bolts proposed is not satisfactory, many special designs being used by various companies.

Tractor Ratings

O. W. Sjogren, a Subdivision of one appointed to formulate a report on Tractor Ratings, has transmitted the following report of the Subcommittee on Tractor Ratings of the American Society of Agricultural Engineers, for the consideration of the Agricultural Power Equipment Division:

TRACTOR RATINGS

> THE object of this report is to recommend a standard method or code for rating tractors, by which either new or old models can be accurately rated. The rating is to cover (*a*) brake horsepower delivered to the pulley on the machine driven and (*b*) horsepower delivered to the drawbar at plowing speeds.
>
> *Brake-Horsepower Rating*
>
> The following tests shall be conducted on each tractor selected by the board of engineers in charge of the tests:
>
> 1—"Limbering-Up" Test
>
> The object of this test is to eliminate the stiffness likely to be found in a new machine. In this run tractors will be used to pull harrows, packers, or anything that will furnish a suitable drawbar-load.
>
> The loads to be pulled will be (*a*) approximately one-third load for 4 hr., (*b*) approximately two-thirds load for 4 hr., and (*c*) approximately full load for 4 hr.
>
> If the tractor manufacturer believes that a 12-hr.

run is not-sufficient to limber-up the tractor, reasonable additional time will be allowed.

2—Brake-Horsepower Test at Maximum Load

The object of this test is to determine the greatest load the tractor engine will carry on the belt with the governor set for rated speed and the carbureter set for maximum power. The brake-load will be increased until the horsepower developed is the greatest.

The rated speed is to be considered the speed of the engine under load.

This test should begin after the temperature of the cooling fluid has become constant.

The duration of the test shall be 1 hr. without interruption or change in load or tractor adjustment.

If the speed should change during the test enough to indicate that conditions had not become constant when the test was started, the test will be repeated with the necessary change in load.

3—Brake-Horsepower Test at 80 per cent of the Maximum Load Recorded in Test 2

The object of this test is to show whether the engine will carry continuously its rated load on the belt and to show the fuel-consumption at the rated load. The governor is to be set to run the engine at rated speed.

The test will commence after the temperature of the cooling fluid has become constant.

The duration of the test will be 2 hr. without interruption or change in load or engine adjustments.

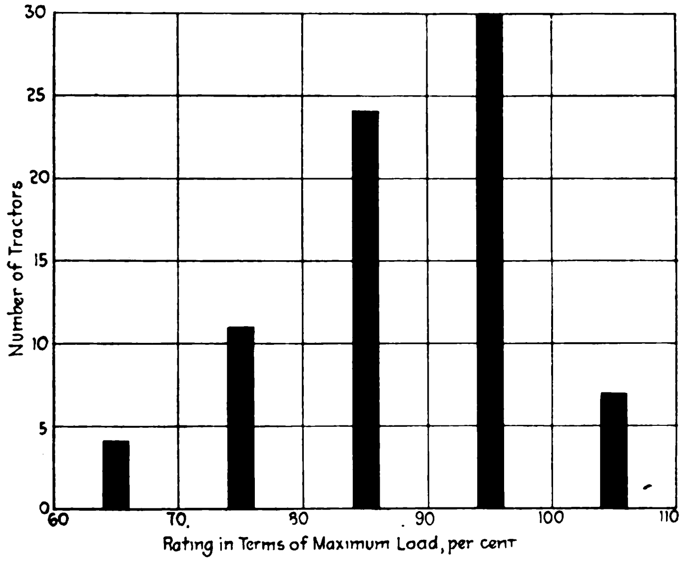


Fig. 2—Tractors Classified in Groups Showing the Rated Belt-Horsepower Expressed as a Percentage of the Maximum

After tests 2 and 3 shall have been conducted to the satisfaction of the board of engineers in charge of the tests, the nearest whole number to the load carried in test 3 is to be considered the brake-horsepower rating of the tractor.

4—Brake-Horsepower Test at Varying Load

The object of this test is to show fuel-consumption and governor control when the load varies.

All adjustments are as in test 3.

The time and the load are as follows:

- 10 min. at rated load or load carried in test 3
- 10 min. at maximum load
- 10 min. at no load
- 10 min. at one-fourth rated-load
- 10 min. at one-half rated-load
- 10 min. at three-fourths rated-load

The total running time is 1 hr. and the test is to be conducted without stopping the engine or changing its adjustments. If load-changes make readjustments necessary, the final report of the test shall state the fact.

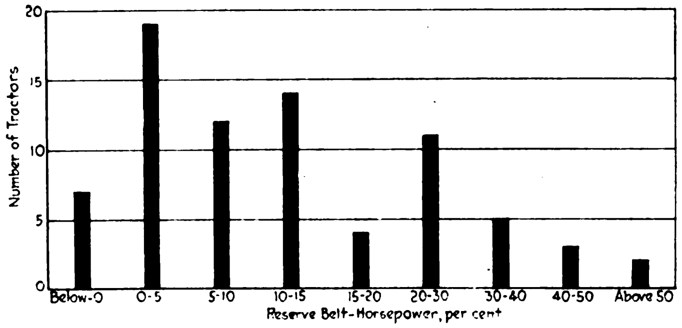


Fig. 3—Tractors Classified to Show the Percentage of Reserve Belt-Horsepower

The variation in speed from the rated speed shall not be more than 10 per cent.

5—Nebraska Results May Be Accepted

In lieu of tests 1, 2, 3 and 4, the board of engineers may accept the results of similar tests on the same model of tractor conducted by and under the rules of the State of Nebraska for tractor tests. The brake-horsepower rating, however, is to be only 80 per cent of the maximum horsepower determined by the Nebraska tractor tests.

Method of Testing

It is recommended by the American Society of Agricultural Engineers that the brake test outlined above be conducted by one or more disinterested engineers who are full Members of one of the following societies:

American Society of Agricultural Engineers
American Society of Mechanical Engineers
Society of Automotive Engineers, Inc.

However, members engineering faculties of State agricultural colleges or universities are eligible to serve as members of the board of engineers.

The engineer or engineers conducting the tests must first be approved by the Council of the American Society of Agricultural Engineers.

Reports of the tests must be signed by all of the engineers conducting them and the signed reports sent to the secretary of the National Association of Farm-Equipment Manufacturers, to be placed on file subject to inspection by anyone interested in them. Duplicates are to be furnished the secretary of the American Society of Agricultural Engineers and the manufacturers of the tractors tested.

These tests are to be conducted on one or more tractors picked at random from the stock run of tractors from the factory by the disinterested engineers conducting the tests. If more than one tractor is tested,

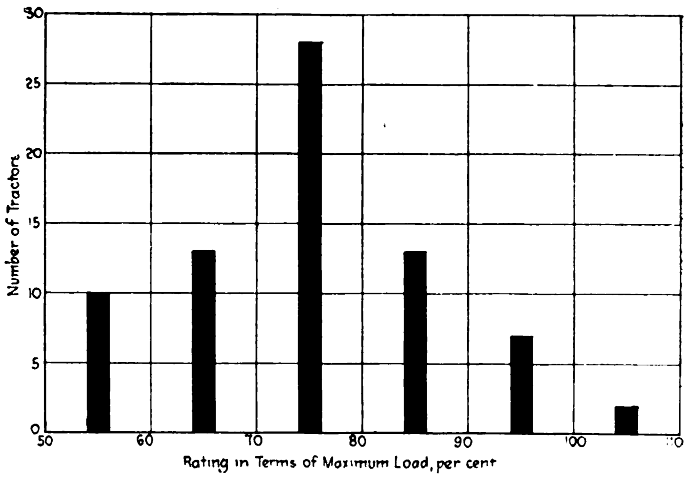


Fig. 4—Tractors Classified in Groups Showing the Rated Drawbar-Horsepower Expressed as a Percentage of the Maximum

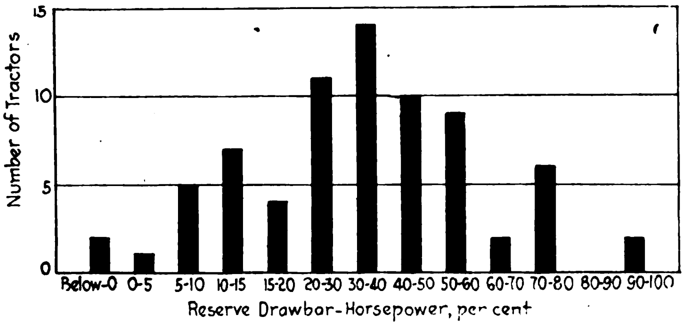


FIG. 5—TRACTORS GROUPED TO SHOW THE PERCENTAGE OF RESERVE DRAWBAR-HORSEPOWER

the averages of all tests shall be used in determining the brake-horsepower rating.

All tests will be made on the *lowest grade* of fuel that the tractor manufacturer recommends for the respective tractors, these fuels to be purchased in the open market and to consist of the low grades of either kerosene or gasoline sold in the locality. All fuels to be tested by the board of engineers.

Manufacturers must specify the kinds and grades of lubricant to be used in the different parts of the tractors tested.

The tractor manufacturers shall furnish an ample number of trained operators with their tractors while they are being tested.

The manufacturer will be required to furnish operators during the entire series of tests, the board of engineers to act in a supervisory capacity and to take all readings and compile the results.

All belt tests must be made with an electric dynamometer, an accurately tested Prony-brake, or other power-measuring device approved by the board of engineers. If a Prony-brake is used, allowance must be made in the results for the power needed to drive it.

The quantity of fuel used in each part of the test shall be determined by weight and the quantity reduced to gallons at 60 deg. fahr. For brake tests a tank will be placed on a scale and set at the same height as the tank on the tractor. Fuel will be drawn from this tank on the scale during the tests. The fuel-tank shall contain at least 2 gal. of fuel during every part of the tests.

The quantity of oil used is to be determined by the standard gallon, quart, or pint measure, or by weight if more convenient.

The quantity of water used in the radiator and cooling system is to be determined by measuring the height of the water at the beginning of the test and filling to the same level at the end of the test, weighing or measuring the water added. If necessary to secure accurate results, the water added will be heated to the same temperature as the water in the radiator or tank.

The quantity of water used in the carbureter, where such water is used for fuel purposes, is to be determined by weight. A tank is to be placed on a scale and set at the same height as the tank on the tractor. Water will be drawn from this tank on the scale during the test. The water-tank shall contain at least 2 gal. of water during any part of that test.

These tests may be conducted at any place that meets the approval of both the manufacturer and the board of engineers conducting them.

Drawbar Rating

With respect to the drawbar rating, the manufacturer shall be given the choice of either of the following two methods:

(a) The drawbar rating to be 60 per cent of the brake-horsepower rating determined by Test 3.

(b) A maximum drawbar-horsepower test conducted under the rules of the Nebraska tractor tests. The drawbar rating is to be 80 per cent of the maximum developed in this test.

In submitting this report to the Division, Mr. Sjogren stated in amplification that

Experience in Nebraska during the past 2 seasons in the testing of more than 80 different tractors indicates that there is too great a variation in the rating of tractors. In fact, there is no standard method by which the tractors can be rated as to the power output, and in some cases the rating is greater than can actually be delivered on test. In some instances the horsepower of the engine is given as that developed when directly connected to the dynamometer, the engine not operating the fan or water-pump. In other cases the tractor is given a very liberal rating based on what it can actually do when tested on the belt, sometimes being rated at only 50 per cent of what it can actually develop.

These are the two extremes and, in the interest of both the manufacturers and the users of tractors, a standard method should be adopted for the rating of tractors. A discussion of the results obtained in the testing of 68 tractors during the season of 1920 was given at the 1921 Farm Power Meeting of the Society at Columbus, Ohio, and published in the May, 1921, issue of THE JOURNAL. Fig. 2 of that discussion[1] indicated that the majority of the tractors tested were rated at more than 80 per cent of the maximum horsepower delivered on the test and that some were rated at actually more than they could deliver. Charts based on all the tests conducted to date are submitted herewith in Figs. 2 to 6.

The formula method of rating tractors is entirely impractical, especially if based on piston displacement. As proof of this contention, reference is made to Fig. 3 of my paper which was printed in the May 1921 issue of THE JOURNAL[1] and to the chart herewith showing the relation of piston displacement to engine weight per maximum brake-horsepower which is reproduced in Fig. 6.

No argument need be presented to indicate the advantages that would be derived by all concerned from a standard method of rating.

I feel that there are two things that the Agricultural Power Equipment Division can do at this time. One is to specify a standard method of rating tractors. The other is to specify a standard testing-code, so that, no matter where these tests are carried out, they will be carried out on a uniform basis, and therefore ratings based on such tests will be uniform.

[1] See THE JOURNAL, May 1921, p. 392.

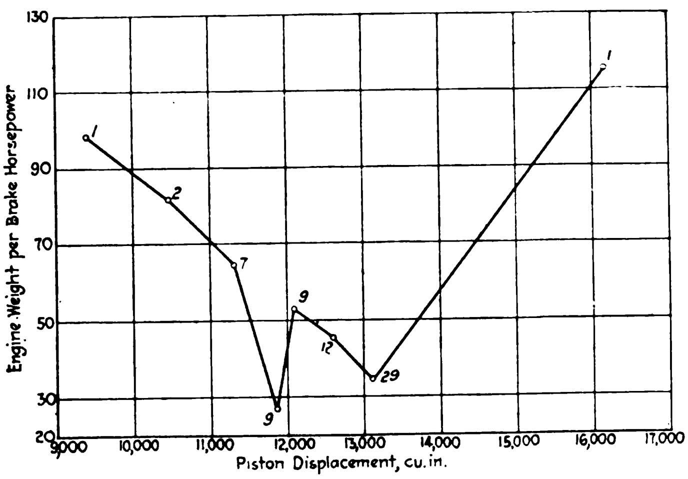


FIG. 6—RELATION BETWEEN PISTON DISPLACEMENT AND THE ENGINE WEIGHT PER MAXIMUM BRAKE-HORSEPOWER

Publications of Interest *to* S. A. E. Members

In this column are given brief items regarding technical books and publications on automotive subjects. As a general rule, no attempt is made to give an exhaustive review of the books, the purpose of this section of THE JOURNAL being rather to indicate from time to time what literature relating to the automotive industry has been published with a short statement of the contents.

DESIGN AND FUNCTIONING OF LAMINATED AUTOMOBILE SUSPENSION SPRING. By A. A. Remington. Paper read before the Institution of Automobile Engineers, 28 Victoria Street, Westminster, S. W., London, England.

The paper contains information and data of considerable value. Mr. Remington says in part:

It would appear that the laminated spring as used at present in automobile suspension is capable of considerable development, particularly in the direction of economy of material, resulting in reduced weight and cost. The suspension is the part of a motor car which has undergone the least change; in fact, the development has been practically nil. A semi-elliptic spring is a special case of a beam. Assuming it to be loaded in the center and supported at the ends, the condition of maximum economy is fulfilled when all the material is uniformly stressed, which in a beam occurs when the resisting moment is equal to the bending moment on all sections. E. Phillips, a French engineer whose researches were published in 1852, showed that a rhomboidal beam can be cut longitudinally into any number of pieces superimposed to form a laminated spring without effect on its action as a beam or spring. A laminated spring so formed is uniformly stressed throughout and, if made with circular curvature, the curvature under deflection will, due to uniformity of stress on all sections, remain circular.

The reliability of the spring must not be judged by the amount of metal in it, as this may only be a measure of its inefficiency, and if the stress distribution is irregular the surplus metal may actually reduce the reliability by causing local overstressing. The behavior of a spring as a component part of a suspension depends upon a number of external factors, all of which affect the smoothness of riding of the vehicle. To avoid temporary separation between the vehicle and the load, in the case of a passenger vehicle the passengers involuntarily leaving their seats, it is necessary to insure that the vertical acceleration of that portion of the vehicle carrying the load shall never exceed that due to gravity. The vertical acceleration depends upon the periodicity and the amplitude of the suspension.

The rate of oscillation of a spring-suspended system is dependent on gravity and follows the same general laws as the pendulum and the governor. The periodicity of a suspension has to be considered as a whole, as all portions act in unison to produce a resultant, and the separate components cannot oscillate independently of each other. In the case of a motor car these can be divided into

(1) The front-suspension vertical period
(2) The rear-suspension vertical period
(3) The rolling period
(4) The pitching period

The rolling period and the pitching period both result from the combined action of all the springs, front and rear. The former is largely influenced by the height of the center of gravity above the suspension level, and is usually considerably slower than either of the vertical periods, while the latter depends on wheelbase and weight distribution longitudinally. The difference between the rolling and the rear vertical periods and the small practical range of periodicity constitute the principal disadvantages of normal suspension systems and indicate the most promising directions in which to seek for improvement.

The effect of friction between the leaves is to damp out the oscillations gradually. Laminated springs present considerable resistance to torsion, especially when under load, due to the lessened tendency for the leaves to separate and this resistance, on account of the methods of mounting adopted, stiffens the usual forms of suspension in their resistance to rolling.

The following tabulation gives a very interesting comparison of practical steel stresses as used in Great Britain and in the United States:

	Great Britain	United States
Minimum Tensile Strength, lb. per sq. in.		
Carbon Steel	179,200	185,000
Alloy-Steel	201,600	200,000 210,000
Maximum Safe Stress, lb. per sq. in.		
Carbon-Steel	112,000	130,000
Alloy-Steel	145,600	165,000 170,000
Maximum Permissible Normal Static Stress, lb. per sq. in.		
Carbon-Steel Front Springs	44,800	40,000
Carbon-Steel Rear Springs	67,200	50,000 56,000
Alloy-Steel Front Springs	53,760	45,000
Alloy-Steel Rear Springs	80,640	54,000 64,000

In all his calculations for stress, flexibility and deflections Mr. Remington uses empirical formulas similar to those used by spring engineers in this Country. His theories pertaining to economic design, uniformity of stress and leaves formed to true radii are correct and are used by a few engineers in this Country.—S. P. H.

CARBON MONOXIDE POISONING IN CLOSED GARAGES. Reprint No. 694 from the Public Health Reports published by the Government Printing Office, City of Washington. 8 pp.

Automobile owners and garage workers should be warned of the danger involved in running a gasoline engine in a small closed space and advised to see that the garage is well ventilated before permitting an engine to run for any considerable period of time. The principal toxic substance in the exhaust gas of gasoline engines is carbon monoxide, which quickly overcomes persons exposed to it above certain concentrations. Some interesting experiments on this subject have recently been carried on in connection with a preliminary study of the problem of ventilation involved in the proposed vehicular tunnel under the Hudson River. These experiments were made in especially prepared gassing chambers and related principally to the length of time it is safe to be exposed to various concentrations of carbon monoxide and the amount of carbon monoxide given off in the exhaust of automobile engines. Human beings, horses and dogs were used as experimental subjects.

A small automobile, rated at 23 hp., was used and it appeared that approximately 1.5 cu. ft. of carbon monoxide per min. was produced by the car. If a car while warming up should give off 1 cu. ft. of carbon monoxide per min. in a closed room 10 x 10 x 20 ft., the atmosphere would reach the dangerous concentration of 15 parts in 10,000 in 3 min.

MATHEMATICAL THEORY OF INDUCED VOLTAGE IN THE HIGH-TENSION MAGNETO. By Francis B. Silsbee. Bureau of Standards Scientific Paper No. 424. 60 pp.

One of the problems that has received attention from the National Advisory Committee for Aeronautics is that of the high-tension magneto, and a detailed study of this device has been carried on at the Bureau of Standards under its

auspices. While in the past the magneto has been developed along empirical lines, there has recently appeared a tendency to place the design of this type of apparatus upon a more definite scientific basis, and as a foundation for such a rational design at least an approximate theory of the internal phenomena that take place within it must be available. The present paper develops from several points of view such an approximate theory for one particular period in the cycle of operation and correlates certain of the theoretical conclusions with a few experimental results.

The magneto is an exceedingly complex electrical system that serves to transform energy from mechanical through magnetic and electrical forms into heat energy in a high-voltage spark. It is probably impossible to give a complete mathematical treatment that will permit of the exact computation of all the phenomena which occur in such a device. Equations can, however, be developed for various combinations of circuits that approximate more or less closely to the actual magneto, and these abstract theoretical circuits will be referred to in the following pages as "models" of the magneto. Such models are useful for two distinct purposes: (a) the change in the performance of the model under various conditions gives a qualitative indication of the corresponding behavior of the actual magneto under similar conditions and (b) a quantitative expression for any specific property of the magneto, such, for example, as the effect of eddy currents or the magnitude of the secondary capacity, may be expressed by the numerical value of the corresponding quantity in the model, and changes made upon such properties as a result of changes in the design can thus be numerically expressed and compared.

Three different circuits, representing in simplified form the essential features of the high-tension magneto, are developed and equations for the electrical performance of each are given. It is shown that by the insertion of proper electrical constants in these equations the resulting performance will be substantially the same as that of an actual magneto. Methods are suggested for the experimental determination of these constants, and the agreement between this theory and the observed results is shown in certain cases.

While this article is a highly technical one, it presents the results of a careful mathematical analysis of the performance of the high-tension magneto in a form that should be readily available to those who are familiar with the problem of magneto design. The material presented is for the most part original and should be of great value to those interested in the subject.

HYDROSTATIC TEST OF AN AIRSHIP MODEL. National Advisory Committee for Aeronautics. Technical Note No. 87. 15 pp.; illustrated.

The report describes an ingenious test on a Goodyear airship model in which the model was filled with water and suspended from a beam, and the deformations of the envelope studied under varied conditions. A study was made to determine the minimum head of water necessary to maintain the longitudinal axis of the envelope under these conditions. The effect of filling with water on the length of the model was also noted. Photographs of the model recording the deflections under each of the varied conditions are given, as well as those taken of the model filled with air and after filling with water, before any adjustment of the suspension was made.

ENGINE LUBRICATION. By E. L. Bass. Paper read before the Institute of Automotive Engineers, 28 Victoria Street, Westminster, S. W., London, England. 67 pp.

The author first engages in a short general discussion of the development of internal-combustion engine lubrication and points out the several respects in which improvement is highly desirable. The chief problem is that of piston lubrication, and the author discusses this subject in some detail and offers helpful suggestions as to piston design, choice and application of lubricants, especially those containing colloidal graphite. The discussion of piston lubrication is amplified by an abridged discussion of the problems of lubrication of the other working parts of the engine, and the generally accepted principles of bearing design and methods of coping with the difficulties that are presented. A short critical survey of the three types of lubrication systems, their advantages and drawbacks, and the various devices employed by these systems is also given. The subject of crankcase dilution is treated in more detail, and the author emphasizes the need for a thorough laboratory examination of the various lubricants to determine their characteristics with respect to their tendencies to absorb fuel fractions, lose viscosity with dilution, etc.

In the several appendices to the paper the author gives some data, useful in the design and construction of bearings. However, several of his statements are subject to correction. For example, in the third appendix he states that "The friction of liquids varies as the square, not as the square root of the speed; hence the friction of a well lubricated bearing is not merely that of the lubricant." This is obviously not the case, for when a bearing is operated under the conditions of "perfect" lubrication, the surfaces are separated by a *thin* film of lubricant moving substantially in *straight-line* flow. The friction, therefore, should theoretically, and does actually, vary directly as the first power of the speed, and also proportionally with viscosity of the liquid in the film. The apparently slow increase of friction with the speed observed in many cases is simply due to a failure to correct for the decreased viscosity of the liquid in the film, due to higher temperatures at the higher speeds. Contrary to Bass's conclusion, the friction at high speeds *can* and *has* been shown to be due entirely to friction of the fluid in the film. In Appendix 6 the author gives a table of safe carrying powers for various lubricants. The pressures quoted can mean little, as the maximum allowable loading of a bearing depends on the viscosity and the speed of rubbing, both of which the author fails to specify for the values quoted. The final appendix consists of a very useful bibliography of treatises on the various phases of lubrication.

As the ground covered by the author could easily furnish sufficient subject matter for a book of considerable size, this paper, even though it covers 67 pages including the appendices, is necessarily incomplete in many details, especially as regards American practice. It is nevertheless a very useful discussion for those concerned with the practical side of engine lubrication.—D. P. B. 4th.

METHODS FOR TESTING PETROLEUM PRODUCTS. Adopted by the Interdepartmental Petroleum Specifications Committee of the Government. Bureau of Mines Technical Paper No. 298. Published by the Superintendent of Documents, City of Washington. 21 illustrations; 58 pp.

This booklet is intended for the use of petroleum laboratories and the methods described are those used officially in the routine testing and inspection of petroleum products bought under Federal specifications. Methods of testing color, pour point, viscosity, distillation range, flash and fire points and many other properties of lubricating oils and fuels are included. A list of the other Bureau of Mines publications on refinery engineering and petroleum chemistry is printed in the booklet.

CHROMIUM STEELS AND IRON. By Leslie Aitchison. Paper read before the Institution of Automobile Engineers, 28 Victoria Street, Westminster, S. W. London, England. 24 pp.; 17 illustrations.

The characteristics of chromium steels and irons are described in this paper for the benefit of the automobile engineer. The chemical and physical characteristics of two series of chromium steels are covered, one having carbon and chromium-contents ranging from 0.34 to 0.47 and 1.28 to 16.07 per cent respectively, while the other has a carbon-content of from 0.07 to 1.01 per cent, with chromium ranging from 11.00 to 12.00 per cent.

The conclusions drawn can be summarized as follows:

(1) Chromium steels can be obtained that possess all the mechanical properties desired by the automobile engineer for structural or engine work. From the standpoint of mechanical properties the chromium-content need not exceed 3 per cent; while for

small parts from 1.00 to 1.50 per cent of chromium is ample

(2) In steels containing a medium proportion of carbon, 0.35 to 0.45 per cent, the effect of increasing amounts of chromium on the mechanical properties is at first great, rising rapidly to a maximum that holds for a considerable distance, after which there is a slight falling away

(3) The influence of carbon on the mechanical properties is the greater, overshadowing the chromium very much. This applies to both low and high-chromium steels

(4) There is a considerable range where both the so-called stainless and mechanical properties are excellent; further, that many of the mechanical requirements of the automobile designer can be met fully by the stainless irons having a carbon-content of from 0.07 to 0.15 per cent. This material possesses as good stainless properties as the higher carbon compositions, and in addition has the advantage of better machining qualities

(5) The mechanical advantages of vanadium additions to chromium steels are not apparent; that is, the increase in strength is not commensurate with the cost.—F. P. G.

RADIATORS FOR AIRCRAFT ENGINES. Bureau of Standards Technologic Paper No. 211. Published by the Superintendent of Documents, Government Printing Office, City of Washington.

The paper describes the laboratory investigations relating to aircraft engine radiators that were conducted by the Bureau during the World War and in the 2 years immediately succeeding it. Individual reports covering many phases of the subject have been published previously by the National Advisory Committee for Aeronautics and in scientific and engineering journals. These reports, however, lack the systematic coordination, uniform terminology and unified mathematical treatment that should characterize a handbook on the subject. Moreover, the problems that were investigated first because of their greater importance and which were the subjects of the earliest reports are for that reason not so well covered as is now possible because the later work has thrown much additional light upon matters not settled at the time of the publication of the earlier reports.

Accordingly, the present paper is much more than a reprint and is in fact a revision and recompilation of the material available. The special investigations reported include: Development of the methods of measuring air-flow in radiator tubes, experiments upon the effect of the nature of surface upon air-flow and upon heat dissipation from the surface of metal tubes to a high velocity airstream, experiments to ascertain the degree of turbulence in the air tubes of a radiator core and the mapping of the temperature distribution axially and transversely in the air tubes of radiator cores.

In addition to recording these special investigations, the paper contains a full description of the laboratory methods and instruments employed and descriptions of the physical properties and geometrical characteristics of the various cores tested. Performance characteristics of 66 types of core are given in graphical form and empirical equations for relating the heat dissipating power of a radiator to the air-flow through the core, for computing the performance of a core of any depth from that of a core of exactly similar construction but of a different depth and for computing the heat dissipating power of such cores from their geometrical dimensions are included.

The effectiveness of indirect cooling surface, that is, cooling surface not backed by flowing water is developed mathematically, reduced to a practical working equation and applied to the computation of the best fin dimensions for given conditions. Also, the equations of thermal conductivity are applied to produce a table showing the temperature drop through metallic water-tubes of radiators having various wall thicknesses of different kinds of material.

The effect upon the heat dissipation of a varying rate of water-flow is considered in detail; also a comparison of different methods of testing radiator cores calorimetrically and a great many other topics having to do with the performance of the radiator in different positions and at various altitudes are discussed.

INTERFERENCE METHODS FOR STANDARDIZING AND TESTING PRECISION GAGE BLOCKS. Bureau of Standards Scientific Paper No. 436. Published by the Superintendent of Documents, Government Printing Office, City of Washington.

This paper describes interference methods by which the planeness and parallelism errors of precision surfaces can be measured and the length of standard gages can be determined by direct comparison with the standard light waves with an uncertainty of not more than a few millionths of an inch. The extensive use of precision gages, which are usually blocks of steel, although other metals are sometimes used, having two opposite faces plane, parallel and a specified distance apart, necessitated by the small tolerances allowed in the manufacture of interchangeable machine parts, has required more accurately determined end standards and more rapid and precise methods for comparing gages with these standards than have been available previously. As comparisons of end standards with line standards by micrometer-microscopes and of precision gages with end standards by contact instruments are subject to appreciable errors, methods that make use of the interference of light waves have been used by the Bureau of Standards in making these measurements. The errors of other gages can be determined by comparison with these calibrated standards with equal precision. The process makes the standard light waves that have been determined to 1 part in 4,000,000 or 5,000,000 relative to the international meter the standards of length for this work. The apparatus used for calibrating the standards and comparing other gages with these standards is illustrated by line drawings and thoroughly explained.

SKIN FRICTIONAL RESISTANCE OF PLANE SURFACES IN AIR. By W. S. Diehl. National Advisory Committee for Aeronautics Technical Note No. 102. 4 pp.; 2 illustrations.

This bulletin gives an abstract of recent German tests of skin frictional resistance in plane cloth surfaces. With doped surfaces, the coefficient of frictional resistance decreased uniformly with the Reynolds number, but with rough surfaces, that is, with the cloth in the original condition, with the nap singed off, the values were erratic.

In the notes incorporated in the abstract, W. S. Diehl of the United States Navy Bureau of Aeronautics expresses the results of the tests on doped surfaces in a convenient engineering formula, accompanied by two graphs to facilitate calculation.

POWER ALCOHOL: ITS PRODUCTION AND UTILIZATION. By G. W. Monier-Williams. Published by Henry Frowde and Hodder & Stoughton, London. 312 pp. 48 illustrations.

This volume should be of considerable interest to both producers and consumers of power alcohol. The author discusses in detail the various sources of alcohol, the economics of its production, methods of distillation and its chemical and physical properties from the motor-fuel standpoint. A final chapter discusses the results of tests made on engines running on alcohol and alcohol mixtures. There are many diagrams and statistical tables and excellent bibliographies at the end of each chapter.

In one respect, at least, the book is an innovation. One chapter deals with the production and use of alcohol from the point of view of the Latin-American and tropical planter, who grows sugar-cane, and depends upon American cars and trucks to solve his transportation problem. He cannot get gasoline and is interested in the possibility of producing alcohol to run his car from the sugar cane that he grows. Here is food for thought for the manufacturer who is interested in extending his foreign market, and is willing to adapt his engines to run on alcohol, if necessary.

Applicants *for* Membership

The applications for membership received between July 15 and Aug. 15, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

BAILEY, WALTER H., Royal Indemnity Co., 84 William Street, *New York City.*

BAUM, SEYMOUR J., works manager, Brewster & Co., *Long Island City, N. Y.*

BOHLEN, CHARLES, draftsman, International Motor Co., *New York City.*

BREWSTER, WILLIAM, president, Brewster & Co., *Long Island City, N. Y.*

CIAFFONE, JAMES V., president and general manager, Private Engineering Automobile School, *Long Island City, N. Y.*

COX, CAPT. M. R., *Fort Sill, Okla.*

DUNSMORE, R. B., manager, Security Mfg. Co., *Los Angeles, Cal.*

FIELD, FRANK JAMES, chief engineer, Feroda Ltd., *Sovereign Mills, Chapel en le Firth, Derbyshire, England.*

GLICK, PHILLIP P., inspector, Moon Motor Car Co., *St. Louis.*

GOODYEAR, JOSEPH P., engineer, Oakland Motor Car Co., *Pontiac, Mich.*

GROVES, FRANK A., president and general manager, Stemco Engineering Co., *Dayton, Ohio.*

HERMAN, F. P., manager of factory sales, Houdaille Co., *Buffalo.*

HIRSH, DAVID H., instructor, Greer College of Automotive Engineering, *Chicago.*

HOLZ, FRED C., service inspector, Gomery-Schwartz Motor Car Co., *Philadelphia.*

JACKMAN, CHARLES W., draftsman, Chevrolet Motor Co., *Detroit.*

KAHRL, ASA, proprietor, Kahrl & Lamm, *Farmington, Mich.*

KARLSON, K. EDWIN M., factory manager, Cleveland Automobile Co., *Cleveland.*

LANG, HARRY O., heat treating engineer, Oakland Motor Car Co., *Pontiac, Mich.*

MACVICHIE, DONALD, industrial engineer, Motor Transit Co., *Los Angeles, Cal.*

MARLEY, GUY R., shop superintendent, Air Service, *Fairfield, Ohio.*

MENEWISCH, WILLIAM T., draftsman, Fox Motor Car Co., *Philadelphia.*

MEYERS, WILLIAM H., draftsman, Nordyke & Marmon Co., *Indianapolis.*

MITRA, ANIL C., foreign representative, Auto Transit Syndicate, *Calcutta, India.*

OAKLEY, WILLIAM E., vice-president, Oakley Valve Co., *New York City.*

ORR, ERNEST M., assistant general manager, Oakland Motor Car Co., *Pontiac, Mich.*

PILIBOSS, EDWARD D., designer and engineer, Chevrolet Motor Car Co., *Flint, Mich.*

PRICE, LEONARD C., student, Cornell University, *Ithaca, N. Y.*

ST. CROIX, JAMES J., body engineer, Fifth Avenue Coach Co., *New York City.*

SALTZMANN, FRED, assistant foreman, Milwaukee Die Casting Co., *Milwaukee.*

SHEAFF, HOWARD, aeronautical engineer, L. W. F. Engineering Corporation, *College Point, N. Y.*

SIMPSON, ALBERT F., field representative, foreign department, White Co., *New York City.*

SINK, ARVEL T., engineer, Fellows Gear Shaper Co., *Springfield, Vt.*

STANLEY, THOMAS R., general manager, Oil City Engine & Power Co., *Oil City, Pa.*

STOREY, EDWIN F., layout draftsman, Lincoln Motor Co., *Dearborn, Mich.*

STOREY, FRANK E., layout and detail, Chevrolet Motor Co., *Detroit.*

TEETOR, LOTHAIR, sales manager, Indiana Piston Ring Co., *Hagerstown, Ind.*

TIFFANY, H. R., superintendent, Sewell Cushion Wheel Co., *Rochester, N. Y.*

TILLEY, NORMAN NEVIL, aeronautical mechanical engineer, McCook Field, *Dayton, Ohio.*

TODD, RALPH R., chief inspector, Oakland Motor Car Co., *Pontiac, Mich.*

TOOLE, GEORGE E., division chief inspector, Oakland Motor Car Co., *Pontiac, Mich.*

TOWNSEND, H. H., chairman of the board and general manager, Citizens Oil Corporation, *Trenton, N. J.*

UPSTILL, EDGAR D., experimental assistant, White Motor Co., *Cleveland.*

VOGEL, FRANK E., student, 225 Fairchild Street, *Iowa City, Iowa.*

WATERS, DEVER, sales manager, Schwarz Wheel Co., *Frankford, Philadelphia.*

WEAVER, CAPT. WILLIAM K., JR., *Camp Holabird, Md.*

WILLIAMSON, EDWARD E., supervisor of motor apparatus, Boston Fire Department, *Boston.*

WORRALL, GILBERT A., service manager, foreign department, White Co., *Long Island City, N. Y.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between July 10 and Aug. 10, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F-M) Foreign Member; (E S) Enrolled Student.

ADAMS, JOHN NEWELL (E S) student, University of Michigan, Ann Arbor, Mich., (mail) 123 Maple Street, *Somerset, Ky.*

AUSTIN, E. W. (A) district manager, Timken Roller Bearing Co., Canton, Ohio, (mail) 533 Bulkley Building, *Cleveland.*

BAUCH, CHARLES E. (J) bureau of aeronautics, Navy Department, *City of Washington.*

BEEGLE, F. N. (A) president, Union Drawn Steel Co., *Beaver Falls, Pa.,* (mail) Darlington Road.

BERRY, JOHN T. (M) superintendent, Carriage Factories, Ltd., *Orillia, Ont., Canada,* (mail) P. O. Box 616.

BLOMBERG, M. (M) motor truck equipment, National Steel Car Corporation, Ltd., *Hamilton, Ont., Canada,* (mail) 121 Charles Street.

BOYER, RALPH L. (E S) student, Ohio State University, Columbus, Ohio, (mail) Central Y. M. C. A., 55 Hanson Place, *Brooklyn, N. Y.*

BRAINERD, HOWARD S. (A) metallurgist, Ingersoll Rand Co., *Phillipsburg, N. J.*

BRAMLEY, M. F. (A) president and general manager, Templar Motors Co., *Cleveland,* (mail) 886, The Arcade.

BRICE, JOHN R. (J) layout draftsman, Yellow Cab Mfg. Co., *Chicago,* (mail) 1716 North Campbell Avenue.

BROOCK, HAROLD (J) 80 Longfellow Avenue, *Detroit.*

BROOKS, HOWARD (A) assistant superintendent, American Smelters Securities Co., *Velardena, Durango, Mexico.*

BRUNNER, ALEXANDER (A) proprietor, A. Brunner & Son, 744 South Twelfth Street, *Newark, N. J.*

BUCKBEE, GEORGE A. (A) salesman, Gurney Ball Bearing Co., Jamestown, N. Y., (mail) 20 East Jackson Boulevard, *Chicago.*

BURTON, ROBERT B. (M) chief draftsman, Rolls-Royce of America, Inc., Page Boulevard, *Springfield, Mass.*

CASTLE, D. W. (A) instructor, South High School, *Omaha, Neb.,* (mail) 1334 South 26th Street.

CLEMENTS, M. R. (J) Studebaker Corporation of America, *South Bend, Ind.,* (mail) 627 Portage Avenue.

COOK, HARMON J. (A) works manager, Standard plant, Torrington Co., *Torrington, Conn.*

CRAWFORD, KENNETH G. (J) draftsman, Sanford Motor Truck Co., Syracuse, N. Y., (mail) 46 Lenox Avenue, *Oneida, N. Y.*

DANIEL, E. H. (M) president and general manager, London Motor Plow Co., *Springfield, Ohio,* (mail) 1101 West Pleasant Street.

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Chronicle and Comment

The Production Meeting

THE success of the Society's first National Production Meeting seems assured. Although production executives are an extremely busy group of men at this time, they are showing interest in the program, preparing papers and arranging factory visits. The papers will cover labor-control systems, variations in gear-tooth generation, gear noise resulting from faulty mounting, cylinder production from the ore through to the engine assembly and other equally instructive subjects. A special *Meetings Bulletin* will reach the members about Oct. 15, with all final details of the program. Progress in the arrangements is reported on p. 373 of this number of THE JOURNAL. Make your plans to visit Detroit on Oct. 26 and 27.

The Sections Meetings

THE Sections of the Society open their season of meetings activity in earnest with the coming of October. Meetings are scheduled in all of the Section centers. Programs are varied and cover a wide field. Metal airplanes, research, petroleum refining, wire and disc wheels, steel bodies and road-testing are among the subjects listed for discussion. Interest in the Sections meetings appears greater than ever before. If you have not taken advantage of this privilege of your membership you are losing one of the major benefits of association with the Society. Turn to p. 380, note the imposing list of October meetings and attend the one in or nearest your home city.

The Employment Service

THERE have been 70 men placed in positions in the automotive industry since April 1 by the Employment Service of the Society. One can appreciate the value of this service to the members and the industry when it is recognized that a large majority of these positions are classed as executive and carry responsibility. A semi-weekly bulletin of positions available is mailed to every applicant for employment. A semi-weekly bulletin of men seeking positions is sent to a selected list of representative firms for their information. The service is probably the only one of its kind specializing in the automotive field, and is free to both employer and employe. Men are placed in all divisions of the industry; sales, production, service, research and engineering. When you are in need of an executive with training suited to a particular line of automotive work, be sure to consult the Society's Employment Service.

Fuel and Engine Developments

IT is well recognized that the production of crude oil has not increased as rapidly as the demand for it, and that the demand for gasoline would not have been met if the oil industry, on the one hand, had continued to supply the very volatile gasoline produced 5 to 10 years ago, or if the automotive industry, on the other hand, had continued to build engines, carbureters and other equipment adapted to only those lighter fuels. Both industries have developed progressively; the oil industry by increasing the gasoline yield through taking a larger fraction of the barrel of fuel, that is, by making heavier gasoline by "cracking," and by the use of casinghead or natural gasoline, and the automotive industry by directing its energies to designing engines and other equipment that would use heavier gasoline to better advantage.

The 1923 Summer Meeting

WHERE shall the Summer Meeting of 1923 be held? This important question is deserving of the most careful consideration by the members of the Society. Those who attended the recent meeting at White Sulphur Springs pronounced it the ideal location for the Society's future summer gatherings. On the other hand, one continually hears criticism of this type of meeting place in conjunction with a demand for a return to the boat trip on the Great Lakes. The 1922 Summer Meeting did not attract as large a number of the members engaged in engineering and design work as seemed desirable. Was it the location, or do these men favor a variation in the type of meeting? These are perplexing questions and the Meetings Committee and the Council are desirous of receiving the opinions of members to guide them in selecting a location and defining the plan of the 1923 meeting. Please address letters on this matter to the Society's office in New York City and they will be referred to the Meetings Committee.

Motor-Fuel Quality

THE volatility of motor gasoline as reported by the Bureau of Mines, as a result of its semi-annual motor-gasoline survey, has improved slightly over that reported 6 months ago. Up to about 2 years ago

motor-fuel quality showed a steady decline; since that time the quality has been improving steadily, contrary to the popular impression.

Of perhaps greater importance to the user is the fact that the *uniformity* as well as the quality is improving. In other words, supplies of gasoline bought at different filling-stations or at different times vary much less than formerly. The Bureau of Mines' report states that the differences between Winter and Summer gasoline is still maintained by the refiners. The difference in volatility is made intentionally to facilitate starting in cold weather.

Comparing the distillation curve of the average gasoline sold throughout the Country in July 1922, as determined by the Bureau of Mines with the Federal specifications on which all motor gasoline is purchased by the Government, it is found that the average commercial gasoline is considerably better than the Government requirements, except for a small range where the two curves are identical.

Service

ABSTRACT definitions of "Service" include: Provision for the carrying out of work for which there is a constant public demand; an agency for meeting a general or recurrent need.

At the last meeting of the Society, F. A. Bonham, on behalf of the Service Committee of the National Automobile Chamber of Commerce, discussed the relationship of engineering to service in connection with the maintenance of motor vehicles. He expressed the opinion that automotive engineers in general do not understand adequately the real basis of such service. He stated that the demand for properly trained and conscientious automobile mechanics so far exceeds the supply that unsatisfactory repairmen will have to be accepted by the industry for a number of years.

The demand is for repairs and the replacement of parts promptly and at low cost. Standard repair operations are being developed. The "flat rate" is in vogue, an advance quotation of the cost of repairs being based on it. But, Mr. Bonham pointed out, the limiting factor of the whole operation of service is design. The engineer's product is operated and repaired by persons who do not know much about it. The resultant expense of carelessness and incompetence, as well as of deficient design, must be borne.

The service problem is an inherent part of design. Mr. Bonham said that major considerations are:

(1) Sequence of operations in manufacture and assembly, as affecting the making of repairs
(2) Dismounting of assembly units by the use of simple tools
(3) Low cost of wearable parts
(4) Design of simple tools for making repairs
(5) Dependability of constituent mechanisms of complete apparatus

The greatest possible simplicity and accessibility of parts are of course essential. Thought must be given assiduously to what the repairman has to do in his work. Service Departments obviously can furnish much valuable information on the whole matter of maintenance necessities and conveniences. Mr. Bonham believes that a comprehensive thorough study of service conditions will have a far-reaching effect on automotive design. The task is to minimize the effect of the inevitable abuse of machines by users and of the carelessness and incompetence of repairmen.

It was urged that engineers do everything possible to make newly designed or changed parts interchangeable with corresponding parts in old car models, as this will reduce greatly the burden of the service-stations in supplying replacement parts. The engineer has a responsibility to the public equal to that of anyone in the industry. The dealer and the serviceman are entitled to his best efforts. The reduction to practice of "pet" engineering theories should not be tolerated if this involves experimentation or unreasonable servicing. This is fundamentally important from the standpoint of unjustified major expense. Ease and low cost of maintenance and repair must be borne in mind constantly.

SEPTEMBER COUNCIL MEETING

THE meeting of the Council held on Sept. 19 at New York City was attended by President Bachman, Vice-Presidents Segner and Brautigam, Councilors Crane, Davis, Scott and Smith, Past-President Beecroft and Treasurer Whittelsey. H. W. Alden, A. J. Scaife and E. P. Warner, nominees for the 1923 Council, also attended.

The financial report as of July 31 showed a net balance of assets over liabilities of $121,863.37, this being $18,114.37 less than the corresponding figure on the same day of 1921. The income of the Society for the first 10 months of the current fiscal year amounted to $137,523.87. The operating expense during the same period was $154,290.99.

A budget for the fiscal year beginning Oct. 1, 1922, was adopted. This budget anticipated an income for the 1923 fiscal year of $268,585.25, and expenses of the same amount.

Fifty-nine applications for individual membership were approved. The following transfers in grade of membership were approved: From Junior to Member, H. W. Simpson, D. T. Stanton, Quentin Twachtman; Associate to Member, O. J. Rohde, Peter M. Erikson.

It was reported that up to Sept. 18, 1922, 546 applications for membership had been received, as compared with 576 received during the first 9 months of 1921, and 946 during the first 9 months of 1920. On Aug. 31 there were 5685 names on the rolls of the Society, including affiliate member representatives and enrolled students, as compared with 5609 on the same day of 1921 and 4781 on the same day of 1920.

The appointment of Archibald Black as member of the Aeronautical Safety Code Sectional Committee and Chairman of Subcommittee 3 on Equipment and Maintenance of Airplanes was approved.

It was reported that since April of this year 68 members have notified the Society office that they have secured positions through the S. A. E. Employment Service. No doubt many others have been placed but failed to mention the fact.

The next meeting of the Council has been scheduled to be held in Detroit on Oct. 25, previous to the Production Meeting on the 26th and 27th.

Vaporization of Motor-Fuels

By P. S. TICE[1]

MID-WEST SECTION PAPER *Illustrated with* PHOTOGRAPHS AND CHARTS

THE author gives a brief and purely qualitative treatment of what a vapor is, where it comes from and how it appears; the necessity of vaporizing a liquid fuel before attempting to burn it; the separate effects of the conditions that control vaporization; and the heat-balance of vaporization. This is done to summarize the conditions surrounding and controlling fuel vaporization in the cycle of operation of a throttle-controlled internal-combustion engine, fitted with an intake-manifold and a carbureter. Charts and photographs are included and commented upon, descriptions being given of actual demonstrations that were made at the time the paper was presented. The conclusion is reached that it is well to depend as little as possible upon the cylinder heat and temperature to complete the vaporization of the fuel.

THE purpose of this paper is to summarize the conditions surrounding and controlling fuel vaporization in the cycle of operation of a throttle-controlled internal-combustion engine, fitted with an intake-manifold and a carbureter. The procedure by which it is hoped to arrive at a reasonably clear and comprehensive picture of the relative values and relations of these conditions will include: (*a*) a description of what a vapor is, where it comes from and how it appears; (*b*) a demonstration of the necessity of vaporizing a liquid fuel before attempting to burn it; (*c*) a statement, with descriptions of actual demonstrations, of the separate effects of the conditions that control vaporization; and (*d*) a discussion and description of actual demonstrations of the heat balance of vaporization. These items will be given only brief and purely qualitative treatment. Generalizations dealing with the effects of impure or mixed fuels will be introduced and an attempt made to show the advantages and limitations of an engine's intake system as an environment in which to stage the vaporization process.

VAPOR AND VAPORIZATION

A vapor has been defined as a gaseous or elastic fluid phase of a volatile liquid at or near its condensation point, or below its critical point so that it can be liquefied by pressure alone; evaporation, as the change by which a substance is converted from the liquid state into, and carried off in, the vapor state; and a volatile liquid, as one that evolves vapor rapidly at ordinary temperatures.

The two terms evaporation and vaporization are commonly used interchangeably, and no sharp distinction can be drawn between them. However, it is convenient to consider the formation of vapor under natural conditions as evaporation, and vaporization as the act or process of forming a vapor by subjecting a liquid to artificially modified conditions that hasten its evaporation. From the molecular standpoint, vaporization or evaporation means the flying off of molecules against the forces of molecular attraction at the surface of the liquid, the molecules losing kinetic energy and gaining potential energy as they leave the liquid surface. The liquid and the vapor of any substance have identical compositions, and differ only in state. This is a short way of saying that the only difference is that of molecular aggregation or closeness of grouping of the molecules.

A simple demonstration with a rudimentary flash-point apparatus will serve to show that the talk of the carbureter people about the virtues of vaporization is not wholly unwarranted. To demonstrate this let us consider four tubes in which have been placed (*a*) gasoline; (*b*) gasoline from which the most volatile constituent has been removed; (*c*) kerosene; and (*d*), nothing; in other words, the last tube is a barometer. The differences between the height of mercury in the barometer and in each of the other tubes containing fuel measure the relative volatilities of the fuels. Samples of each of the three fuels, cooled to about 0 deg. fahr., are placed in three small covered beakers in the same order as in the tubes and sparks are caused to pass continually over their surfaces until combustion starts. This will occur as soon as the fuels have attained temperatures at which vapor is given off at a rate to support combustion, and the three fuels will start to burn in the order of their volatility.

Such a demonstration suggests four very important things: (*a*) the striking differences in volatility that obtain among liquid fuels; (*b*) that fuels will not burn in the liquid state; (*c*) that vapor must be present in quantities to support combustion; and (*d*) that the greater the proportion of the vapor to the liquid in a given space, the more rapid the combustion will be, provided there is sufficient oxygen present to support it.

CONDITIONS CONTROLLING VAPORIZATION

It is a matter of common experience that evaporation is retarded and finally ceases if the vapor is confined at the liquid surface. This is shown in each of the barometer tubes containing only a liquid and its vapor, and in which the heights of the mercury have been constant for some time. It then becomes evident that the evolution of vapor has stopped. The reason that evaporation stops in such a case is that some of the molecules of the vapor, in their normal motions, strike the surface and join the liquid again; and, as the number of vapor molecules in a given space increases, due to the increased density of the vapor, the number of molecules so returning to the liquid in a given time will likewise increase, until finally the average number returning to the liquid will equal the average number leaving it. Under this condition the vapor is in equilibrium with the liquid.

When this equilibrium condition is reached, the space above the liquid is said to be saturated with vapor; and the density, and therefore the pressure, of the vapor are then the maxima that can exist in the presence of the liquid at the temperature of the experiment. This maximum pressure is called the saturation pressure. Proof that this pressure is the maximum that can exist is afforded by shifting the leveling bulb that forms a part of the demonstrating apparatus. The only result will be to change the vapor volume. The pressure remains constant as shown by the fixed difference in height

[1] M.S.A.E.—Engineer directing the carbureter division, Stewart-Warner Speedometer Corporation, Chicago.

of the mercury columns. The value of this pressure depends only upon the temperature; that is, upon the average molecular velocity of the liquid, and is unaffected by the presence of any non-combining gas or vapor. This is strictly true only for single liquids such as water or benzol. With mixed liquids such as gasoline the pressure or the temperature changes somewhat with changes in the volume of the vapor space.

Evaporation is accelerated by passing a current of air over the surface of a liquid, or by any expedient that removes the vapor from the liquid surface. This is demonstrated by the high rate of evaporation when a small stream of air is used to sweep the vapor away from the liquid surface. Since the vapor is not allowed to accumulate, it must remain unsaturated, equilibrium cannot be reached, and the liquid disappears by evaporation.

The vapor-pressure, a term denoting the saturation-pressure when used without qualification, invariably increases rapidly with a rise of temperature. A rise of pressure occurs as the temperature of the liquid is raised gently by heat from a small resistance-coil wound around the tube in which the liquid is contained. A rise of temperature is another way of saying that the molecules are moving faster; the faster the liquid molecules move the more numerous are those that fly off from the surface. To keep the liquid in the tube where heat can be applied directly to it, the leveling bulb must be raised. The change in pressure is measured by the height through which the bulb is raised.

If the temperature of a liquid is raised sufficiently to cause its vapor-pressure to become equal to the external pressure, vapor bubbles form freely in the interior of the mass of liquid, by the well known process of boiling. The temperature at which this occurs, under standard atmospheric pressure, is called the boiling-point.

When a flask contains fairly hot water and is in open communication with the atmosphere, the water is at a high enough temperature to exert a vapor-pressure nearly equal to that of the atmosphere. A further rise of temperature, which can be brought about by setting the flask on a hot plate, results in the formation of vapor bubbles as soon as the pressure of the water vapor equals that of the atmosphere.

If the external pressure remains constant, the temperature of a pure boiling liquid will remain constant at its boiling-point, while the liquid passes off by vaporization. When water at sea level boils vigorously, a thermometer inserted in it will continue to register 212 deg. fahr. as long as the boiling continues. It is evident that water in the vapor state is leaving the surface and does not return to it. But, if the liquid is contained in a closed space, it can be made to boil at temperatures below its normal boiling-point by reducing the pressure. This is made evident when the liquid in the vapor tube is allowed to cool slightly below its normal boiling-point. If the pressure in the tube is reduced, as is the case when the leveling bulb is lowered, the liquid will boil again at its now reduced temperature, as indicated by the bubbles seen to form in it. The same phenomenon can be shown by removing from the hot plate the flask in which water is boiling, sealing it and applying ice to the vapor space above the liquid. In this case the ebullition will be more spectacular than in a tube, because the mass of liquid is greater and the pressure can be reduced conveniently to a much lower value. As the boiling progresses, the temperature will drop rapidly. Removal of the ice, the presence of which causes the pressure reduction, will stop the boiling, while the replacement of the ice will cause the boiling to start again. The boiling may be made to continue until the temperature of the liquid is about 20 deg. fahr. below that of the room.

Correspondingly, if the pressure is raised, the temperature of a liquid must be raised above its normal boiling-point to secure ebullition. There is for every temperature a corresponding equilibrium or saturation-pressure of the vapor, and vice versa. But the temperature and pressure cannot be raised indefinitely. As the temperature is raised, the density of the saturated vapor increases, while that of the liquid decreases. If the temperature is raised sufficiently, the densities of the liquid and of its vapor become equal at a definite temperature depending upon the substance. This temperature is called the critical temperature, because it is the limiting temperature at which a separation of the liquid and the vapor states can be observed. If we seal a liquid and its vapor in a small tube and raise the temperature of the tube, a point will be reached at which it will be impossible to distinguish between the liquid and the vapor. Reducing the density of the liquid causes its upper boundary to rise until the liquid completely fills the tube. All parts of the tube are now shared equally by the liquid and its vapor. When the temperature drops, the tube, from bottom to top, becomes filled with a fog of reaggregated molecules, and the meniscus separating the two states rapidly settles as the density of the liquid increases.

At temperatures above the critical temperature there is what is known as continuity of state, since it is impossible to cause separation of the two states by modification of the pressure alone. Attention is directed to Fig. 1, which is a plot of isotherms on pressure-volume coordinates for carbon dioxide. The properties shown in this graph are characteristic of all substances. It is a plot of the typical pressure-volume relations of a substance at temperatures below and above the critical value. The horizontal portions of the isotherms enclosed by the dotted curve show the characteristic change of state, from liquid to vapor, or vice versa, at constant pressure and constant temperature. To the right of the horizontal portions, the substance is completely vaporized and behaves like a gas. To the left, it is all liquid. Note that as the critical temperature is approached there is a continually smaller change of volume between the all-liquid and all-vapor conditions. This illustrates a statement to be made later to the effect that the latent heat becomes less with increased temperature, and at the critical temperature has a zero value.

The two controlling factors in the process of vaporization are the temperature of the liquid and the vapor-pressure exerted on its surface. Raising the temperature at a constant vapor-pressure, or lowering the vapor-pressure at a constant temperature, or doing both simultaneously, causes a further evolution of vapor from the liquid. This is the sole means available for accelerating the vaporization of a liquid.

The controlling pressure in vaporization is only the pressure exerted by the vapor upon the liquid surface. A demonstration of the fact that the saturation pressure of a vapor is unaffected by the presence of any non-combining gas or vapor makes clear what is meant by the term partial pressure. This can be done by using a normal barometer tube devoid of air above the mercury. If a small quantity of liquid is introduced into this tube, its vapor at once fills the space above the mercury, and the mercury column falls an amount proportional to the vapor-pressure of that liquid at the prevailing temperature. When a vapor-pressure tube from which all air has been expelled, is used as a barometer, if the tube cock is opened,

an amount of air admitted, the tube resealed, and the bulb returned to its former position at which the barometric height was shown, the pressure above the mercury will be higher than before. The difference between the new height of the mercury column and the barometric height is a measure of the pressure due to the air in the tube. If, now, we introduce into this tube some of the same liquid that was put into the conventional barometer tube, and set the leveling bulb so that the volume above the mercury in the tube is the same as before the liquid was admitted to it, it will be found that the pressure-rise due to the vapor in the tube containing air is exactly the same as that in the first barometer tube which contained no air when the liquid was put in.

This shows that a given equilibrium-pressure exists for a given liquid at a given temperature, whether that vapor-pressure is the total pressure exerted on the liquid surface, or it is only a part of the total pressure so exerted. The only difference to be noted in the two cases is that the equilibrium pressure is attained much more rapidly when only the vapor occupies the space above the liquid. The molecules of a gas interfere with the free dispersion of the vapor molecules, causing a greater average vapor density at the surface of the liquid in one case than in the other.

When a vapor is in equilibrium with its liquid and the vapor exerts only a fraction of the total pressure, a change of pressure of the total fluid above the liquid, such as is produced by a change of volume, will change the pressure of the vapor in a like ratio and destroy the equilibrium relation. But, if the temperature of the liquid is kept constant, equilibrium will become reestablished at exactly the same vapor-pressure as before. Depending upon whether the volume is increased or decreased to change the total pressure, the partial pressure of the vapor, and therefore the relative vapor-content of the space, will become correspondingly greater or less. This is a matter of great importance in the practical carburetion of an engine, as will be shown later.

Heat Balance of Vaporization

The conception of heat as a form of energy, the common manifestation of which is temperature, is familiar to everyone. Degrees of temperature represent intensities of heat and not quantities of heat, because, when equal quantities of heat are imparted to two bodies of equal mass but of unlike substance, one is found to be hotter than the other. The specific heats of the two substances are then said to be different, the one showing the higher temperature having the smaller specific-heat capacity. Besides raising the temperature, heat usually causes an increase of volume. Part or the whole may go toward producing changes of state by fusion or vaporization and may cause chemical reactions. Part may also be transformed into other forms of energy, producing the phenomena of light, electricity and the like. Heat may be imparted to a body (*a*) by conduction, as along a metal rod; (*b*) by convection, as through the rooms of a house by air currents; and (*c*) by radiation, as from the sun to the earth.

To convert a substance from a liquid to a vapor without change of temperature, it is necessary to impart to it a certain amount of heat. The quantity of heat so required per unit mass of the liquid is the latent heat of vaporization. It is called latent because it causes no temperature-rise and disappears, so far as our senses are concerned, in performing the change of state. Since vaporization of a liquid produces a great increase in volume, which acts against a definite pressure, external work must be done by the vapor as it is formed. In the performance of this work the heat disappears or becomes latent, as was demonstrated by the water in the flask set on a hot plate boiling without any change of temperature. The latent heat of vaporization diminishes with increasing temperature, and becomes zero at the critical temperature, where the distinction between a liquid and a vapor vanishes.

The total heat of vaporization of a saturated vapor at any temperature is the quantity of heat required to raise a unit mass of the liquid from any convenient zero to the temperature in question, and then to evaporate it at that temperature under the constant corresponding pressure of saturation. If, instead of converting a liquid into its vapor, we reverse the process or condense the vapor into its liquid, and bring the temperature of the recovered liquid back to the value from which it was initially raised, the total heat of vaporization is recovered and transmitted from the liquid to some other body.

Let us take two bulbs, sealed to each other in the form of an inverted U, and containing only water and its vapor. The water is all in the outside bulb, and the other one, immersed in the beaker, contains only water vapor. Applying heat to the exposed bulb causes the water to vaporize, making heat latent in so doing. The vapor thus evolved is condensed in the other bulb, in which liquid begins to appear. Upon condensation of the vapor, the heat that was latent in it reappears and is taken up by the walls of the bulb and the air in the beaker surrounding it. That heat will reappear at the bulb immersed in the beaker can be made evident from observation of the convection currents set up in a liquid poured into the beaker and the vapor coming off that

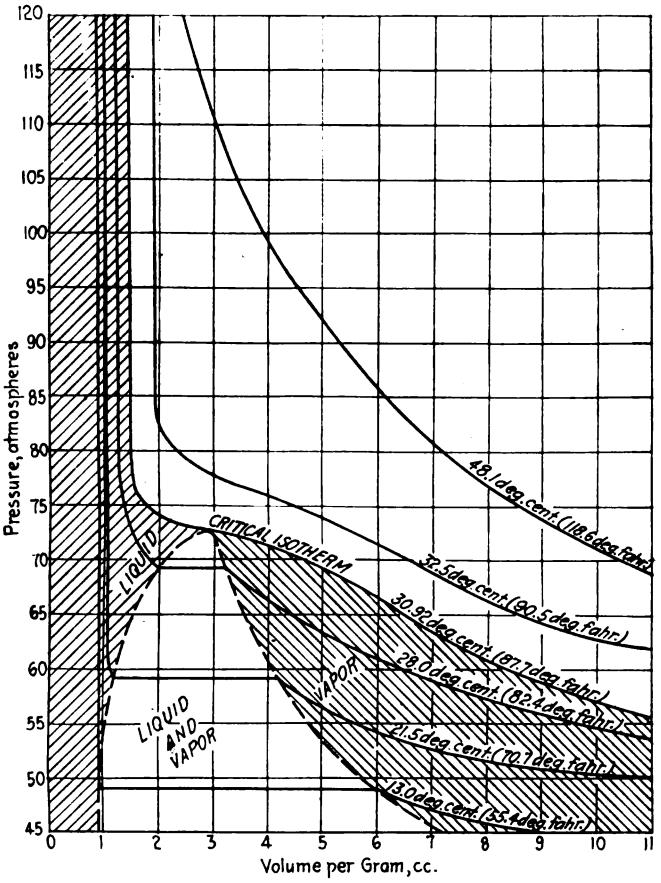

Fig. 1—Series of Curves Showing the Typical Pressure-Volume Relations of a Substance at Temperatures Above and Below the Critical Value

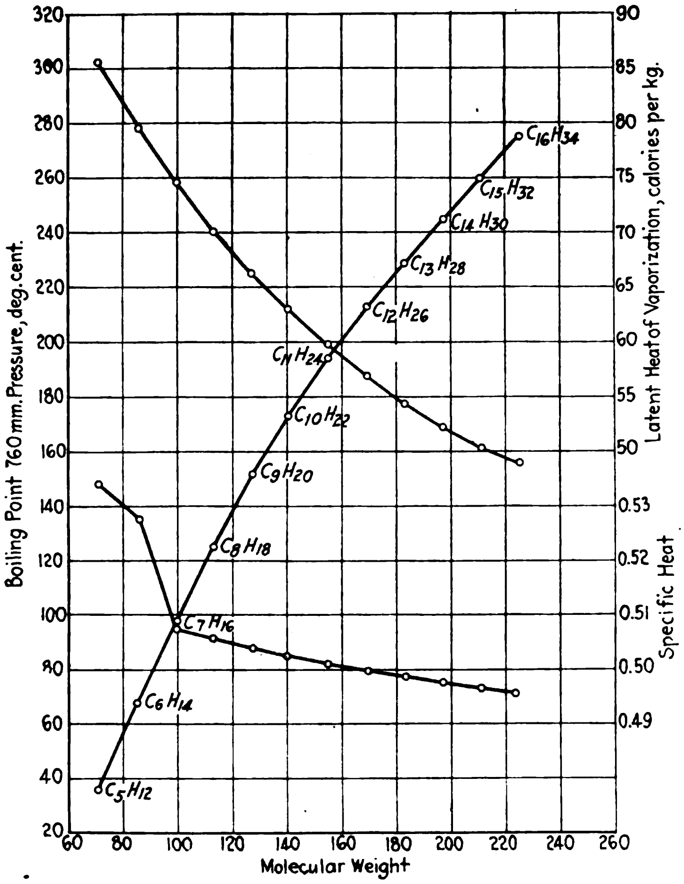

Fig. 2—Grouping of Physical Constants Referred to Molecular Weight for the Members of the Paraffin Series Found in Our Fuels

liquid. With suitable precautions to prevent loss of heat to other bodies than the liquid in the beaker, the whole heat made latent in the vapor formed in the outer bulb would be recovered at the immersed one.

If a liquid is vaporized by the rapid removal of its vapor, without applying heat at a rate equal to that at which the change of state makes it latent, the temperature of the liquid falls rapidly, and it takes what heat it can from its surroundings. To demonstrate this we can place a small tube containing water inside a larger one containing a small amount of ether. Blowing air through the ether in the outer tube removes its vapor rapidly, and the ether evaporates and cools. Continuing the process for a moment so lowers the temperatures of both liquids that the water freezes. This drop in temperature is explained on the ground that the liquid molecules, which are moving most rapidly, are the first to fly off from the surface. Hence, the average kinetic energy of the molecules left behind is less than the initial average for the liquid. Since the temperature is a function of the average molecular velocity, the liquid is cooled by the evaporation. In the cases of liquids having high latent-heats of vaporization, it is possible by this means to lower the temperature of the vaporizing liquid so far that it will solidify or freeze long before its vaporization is complete.

Again using a sealed double bulb containing only water and its vapor, if the water is all run into one of the bulbs and the other is plunged into a freezing mixture, the vapor will be removed so rapidly from the water surface by condensation in the cold bulb, thereby reducing the vapor-pressure throughout, that the temperature of the water in the exposed bulb will drop to the freezing point and the water will solidify into a cake of ice. This apparatus is called a cryophorus, from the fact that when it is treated in this manner it bears frost.

The points it is desired to make are: (*a*) that vaporization is accomplished only as a result of the application of heat; and (*b*) that the vaporization of a given quantity of a given liquid requires a definite amount of heat, no matter how the other conditions by which the process is surrounded may be altered.

Motor-Fuels Are Heterogeneous Mixtures

All present-day motor-fuels are mixtures of a great number of substances. Each component has, of course, its own particular heat of vaporization and vapor-pressure. Fortunately for both producer and consumer, the vapor-pressures of these mixtures agree very closely with that of the most volatile constituent. The result is that, if we pump enough fuel into an engine, we can get it started on the trace of volatile material in the fuel. Attention is directed to the result with the two kinds of gasoline in the first demonstration.

The specific heats do not vary greatly among fuels or their constituents. Among petroleum fuels latent-heat values decrease with decreasing volatility. Naturally, the less volatile the fuel components are, the higher will be the temperature at which a given relative vapor-density is attained. Assuming that our interest is in the maintenance of a given rate of vaporization of the fuel, this fact entirely overshadows the advantage that might be expected from a consideration of the latent heats alone. Not only is a greater total heat needed to vaporize a less volatile fuel but less heat is made latent in the process. As the fuel becomes less volatile, the normal intake temperature in an engine will of necessity increase if a given relative vapor-content is to be maintained in the charge. The relations of the pertinent physical constants of the majority of the paraffin petroleum compounds appearing in motor fuels are shown in detail in Figs. 2 and 3.

Fig. 2 is a grouping of physical constants referred to molecular weight, for the members of the paraffin series found in our fuels. These values have been taken from the work of C. F. Mabery and A. H. Goldstein.[3] Fig. 3 shows an interesting heat relation developed from the values shown graphically in Fig. 2. The facts here presented are: (*a*) the great increase in total heat of vaporization in the face of lowered values of the latent heat accompanying reduced volatility; and (*b*) the enormous increase in the sensible heat expressed as a fraction of the total heat, as the volatility is reduced. When it is considered that within the last 5 or 6 years the mean characteristics of our motor gasoline have shifted from those of hexane C_6H_{14} to somewhere in the immediate neighborhood of those of octane C_8H_{18}, the reasons for our need to apply more heat and to maintain higher charge temperatures are explained somewhat by this graph.

Vaporization in the Engine Intake

To apply the facts that have appeared thus far, and to relate them to the conditions in the intake system of an engine, it is necessary only to examine those peculiarities of the intake that modify the rate of evolution of vapor. The simplest way to do this is to follow the fuel from the carbureter nozzle to the cylinders. As soon as the fuel issues into the airstream, some of its molecules fly off into the gas space, in numbers depending upon the temperature of the fuel. Since the partial pressure due to the fuel vapor is zero, or at least is negligibly small, the number of molecules leaving a unit surface in a unit

[3] See *American Chemical Journal*, vol. 28, pp. 66 and 165.

time is no doubt the maximum possible at that temperature of the liquid.

Let us imagine two fuel-sprayers passing fuel at the same rate and temperature, the only difference between them being that one causes twice the initial liquid-surface exposure as the other. It is evident that twice the number of molecules will fly off into the gas space in one case as in the other. A very important expedient for accelerating the rate of vaporization is indicated here; and it can be stated that the initial rate of vaporization is directly proportional to the extent of the liquid surface exposed. But, as we have seen, heat disappears in doing the work of vaporization. Thus, from the first instant of issuance into the air-filled space, the temperature of the liquid will fall unless a suitable quantity of heat is imparted to it.

If twice the number of molecules fly off initially from the liquid discharged from one fuel sprayer as from the other, it is evident that a suitable quantity of heat to maintain the temperature of the liquid must be imparted at twice the rate in one case as in the other. The quantity of heat used in vaporizing a given amount of liquid fuel is the same in both cases; but, to maintain constancy of temperature, the rate of heat supply must be in direct ratio with the area of the liquid surface exposed. If heat is not supplied to maintain the temperature, it follows that the initial rate of heat loss from the liquid fuel will be twice as great in the case of the sprayer in which twice the surface is exposed. Since the temperature varies as the heat-content of a body, it is seen that the initial rate of temperature drop in the liquid fuel is in direct proportion to the surface exposed.

Allowing the temperature to fall in this way lowers the saturation-pressure and so causes a nearer approach to equilibrium between vapor and liquid, with a consequent reduction in rate of vaporization. Considering the discharges of the two sprayers, if no external heat is applied it is evident that equal drops in temperature mean equal amounts of fuel vaporized, and that ultimately the temperatures and the vapor-densities will be the same. But there is no time for dalliance in the intake of a modern engine; and so it is that, within the time available in the system, greater extension of the fuel surface causes a greater lowering of the temperature and also accounts for a greater vapor-density in the charge.

Let it now be supposed that the initial rate of vaporization is insufficient, and that heat is applied in excess of that needed to maintain the initial fuel-temperature. The saturation-pressures are then raised, the equilibrium condition is deferred and the rate of vaporization is accelerated in consequence. But the rate of passage of heat into a body varies directly with its exposed surface; and the rate of vaporization varies directly with the rate at which heat is taken on. Thus it is seen that the sustained rate of vaporization is directly proportional to the liquid surface exposed.

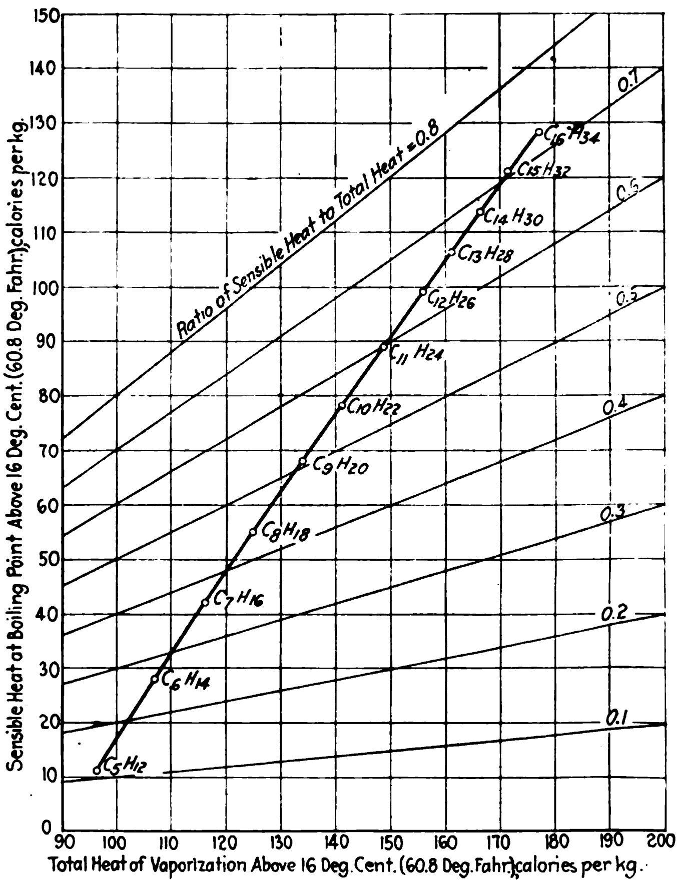

FIG. 3—RELATIONSHIP OF THE TOTAL HEAT OF VAPORIZATION TO THE SENSIBLE HEAT

In the case of an engine and intake system arranged to apply such an excess of heat that the final charge-temperature is well above the initial temperature of the fuel, as in a typical modern engine, the direct result of an appreciable extension of the liquid surface will be to increase the vapor-density of the charge, or to lower the charge-temperature, or to accomplish both these things simultaneously.

EFFECT OF INTAKE PRESSURE

The general effect of pressure on vaporization will not require discussion at this time. However, *changes of pressure in the engine intake are of extreme importance*, particularly those following manipulation of the throttle.

Let us suppose an engine to be throttled, as in driving a car within the speed limit on a smooth level road. The manifold pressure is then approximately one-half an

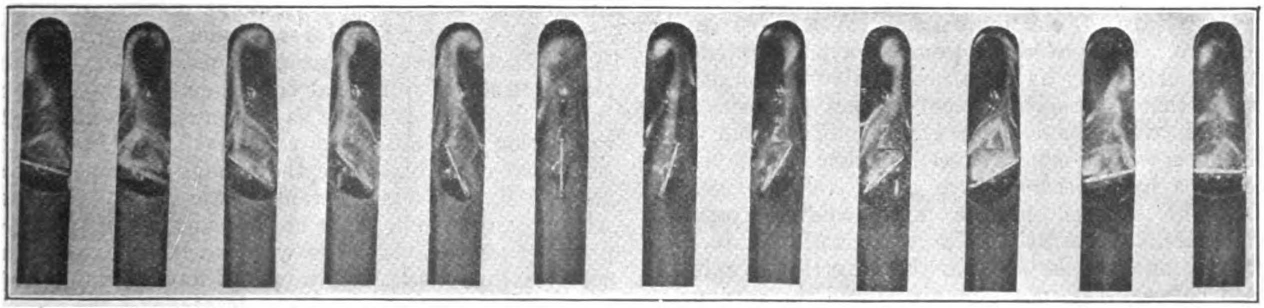

FIG. 4—SERIES OF PHOTOGRAPHS SHOWING THE ACTION THAT TAKES PLACE IN THE INTAKE-MANIFOLD AT AND ABOVE THE THROTTLE AS ITS POSITION IS CHANGED FROM CLOSED TO FULL OPEN AND BACK TO CLOSED

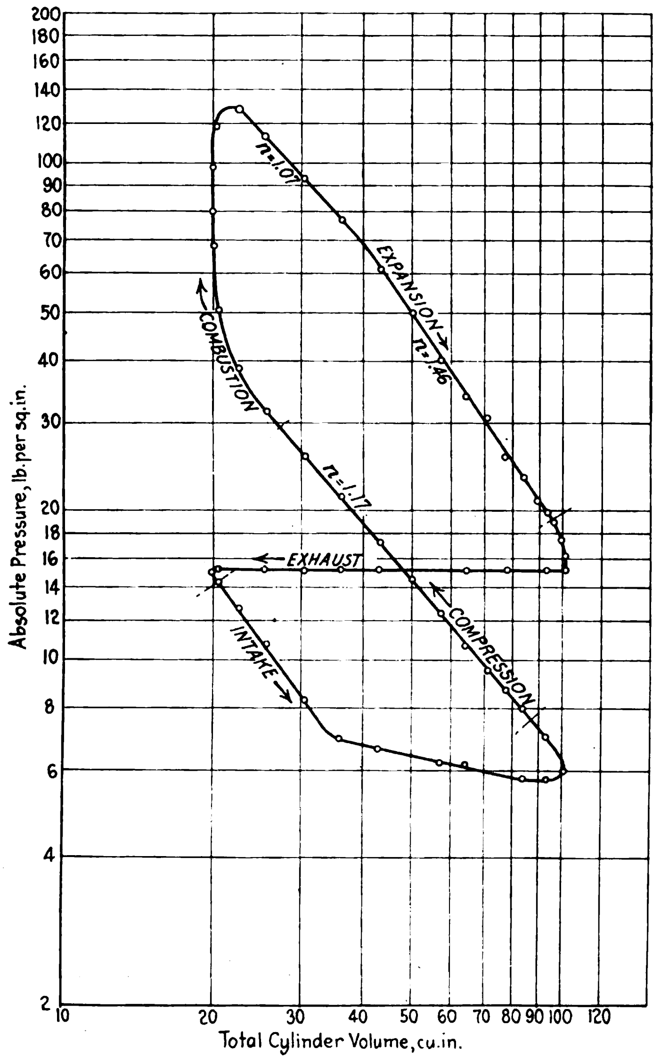

FIG. 5—TYPICAL PART-THROTTLE INDICATOR-CARD PLOTTED ON LOGARITHMIC COORDINATES

atmosphere. Only a small amount of air is passing to the engine. Under these conditions the relative vaporization rate will be substantially at its maximum, since the pumping strokes of the engine withdraw the vapor from the liquid surface as fast as it is formed, and maintain a low total pressure of which the vapor-pressure is only a part. The result is that relatively little liquid exists in the manifold, and its walls are as nearly dry as they ever can be.

Now let us open the throttle somewhat. The immediate results are rise of pressure and the passage of more air and more fuel. From a vaporization standpoint, the circumstances are: (*a*) increased density of the fuel vapor at the liquid surface; (*b*) reduced relative surface exposure of the liquid; and (*c*) lowered intake temperature. Each of these items is a powerful deterrent to vaporization; and the net result of opening the throttle is that the relative vapor-content of the charge, upon which we depend for regular ignition and good performance, will be much reduced until such time as the intake walls have become more extensively wetted, increasing the exposed surface of the liquid fuel and thereby partially restoring the vaporization rate. In the conventional intake system, the throttle-valve is interposed between the intake-manifold and the sprayer or nozzle of the carbureter. As the fuel passes the nearly closed throttle it is subjected to extensive spraying, and therefore to extension of its surface, which spraying is by no means equalled by that at the ordinary nozzle under any condition of operation. Attention is directed to Fig. 4 which gives a very good idea of what happens at the throttle of a carbureter at its several positions. This photograph was published in my paper on Intake Flow in Manifolds and Cylinders.[3] Attention is called to the marked differences in agitation and churning-up of the fuel as the throttle position is changed.

If the throttle is now closed to bring us again within the speed limit we accomplish: (*a*) reduction of density of vapor at the liquid surface; (*b*) extension of the exposed liquid surface; and (*c*) rise of intake temperature. Each of these is a powerful accelerator of vaporization; and, opposed to them, the heavy accumulation of liquid on the manifold walls does not stay there very long, but goes into the cylinders and causes excessive enrichment.

VAPORIZATION IN THE CYLINDERS

The question as to what fraction of the fuel supply can enter the cylinders as liquid and still be completely burned is almost impossible to answer. No doubt the size of the admissible unvaporized fraction varies chiefly with the fuel, but it varies also with the extent of its mechanical division, with the temperatures of the walls it falls upon, with the temperatures to which it is raised and with the pressures to which it is subjected during compression and the early part of the combustion.

The term polymerization is defined as the act or process of changing one substance into another which has the same composition but a different molecular weight. The polymerization products of petroleum compounds attain extraordinary molecular weights; the very heavy compounds are resinous and tarry, and their combustion is almost impossible. The increased rate of carbon-deposit formation that has been generally experienced during the past year is usually explained on the ground of high polymerization of our later fuels.

If the fuel is one that does not break-down, and does not polymerize, it is entirely possible that the liquid surviving the compression will vaporize or may attain its critical temperature during the combustion; and will then burn rather well along in the expansion stroke, appearing on the card as a sustained combustion, or even an after-burn. Most cards from automobile engines, when plotted on logarithmic pressure-volume coordinates, show an appreciable length of the early expansion-line to have a slope close to unity, as in Fig. 5. This can mean only that the combustion is continuing at a rate sufficient to maintain the temperature of the gases at a nearly constant value, in spite of the expansion.

Fig. 5 shows a typical part-throttle indicator-card plotted on logarithmic coordinates. The data from which this plotting was made were obtained at the Bureau of Standards, with the point-to-point diaphragm-type indicator developed at the Bureau under the direction of Dr. H. C. Dickinson. The graph gives a good idea of the closeness of the work that can be done with this form of indicator. In such a plot as this one, the exponent in the expression for adiabatic compression or expansion is scaled off directly. Note the change in slope of the expansion line, to which attention was directed previously. Also note the small value of the exponent n for the compression stroke. If there were no heat losses from the compressing gases, this n would have a value of 1.41. But, of course, there are heat losses, and the major one is, no doubt, the heat loss to the fuel that is being vaporized during this part of the cycle.

[3] See THE JOURNAL, March 1921, p. 282.

With our present fuels, it is well to depend as little as possible upon the cylinder heat and temperatures to complete the vaporization of the fuel. Most of our fuels break-down too easily, and many of them polymerize extensively. Both these things cause rapid fouling of the combustion spaces, and particularly of the intake-valve heads, since the latter are thoroughly wetted by unvaporized fuel.

A further argument is that the greater the dependence upon vaporization in the cylinders, the greater will be the liquid mass in the intake-manifold to be distributed among the cylinders and, obviously, the more difficult it will be to secure its equal distribution. Small errors in the distribution of the liquid in the manifold represent very gross ones in mixture proportions within the cylinders. But, no matter how much liquid fuel we may be willing to have in the intake passages and cylinders, there always must be sufficient vapor in the charge at the time of ignition to support combustion, or the engine will not operate. Bearing in mind the valuable effects of (*a*) extension of liquid surface; (*b*) rise of liquid temperature; and (*c*) reduction of vapor-density at the liquid surface, and having considered the conditions in the intake passages, it is interesting and useful to examine the interior of the cylinder.

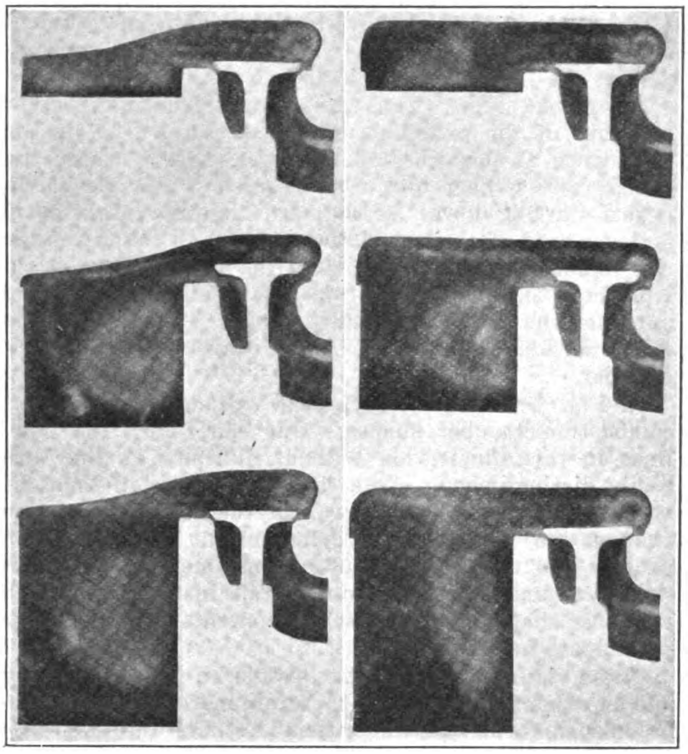

FIG. 7—INTAKE FLOW IN TWO FORMS OF L-HEAD COMBUSTION SPACE

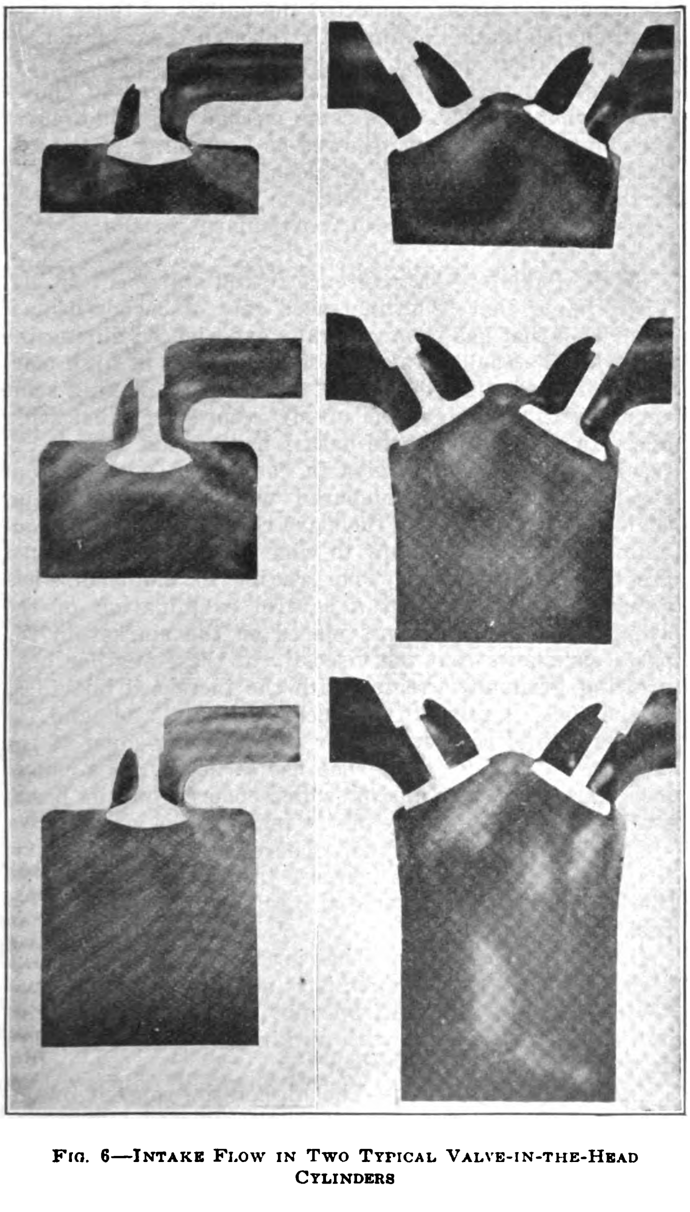

FIG. 6—INTAKE FLOW IN TWO TYPICAL VALVE-IN-THE-HEAD CYLINDERS

In the intake, the fuel is sprayed at the carbureter nozzle, churned-up at the throttle and spread over the passage walls to increase its surface. This increases not only the area from which vapor can emanate but also the area through which heat can pass to the liquid to maintain or raise its temperature. Furthermore, at the same time the fuel is being spread out, as it were, its vapor is being swept away from it, accelerating its vaporization and increasing its ability to absorb heat by lowering its temperature.

Referring again to Fig. 4, the throttle is here taken through a complete cycle of 180 deg. Notice the differences in agitation, churning-up and, therefore, extension of liquid surface among these several throttle positions. It is perfectly obvious that the greatest spraying effect is obtained with the least throttle-opening.

By the time the liquid that is left gets into the cylinders, the density of the vapor in contact with it is greater than was the case in the intake passage. The sources of heat supply with which it now comes into contact are greater and may have higher temperatures than those of the intake. For this reason, vaporization may progress right up to and even after ignition. No doubt it does so under some conditions of operation, no matter what the fuel; and perhaps it does so under all conditions of operation with some fuels. However that may be, if the characteristics of the fuel and the temperature gradients and the pressures that exist permit further vaporization, the only way to carry it forward at the highest possible rate is to extend the liquid surface, just as in the intake. The greatest effect of this sort must follow the highest possible rate of homogeneous and therefore orderly internal motion of the charge. This motion is described as orderly turbulence. That the turbulence in the intake may be reasonably orderly and at the same time high, is clear.

Let us look into a few cylinders, as shown in Figs. 6 and 7. In Fig. 6, we have two valve-in-head cylinders of different design. On the right is something approximating the Liberty cylinder, and on the left an ordinary valve-in-head type. The three stages shown represent positions of the piston on the intake stroke. Note the turbulence or flow-lines in the upper views, where the intake-valve is open only a small amount and the piston is just starting down. In the next stage the intake-valve is wider open, the piston farther down. In the last stage the intake is nearly closed and the piston clear down. Note how the turbulence or internal motion fades as we approach the end of the intake stroke. This condition is even more marked in the more conventional form of cylinder.

In Fig. 7 we have two L-head cylinders of different combustion-chamber shapes. One can follow the flow-lines in this illustration without difficulty as they are easily distinguished. Note that the change in slope, as we may call it, of the top wall of the combustion space changes the volume of the cylinder that is swept by an orderly and therefore useful internal motion. The turbulent volume is flattened out in one instance but, with the other shape, nearly the whole contents of the cylinder participate in the motion.

These views show that the turbulence is more orderly and more homogeneous, and therefore more active and useful, in some forms of cylinder than in others. But in no case does the cylinder turbulence approach that of the intake. High cylinder-turbulence is a desirable thing and should be realized to the utmost in all engines; but, because of the limitations imposed upon its usefulness by fuel characteristics and by the values and distributions of the cylinder temperatures, it cannot be greatly or consistently relied upon to build up a usable vapor-content in the charge.

THE DISCUSSION

CHAIRMAN H. L. HORNING:—Will Mr. Tice repeat the partial-pressure demonstration with the barometer tube?

P. S. TICE:—The demonstration is made with a tube having within it an unoccupied space; that is, it is a normal barometer. The introduction of a liquid into the tube fills the space with vapor and, because of the temperature at which the liquid exists, the vapor has a certain pressure, which is represented by a fall in a mercury column. This fall is read and noted. When the contents have been expelled and the tube sealed, the vapor-pressure tube becomes a barometer also. By keeping the leveling bulb down and using the apparatus as a barometer, air is admitted which raises the pressure. The change in the height of the mercury column represents the increase of pressure in the tube, due to the air. Pouring some of the same liquid used in the barometer tube into the vapor-pressure tube by means of a funnel and then readjusting the leveling bulb so that the air and the vapor together occupy the same volume that the air alone occupied before the introduction of the liquid, it is found that the change in pressure due to the vapor is the same both in the barometer tube having no air and in the vapor-pressure tube having air.

CHAIRMAN HORNING:—If another liquid were introduced, there would be a still greater drop; and the introduction of still another liquid would cause another drop. The fundamental lesson in this is that there are the same number of molecules in 1 cu. in. of any gas at the same temperature and pressure. That is a very important law. If we continue to introduce more liquid, the pressure continues to rise because the pressure is controlled by the number of molecular bombardments there are in a unit of time, and pressure refers to how fast the molecules are going. In this demonstration, as soon as heat was introduced the velocity began to increase, and the heat absorption appeared in the velocity of the particles, which were thrown into a vapor state. When they cooled down, what the cooling down meant was that they had come into touch with something that was moving rather slowly and that this had taken the velocity out of them. But the slow-moving object also gained a little velocity, and finally it had as much velocity as the particles. The liquid state and the gaseous state were then practically the same.

FREDERICK PURDY:—How were the photographs taken of the phenomena inside the cylinder and the manifold?

MR. TICE:—Before undertaking the work, we decided that the flow in both halves of a symmetrical passage, such as the intake passage, would be symmetrical when the halves were divided by a plane; and that we could put such a plane through the passage without destroying the flow-lines. In other words, flows coming through the back of the passage normal to a glass plate would be met by equal flows from the other half of the passage. We split some intake pipes down the middle and cemented them on glass plates, connected the outlets of those pipes to an engine intake, pumped air through them by motoring the engine, sprayed liquid into the air entering this passage, had throttles in place, and proceeded to photograph what we saw. The same method was followed in the cylinders. What we showed as happening in the cylinders was probably not so close to the truth as it is in the case of the intake, because of the unsymmetrical shape of the cylinder-head, particularly in the case of the L-head type.

W. F. PARISH:—In reference to the statement in Mr. Tice's paper that "The increased rate of carbon-deposit formation that has been generally experienced during the past year is usually explained on the ground of high polymerization of our later fuels," I have been making some very careful examinations of my records of lubricating oils. In assembling these data I have found a most extraordinary state of affairs in connection with present engine lubricants as compared with the same grades some years ago. Liquid fuels are becoming heavier gradually but we do not know to what extent the lubricants are becoming heavier. The grading of motor oils has remained the same, but a careful examination of the averages of the lubricants placed on the market in the past year shows that the viscosity of these oils has been growing gradually greater with the increased heaviness of the fuel. I refer to the increased public demand for heavier oils to compensate for the thinning-down of the lubricant in the engines. The use of heavier oils causes more carbon, a large part of which comes from the heavy stocks that are put in the oils to increase the viscosity.

MR. TICE:—That is true. There is evidence to show that a large part of the increase in carbon formation in the last year is attributable also to polymerization of the fuel.

CHAIRMAN HORNING:—That is caused by the increased content of unsaturated fuel, is it not?

MR. TICE:—Somewhat, but not entirely. The things do not go hand-in-hand with all kinds of petroleum; they do with some.

CHAIRMAN HORNING:—The higher the content of unsaturated fuel, the greater the polymerization is apt to be. The unsaturated fuels are the ones that have the two extra hydrogen atoms left out. Some are called

olefins, but as they become heavier, they have a tendency to break in two and make a more complicated substance. They do just the opposite to cracking; the cracking breaks-down into the simpler system and the polymerization steps-up into the more complicated. Anyone who has had an engine with the intake opening before dead center will know what I mean. In such an engine the hot gases are pushed back into the intake passages and the valve will coat-up rapidly with a gummy substance, like varnish. We had an engine go wrong in that way and we found that the timing was wrong, although I tried to prove that the fuel was wrong. The deposit on an intake-valve is due to polymerization. In order to find what causes polymerization, we tested many unsaturated fuels in the laboratory and found both moisture and oxygen to be necessary. We found that an almost imperceptible amount of moisture would start the trouble; but there was no trouble when the fuel was absolutely free from moisture or any oxygen-bearing material.

MR. TICE: — This critical-temperature phenomenon is something with which most of us are unfamiliar; it should be observed in detail.

B. STOCKFLETH:—What is the time element for vaporization in the manifold?

MR. TICE:—The time available for vaporization in the intake is only the time that is required for the liquid to travel from the carbureter nozzle into the intake. It depends upon the velocity of the fluids in the intake and the length of the intake.

MR. STOCKFLETH:—If a certain amount of kerosene is put into the manifold by the carbureter nozzle, and if there is a heated chamber into which it will pass, it will take a certain time before the fuel will become a dry gas?

MR. TICE:—Yes. The time required for that to be accomplished depends upon the rate at which the temperature is raised, the extent to which the fuel is spread out, the area of the surface and the rate at which the vapor is taken away from the liquid. The more useful way to consider the rise of temperature of the liquid is to look at it from the point of view of the rate at which we put heat into it, although the controlling factor is actually the temperature of the liquid.

MR. STOCKFLETH:—In an experiment I am making at present, instead of using the ordinary carbureter-nozzle that creates a fog in the manifold, I poured the kerosene through a fixed orifice into various channels or troughs and let it vaporize only as the heat came to it from the exhaust gases surrounding these pockets. In that way I was able to get a very much clearer mixture than I did by having it pass through the carbureter. This was evident when looking through a peep-hole.

MR. TICE:—Yes, there would be a dry mixture under those conditions. We would all like to do it that way, but can you do that at a useful rate?

MR. STOCKFLETH:—I kept the fuel at a constant flow. I was not attempting any great degree of flexibility, but was surprised by the flexibility shown.

MR. TICE:—I mean, could you run an engine by this method and accelerate it? Could you do the necessary things?

MR. STOCKFLETH: — The proposition is only experimental. It shows that the carbureter action is not needed except for flexibility.

MR. TICE:—Exactly. You accomplished the result by doing exactly the same things as in a conventional arrangement, but gave them a little different order of prominence. You cut out practically the extension of surface and did it entirely by sweeping away the vapor and by heating the liquid. Of course, the rate at which you evaporated the fuel was much lower and the weight of fuel evaporated per unit of time was much less than if you had sprayed the fuel.

J. W. STACK:—Has Mr. Tice determined within a reasonable range what vapor-tension is necessary in a fuel in order to have sufficient vapor to turn over an engine at zero atmospheric temperature?

MR. TICE:—I have never attempted to measure that. I do not see what good it would do us to know it. We are limited to certain sets of conditions.

MR. STACK:—In other words, gasoline can be made with a certain percentage of hydrocarbons boiling over below a certain temperature. What is the maximum or the minimum percentage of those light boiling-points necessary to produce combustion to heat the engine at a low temperature?

MR. TICE:—I do not know. I have never attempted to determine that. So long as the fuel has components that will flash at our starting temperatures, we can start by choking enough fuel into the engine. In his booklet[4] on Economic Utilization of Fuel, Prof. C. A. Norman has computed the mixture temperatures at which we can support as vapor the necessary fuel to burn, for fuels of different mixtures.

As stated in his paper on the Condensation Temperatures of Gasoline and Kerosene-Air Mixtures,[5] Prof. R. E. Wilson has made experiments and obtained some data on the temperatures at which different ratios of fuel mixtures can exist and still have all the different fuels in the vapor state. These data are all interesting and useful but they do not tell us how to arrive at that condition. They tell us how low the temperature may go and the fuel still be in condition to burn, but they do not tell us how to reach that condition within the time available. The rate of vaporization is our limiting factor. Professor Wilson emphasizes this point. What we need is a higher rate of vaporization. Professor Norman discusses the desirable height of mixture temperatures, the results of using higher temperatures, the losses to be sustained and the like. If we could devise some compact apparatus that would evolve vapor at a rate to permit us to form the charge and keep it dry, we could work at the temperatures computed or determined experimentally; but, because we cannot usually do that in practice, we necessarily must work at a higher temperature. The interchange of heat ordinarily is not complete by the time we have to burn the liquid. Mr. Stockfleth's experiment is right in that respect. He obtained a drier charge by allowing more time. He took much longer to evaporate the liquid. If we apply such an apparatus as that to an engine, it must be an apparatus involving a considerable time-interval.

MR. STOCKFLETH: — In regard to Professor Wilson's and Professor Norman's determinations of the condensation points of kerosene and of ordinary gasoline, as shown by tests performed recently, I find, by looking through the peep-hole, that even at a temperature of about 110 deg. fahr. above the condensation temperature as given by Professor Wilson, a slight fog is indicated in the manifold. The temperature Professor Wilson gives is the condensation-point for a dry mixture of kerosene as about 220 to 230 deg. fahr. The maximum mixture temperature in my experiment was 195 deg. fahr. and showed a large quantity of fog. How much more heat would be needed to obtain a temperature as high as 110 deg. fahr. or more above the vaporization point as was the case of the 440-deg. end-point gasoline

[4] See Ohio State University Bulletin No. 19.
[5] See THE JOURNAL, November 1921, pp. 313, 314 and 318.

to produce a dry mixture? Would it be necessary to go very much higher for kerosene, to get it above the condensation-point?

MR. TICE:—The vapor temperatures would need to be higher than those given by Professor Wilson or Professor Norman to form a dry charge. Suppose we put liquid fuel in one end of a chamber and draw nothing but vapor from the other end. In its passage through the chamber the fuel is completely vaporized. The vapor outlet is in communication with an air passage. The vapor is saturated; we are vaporizing in a chamber where there is no air. In other words, the vapor-pressure is the whole pressure within it, and the temperature of the liquid must be high enough to support that pressure. When we allow this vapor to pass out and mix with the air, its initial admixture is very incomplete and non-homogeneous. We will get a partial condensation, a liberation of heat, a rise in temperature of the air and a foggy charge; after that we will need to make a further application of heat to dry the charge again because of the imperfections in our means for transferring heat from one substance to the other, or to the liquid particles.

CHAIRMAN HORNING:—Some of the constituents are bound to gather in the form of globules as they go along. As Mr. Tice pointed out, we can burn these globules very efficiently provided they do not become too large. It is difficult to determine just what size is permissible. To obtain a perfectly clear, dry mixture, would require exceedingly high temperatures. Dr. H. C. Dickinson covered that point in his test of the Chalmers engine at the Bureau of Standards, in checking the fuel report that the Society published concerning hot-spots. He observed the temperatures up to 500 deg. fahr. and had liquids running through the manifold even when using a gasoline such as Red Crown. There were some particles that would not stay in the form of vapor at 500 deg. fahr. We would like to have the mixture dry, but there is a point beyond which we do not like to go. I carried on a series of economy tests on one of Mr. Stockfleth's engines for an entire week, to illustrate that. The intake-manifold was entirely surrounded by the exhaust-manifold. The exhaust-manifold was red-hot on the outside; therefore, we had the highest temperature possible.

When Mr. Stockfleth's engine was giving slightly more than 0.570 lb. per b. hp-hr. on gasoline, it would show 0.591 lb. per b. hp-hr. on kerosene. The difference was almost insignificant. During the week's tests the difference in thermal efficiency between gasoline and kerosene was consistently 0.021 lb. per b. hp-hr. The temperatures were high enough to vaporize all the fuel, or at least to get it into such shape that it would do the work. The difference was due merely to some difference in volumetric efficiency, referred to by Mr. Tice. On account of the temperature changes and the vaporization, we got more in with the gasoline than we did with the kerosene; therefore, we obtained a higher thermal efficiency.

MR. TICE:—With the same specific consumption per horsepower-hour, the thermal efficiency is less for kerosene than for gasoline?

CHAIRMAN HORNING:—Kerosene has a higher heat-content than gasoline, and yet we do not get the returns. That is a very important distinction.

MR. TICE:—There is one range of conditions in which we do get an equal or higher thermal efficiency or, in other words, a much less specific fuel-consumption of kerosene than of gasoline; that is, from about one-half load down to the minimum load. It is common to find differences in specific consumption in favor of kerosene of some 6 to 8 per cent.

H. C. GIBSON:—A development in connection with the Knight engine is pertinent to the present discussion. I have contended for many years that we can get more out of fuel by attention to what can be called the alimentary canal of an engine. Engines do not assimilate all the present-day fuel that is fed to them, principally because they do not masticate it for a sufficiently long period.

The inherent rate of vaporization of the fuel is the controlling factor. It has been my belief, and this is now confirmed, that we can evaporate the fuels to a usable condition without introducing liquid fuels into the cylinder. I entirely disagree with any belief in our ability to burn liquid fuel economically, completely or satisfactorily within the cylinder of an automotive engine. I designed and built a car with a 133-in. wheelbase, weighing approximately 5400 lb. loaded, having a rear-axle ratio of 3 10/13 to 1 and a six-cylinder engine of 3¾-in. bore and 6-in. stroke, or 397-cu. in. capacity. It showed remarkable flexibility without changing gears from 2 to 75 m.p.h., without any missing or choking. The fuel consumption on a concrete road at 25 m.p.h. was 1 gal. per 51.4 ton-miles, which is approximately 18.6 miles per gal. According to all the rules, I think an engine of such size in such a car should have shown something less than 12 miles per gal. under ordinary conditions. The crankcase-oil dilution, after 3000 miles of operation without having drained the crankcase, was 8.8 per cent of comparatively heavy distillate, none of which distilled over at less than 410 deg. fahr., the final temperature of the test being 570 deg. fahr. The distillation operation lasted for 90 min., which was time enough to give every opportunity for distilling every drop, including whatever was cracked out of the lubricating oil. It was shown that a very small proportion of true gasoline was in the crankcase oil. Such gasoline as was there comprised only the heaviest ends, and I am rather inclined to doubt that there was any gasoline there at all. Distillation tests run at the standard rate showed no gasoline whatever. The carbon deposit after 3000 miles was really negligible; it could not be measured and could be wiped off. It seemed to have reached a point where it just coated the surfaces, and it did not increase after that first coating had been put on.

The wear of frictional surfaces was not measurable, and there was not a noticeable pound, detonation, or any other disagreeable noise in the engine of any kind under any conditions. In fact, we could not make the engine "ping," although the compression pressure was 88 lb. per sq. in. gage, by three different methods of test, or a total of 103 lb. per sq. in. absolute. It was specifically timed for slow-speed operation, not to exceed in ordinary use 1500 or 1600 r.p.m. All the conditions of that engine are utterly wrong according to general practice, but since they worked out so extraordinarily well they must be worth noting.

This was all attributable to the particular attention paid to and treatment of the alimentary canal of the engine, so to speak. It started at the entrance of the air to the hot oven, which delivered air at a considerable temperature to the carbureter; and such metered mixture of air, gas vapor and liquid as came from the carbureter was inspired into a manifold in which no attempt whatever had been made to maintain the velocity of the gases. In fact, it was just the opposite; the manifold passages were of much greater area than the carbureter-flange passage. It was a plain manifold running straight to each of the cylinders, with a little obstruction to each of the nearer cylinders; that is, more obstruction to cylinders Nos. 3 and 4 than to cylinders Nos. 2 and 5; and a little more obstruction to cylinders Nos. 2 and 5 than to

cylinders Nos. 1 and 6. But the main feature of that manifold was very similar to the one mentioned by Mr. Horning, in which the whole of the manifold was jacketed by exhaust gases. The intention was to get as much area as possible available for the evaporation of the liquids, and special attention was paid to and provision made for the catching or trapping of the liquid particles that would be in the manifold under all conditions with the present fuel, and the cooking of those liquid particles off into a gas. That was accomplished by the use of long ribs in the upper and lower halves of the manifold. The manifold was split throughout the length so as to get a perfectly clean job. The ribs, acting like the banks and the bottom of a river and retarding the movement of the gases through the manifold, had the effect of giving a somewhat circulatory motion to the gases traveling along the lower half and, similarly, to those traveling along the upper half. By centrifugal force, that motion threw out some of the liquids that were in suspension in the airstream, so that they were deposited on the rough surfaces of the ribs and were cooked off.

The temperatures of the mixtures going into the cylinders were comparatively low, say 150 to 180 deg. fahr. We could run that engine on any one cylinder after getting it warmed-up, and, so far as we could judge with good test apparatus, we could get practically the same results with any cylinder working with all the others cut out. That showed pretty good distribution.

F. G. SHOEMAKER:—In manifolds in which an attempt is made to supply the heat of vaporization at a very high rate, assuming that heat can be supplied at a very high rate, is there any chance that the fuel will crack if it is in the presence of all the air that is traveling with sufficient velocity to blow it over the liquid fuel?

MR. TICE:—I will not say that it is cracking or what it is, but we do get a deposit in the intake passages. It never amounts to much in thickness; it is rarely found to be more than 0.001 in. thick. One cannot chip enough of it off to measure its thickness accurately, but we do get a deposit which, so far as we can distinguish, is identical with the accumulation at and around the intake-valve.

MR. SHOEMAKER:—In an intake passage that was ribbed in such a manner that an appreciable depth of liquid fuel was caught and held so that air in passing through the intake would not sweep the vapors off and the liquid fuel would boil, would one be more likely to have cracking?

MR. TICE:—Yes, if the intake were arranged to trap liquid, as Mr. Gibson has suggested. Let us consider what happens when we open and close the throttle? We have these extensive accumulations in the intake pipe and any subsequent reduction of pressure in it will cause a higher rate of evolution of vapor from them. The accumulation of liquid will be at its maximum under open-throttle conditions.

When conditions controlling vaporization in such an intake are at their best, we get an excessively rich mixture. Correspondingly, if we provide these pockets, the mixture will be impoverished during the time the pockets are filling. We are taking liquid out of the mixture and, while going from a small throttle-opening to a large one, we will impoverish the mixture in the cylinders. Then, when we close the throttle, we take the liquid out of the pockets.

MR. SHOEMAKER:—Tests were conducted in connection with that point, at Ottawa Beach, in 1920. The conclusion was reached in the report[6] that the rate of acceleration was increased by increasing the intake temperature. It seemed that the obvious point was overlooked, that the improved acceleration was not due to the increased temperature of the intake gases, but to the smaller amount of liquid fuel in the manifold.

MR. TICE:—Upon opening the throttle, the vaporization rate is spoiled as would be the case if much more liquid were in there but, relatively, it is not spoiled so much.

CHAIRMAN HORNING:—Acceleration is the function of the energy content in the mixture at the moment the spark jumps. Therefore, the more fuel in the form of a vapor at the moment the spark jumps, the greater is the energy released. If a large part of the fuel is in liquid form, it will not be released. Dr. Dickinson's demonstration was beautifully illustrative of this point.

MR. TICE:—Fuel in the liquid state does not burn and that is all there is to it.

MR. SHOEMAKER:—In testing out an oil-pressure regulator, I connected a milk bottle as a trap on the vacuum line to the manifold and noticed that the vapor apparently was rushing back-and-forth into the milk bottle from the manifold. I could not believe that it was coming through a length of 1/4-in. tubing, and came to the conclusion that the action was due to the pulsation in the manifold, which caused the vapor in the bottle to pass through the dew-point. I believe we see some of the same effect in a manifold when we put in glass windows, and that the momentary waves of high pressure passing back-and-forth through the manifold make us think it is fog, whereas it may be a dry mixture. I have had that occur in manifolds that were bone-dry on the walls, and could not get rid of the apparent fog.

MR. TICE:—Regardless of what the throttle position is or what the load on the engine is, there is a pulsation pressure in the intake pipe. With the ordinary four-cylinder engine this pressure will run to about 70 mm. (2.756 in.) of mercury. In a four-cylinder engine it varies by about that amount twice per revolution. The change of the total pressure in the intake represents a corresponding change in vapor-pressure. We get exactly what we must expect, in condensation and re-evaporation of the fuel, every time that pressure changes. This is most clearly to be observed when the engine is running at a low speed. If we have a milk bottle or any chamber on the intake, the pulsation in pressure gradually pumps vapor into the bottle and pumps air out until, finally, we have substantially the same mixture in the bottle as in the intake pipe. A small hole merely delays the action. A larger passage allows it to happen earlier. The pressure is changing twice per revolution in that bottle.

W. H. HOLLISTER:—Mr. Tice showed the different openings of the conventional type of throttle and the turbulence in the manifold. Did he ever experiment with a throttle of a different type that would open from the center with a gradually enlarging aperture? If so, what effect would that have on the turbulence and the wet manifold?

MR. TICE:—That has been done. With a conventional butterfly throttle valve, standing at an angle and being partially open, the greater part of the flow gets past the downstream edge. There is an eddy space behind the throttle where the fluid is revolving. The liquid content of that eddy in an intake will increase to a certain amount but will not become any greater. Liquid is passing into and out of the eddy. With this throttle-valve, except for the content of the eddy, the whole of the passage is being swept by the air.

[6] See THE JOURNAL, September 1920, p. 228.

Suppose we have a throttle that opens up like the iris diaphragm of a camera. The mixture is flowing upward, and we get a symmetrical ring, or doughnut-shaped eddy above the iris. The liquid content of the eddy is relatively greater and the shape does not permit the eddy to elongate; in other words, the eddy remains circular in section, which is the most favorable shape for retaining the maximum amount of liquid. In the butterfly throttle, the eddy includes only a very narrow sectional area and the liquid content of the eddy is less.

In the intake-manifolds I have shown, when the throttle-opening, corresponding to an almost closed iris throttle, was very small and we got a short distance above the throttle plate, the whole cross-sectional area of the pipe was working; but that would not be the case with the iris throttle. We would have then an excessive wall accumulation of liquid and minor eddies clear up the wall until the stream expanded to sweep the walls again.

CHAIRMAN HORNING:—To obtain a distillation diagram, the fuel is put into an Engler flask to which a flame is applied. The temperature of the vapor will assume a definite value which we plot. The curve representing the percentages coming off is known as the distillation curve, which is used generally by the oil industry. If we take Mr. Tice's kerosene and mix it with the very fine unknown substance that he used, some of the heavy particles will come off first; in other words, it does not separate in a well defined way. There is an appreciable quantity of this very heavy end-point in the first distillate that comes off at what is called the initial point. Likewise, when we come to the end-point, we have not yet succeeded in distilling off all this light stuff, although we have been fairly successful. There are some things coming off at the time which we call the end-point, when the material is almost dry. Since temperatures and percentages are involved, a distillation curve is a methematical expression of a large number of complicated physical factors.

The interesting thing that Professor Wilson has shown is that the temperature of condensation of a 12-to-1 mixture is very important. It proves that the temperature at which Red Crown gasoline will start to condense from a 12-to-1 mixture, which is a rather rich mixture, is far below the point at which the first part of the gasoline comes off in distillation. The standard specification calls for an initial temperature of 140-deg. fahr.; therefore the initial point of condensation is about 20 deg. under the initial boiling-point. This statement holds true for all fuels except those made in Oklahoma from very heavy stuff, and from some of the very light stuff when it is possible to get the condensation point above the initial-point.

If we start a liquid to boiling in an Engler flask, we keep taking off and replacing just the amount that comes off; in other words, what comes off first is mostly the lighter ends, but we are always dragging off some of the heavier ends. Finally, we come to the point where there is no change in temperature; in other words, what is coming off and the liquid in the bowl have the same composition. Consider the drops that rush through a manifold at the rate of 150 per sec., as they often do. The outside layers give off their lighter constituents and finally, as the globules get along far enough, the vapor coming off is the same composition as that of the liquid left on the walls.

To show what distillation means and what the end-point means, suppose we take some of this liquid that has the same composition as the stuff coming off, and distill it. We shall find that it is not this heavy fuel at 30 deg.; that it has an initial-point considerably above this, and that the distillation curve runs parallel and ends 40 deg. higher than the end-point of the fuel. This shows that the end-point of the original distillation was merely a mixture with some lighter fuels. That is very important. When we have talked about end-point, we have thought that we were talking about the heaviest constituents in the fuel. The heaviest fuel is there in large quantities, but not exclusively. The curves of the original fuel and the distillation curves of the equilibrium mixture, all run about parallel.

Professor Wilson has worked out a rule by which, if we take the temperature of the 85-per cent point of the fuel and subtract a constant number of degrees, we get a point at which we have ideal condensation of a 12-to-1 mixture. This is a rough rule and a fairly good one; 243 deg. fahr. from the 85-per cent point would give the distillation point.

That is a very important advance in our information, and it is only because our apparatus is imperfect that we must carry temperatures higher than those of the condensation-points. It, therefore, behooves us to make the very best and most skilful use of surface and of temperature distribution, along the line that Mr. Tice has pointed out, in order to keep these temperatures low; because, it is these temperatures that determine volumetric efficiency, the volumetric efficiency determines the weight of air, the weight of air determines the amount of oxygen, and the amount of oxygen determines the amount of fuel that will burn completely. Therefore, the temperature is a very important thing. The weight of air is one of six or eight factors, but it is the most important factor in determining the horsepower that can be developed in an engine.

MR. TICE:—The weight of air determines the power, other things being equal.

CHAIRMAN HORNING: — Another important thing is that, when the mixture is lean, the amount of energy available depends entirely on the amount of fuel. The moment that one gets the correct proportion, the problem changes. When there is not enough air to neutralize the fuel, we have fuel left over. With the fuel left over, if we go just a reasonable distance beyond, the power will still go up, due to the volume of gas in the cylinder, although the excess liquid does not enter into useful combustion.

MR. TICE:—If we have an engine running on a lean mixture and we progressively enrich the mixture, we progressively increase the power until we arrive at a certain point where any further enrichment of the mixture actually increases the amount of air taken into the engine. The brake mean effective pressure curve plotted against the mixture-ratio starts up; by continually enriching the mixture, the curve becomes flat, then goes up, then becomes flat again. Finally, if enrichment be continued, we reach a point where the temperature is not reduced further and the amount of air taken in is not increased.

The relative importance of the conditions discussed as controlling vaporization, as they exist and can be made to exist in the intake system, is best understood if we think in terms of the rate of vaporization. This latter quantity is solely dependent upon the rate at which vapor is removed from the liquid surface and the rate at which heat is imparted to the liquid. Assuming a given degree of turbulence, or internal motion in the gas and vapor-content of the intake, and a given total pressure, the rate of removal of vapor from the liquid surface is relatively a fixed quantity, except as it varies with the rate

of vapor evolution from the liquid. Since this is the case, the general statement can be made that the control of the vaporization rate in the intake is entirely a problem of the ratio of heat input to the quantity of liquid fuel.

Professor Wilson's paper[7] gives a complete and concise statement of the possible conditions of the charge that accompany given intake temperatures. It is based on experimental evidence and is applicable to several typical fuels. But note particularly and do not forget the following comments:

> It must be emphasized at the outset that results secured by such methods represent conditions prevailing at equilibrium. It is possible to obtain "wet" mixtures at temperatures well above the dew-point, because the drops of fuel that are sprayed into the airstream do not have sufficient time to absorb heat and vaporize before the mixture reaches the engine. The results simply indicate what is theoretically obtainable if the time, the degree of atomization, etc., are sufficient to permit equilibrium to be approached fairly closely
>
> These results indicate clearly that any difficulties in securing complete vaporization of the present commercial gasoline are not due to any inherent limitation in the gasoline itself, or to too low manifold temperatures, once the engine is warmed-up. Indeed, where any serious attempt has been made to heat the incoming gases or the intake-manifold, the temperature attained by the airstream is almost invariably far higher than theoretically necessary. Improvement in vaporization apparently should be secured by better atomization, longer times of contact or by throwing the unvaporized particles out of the insulating airstream onto a hot-spot, rather than by raising the temperature of the mixture as a whole with the attendant disadvantages of this

The obvious means of accelerating the vaporization rate is an augmented heat-input following the extension of the liquid surface and an increase of the temperature gradient across that surface. Let us first consider how we can extend the liquid surface. Since, ordinarily, the entire wall of the intake passage is well wetted in any case, there is no opportunity to go farther in this direction without building longer passages or putting alcoves on those we have. Then, too, the extension of surface accomplished by this means could be only relatively small and wholly inadequate.

The feasible alternative then in this direction is extension of the liquid surface by finer division or spraying of the fuel. Considering that the sprayed fuel assumes a globular or spherical form, with any reasonable degree of division, it is useful to examine the relations between surface and mass for different possible dimensions of globules, since, other things being equal, this is the relation that determines the rate at which heat can be taken on. The surface of a sphere is equal to πd^2. Its volume, and therefore its relative mass for any single liquid, is equal to $\pi d^3/6$. The surface is as the square and the volume as the cube of the diameter. Setting surface over volume, the relation or ratio $S/V = 6/d$ is found, which shows that the surface-to-volume ratio is inversely as the diameter. To make the case concrete, let us say that one sprayer divides the liquid into globules having a diameter of 0.030 in., just under 1/32 in., and that another sprayer discharges the same amount of liquid but in globules of 0.002-in. diameter. Both of these diameters are entirely reasonable. The smaller globules are 1/15 the diameter of the larger, the surface exposure of the same mass of liquid is 15 times as great in the smaller globules and, if the temperature gradient across the surfaces is the same, the smaller globules will *begin* to vaporize at 15 times the rate of the larger. But as vaporization goes on in these two cases with a fixed temperature-gradient, the difference in rate mounts up enormously and will be thousands of times greater for the smaller globules at the time when their last bits change to the vapor state. But there are very serious difficulties in the way of realizing the foregoing. The smaller the globules diffused through the airstream are, the greater the thermal influence of a globule will be upon its neighbors, the more perfectly they will follow the internal motions in the airstream, and the greater the difficulty is in maintaining a fixed or suitable temperature-gradient from a source of heat supply.

If the object sought is a dry or completely vaporized charge, formed in the least possible time, it is necessary largely to prevent diffusion of these small globules into the airstream, and to remove them from the heat-insulating airstream to a place where heat can be taken on nearly equally by each of them, subsequently returning only their vapors to the airstream. The need to separate the globules from the air and to apply heat to them alone follows from the fact that we desire the minimum total heat to be given to the charge. Obviously, if the source of heat supply is accessible to the airstream, the latter will take on heat that will be of little or no use in forwarding vaporization, and will appear as a needlessly high temperature of the charge.

When the fuel is thus separated from the airstream, which it must accompany originally for the sake of metering in the carbureter, the need for a fine division of the liquid increases rather than diminishes, since only by this means can the liquid be distributed with reasonable evenness over the hot surface, and only by this means can the duration of contact be made sufficiently short to prevent any accumulation of the liquid, the undesirable features of which have been discussed earlier.

In an intake system realizing this segregation of very finely divided fuel from the airstream, and its reasonably equal distribution over the surface of the chamber in which it is segregated, heat can be applied almost ad libitum to the walls of the chamber to hasten vaporization, without altering the final charge-temperature. Provided the chamber is neither too small nor insufficiently heated to vaporize the fuel, its size and temperature are without influence upon the temperature of the resulting charge, since the fuel takes on enough heat to vaporize it, and superheating of the vapor is prevented by the presence of and intimate contact with the incoming liquid spray, thus maintaining saturation and equilibrium between the vapor and the liquid.

But even in this case the possible low charge-temperatures discussed by Professor Wilson as supporting the entire fuel in the vapor state are not necessarily realized. When the fuel is separately vaporized and its vapor then mixed with the air, there is a definite temperature-rise in the air following the mixing, depending upon the heat of vaporization of the fuel used, the temperature of the vapor and the mixture-proportion. Therefore, if the entering air has a temperature so low that its value plus this definite temperature-rise upon mixing is less than that required to maintain the fuel as a vapor, some of the fuel will condense. Likewise, if the temperature of the air plus this temperature-rise gives a sum greater than that which maintains all the fuel as vapor, the charge, while entirely gaseous, will unavoidably stand at a temperature higher than that absolutely necessary for dryness.

[7] See The Journal, November 1921, p. 318.

(*Concluded on p.* 322)

Airplane Design and Performance Improvements Since the Armistice

By Lieut. C. N. Monteith, U. S. A.[1] Dayton Section Paper

THE author presents, in outline only, the various features of airplane-development investigation that have been prosecuted. After mentioning the principal types of airplane designed and built shortly before the armistice and the types in service on the battle front at that time, four specific requirements for increasing the speed, the rate of climb and the ability to reach great altitudes are enumerated and commented upon, the further statement being made that an increase in performance can esult from any one or from a combination of all four.

Remarks upon design features are interspersed with the discussion of performance improvements, brief explanations being given of the variable-area and the variable-camber-wing schemes, the idea of having a thick wing-section with trailing and leading edges hinged, and that of modifying the wing-section by making the leading edge a small detachable airfoil that can be shifted. All-metal airplane construction is considered under four specific headings in conclusion.

IN presenting this subject briefly, it will be possible only to outline the various lines of investigation that have been carried on. Many of the improvements brought to the attention of the public within the past 2 years have been the results of work actually accomplished just prior to the armistice. Necessity for economy following the war has slowed up aeronautical research to an appreciable extent, and the production of airplanes in large quantities virtually has ceased. The Gordon Bennett race in France in September 1920 brought out several representative airplanes. With one or two notable exceptions, these were types that were under construction in the respective countries at the close of the war; every airplane entered in this race was a standard pursuit craft, but each had refinements made upon it for the race with the single idea of increasing its speed, which already was high. For instance, the Nieuport that won the race was a stock airplane with "clipped" wings; that is, the wings had less area than those built for the standard airplane. The American Army racer had been adapted to take a 600-hp. Packard engine in place of a 300-hp. Wright-Hispano engine.

It can be said that most of the airplanes that have been attracting attention were designed and built shortly before the armistice. This includes machines such as the Martinsyde Semi-Quaver, the English Nieuport Night Hawk, the French Nieuport No. 29, the Spad Herbemont, the Borel, the Thomas-Morse MB-3 and the Verville VCP. These airplanes have developed high speed, varying from 150 to 175 m.p.h., while those in service at the front at the time of the armistice, such as the English Sopwith Snipe, the French Spad XIII, the Italian SVA and the Fokker D-VII, showed high speeds of from 125 to 145 m.p.h.

Performance

An increase in performance, meaning speed, rate of climb and ability to reach great altitudes, can result from any one or from a combination of all of the following four sources:

(1) Engines of high power and lower weight per horsepower
(2) Reduction of parasite resistance, or the power required to pull exposed struts, wires, wheels and the like through the air
(3) An increase of speed-range; that is, the ratio of extreme high speed to the landing speed
(4) A decrease in structural weight for carrying the same load with the same factor of safety

The increased performance of the new airplanes mentioned above can be traced directly to the development of higher-powered engines of low weight. A comparison of the Spad XIII with the Thomas-Morse MB-3, for instance, shows that the number of wires and struts and other parasitic resistances is proportionately the same for each. In the airplanes in use on the front at the time of the armistice the power of the engines used averaged about 230 hp. With the development of the Wright-Hispano engine, weighing 632 lb. when dry and developing about 325 hp. at 1800 r.p.m., a large increase in power with a small increase in weight was available. High speed varies directly as the cube root of the ratio of power available, other things being equal. The average speed for pursuit machines at the time of the armistice being 135 m.p.h. for 230 hp., and the speed of the new airplanes averaging 150 m.p.h. for 325 hp., it will be seen readily that this relation holds and demonstrates that the increase in engine power is largely responsible for the increase in performance.

Design Features

Some German designers were endeavoring to increase the performance of their airplanes by the elimination of parasite resistance, making the wings thick enough to brace them internally and approximately as efficient as the thin wings. Professor Junkers and Herr Fokker did a great amount of work on this. The internally braced wing appeared in the Fokker triplane late in 1917, followed later by the Fokker types D-VI and D-VII biplanes. These were almost identical airplanes but had different engines. Dr. Junkers went even further and developed a single-seater fighting airplane similar to his passenger-carrying machine which is so well known in this Country, but only four of these were in commission at the time of the armistice, and they had inconsiderable service at the front.

A comparison of four airplanes will demonstrate clearly what is meant by the elimination of parasite resistance. Let us consider in the order stated the Curtiss JN-4D, the DeH-4, the Fokker biplane and the Junkers monoplane. The German experimenters have done much work on thick high-lift wing-sections, adapted to internally braced wings, but the English and French apparently do not favor this type of design. Their latest designs adhere closely to the conventional biplane having the strut and wire interplane bracing. The French de-

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signers have not, it seems, departed very much from standard practice, although some of their new designs vary somewhat from the standard arrangement with respect to struts and wires. American designers have not adhered to any one style, and the thick internally braced wing, the well known thin wings that require external struts and wires and a combination of both are appearing in new designs.

In connection with the internally braced wing, two very good sections have been developed during the past year by V. E. Clark, now chief engineer of the aeronautical department of the General Motors Corporation. The RAF-15, a thin section developed by the English, has been for some time the best section for ordinary thin-wing construction, and is generally used as a basis for comparison. The two sections designed by V. E. Clark, known as the USA-27 and the USA-27-C, compare very favorably with the RAF-15, as is shown approximately in Table 1. In this table the RAF-15 is taken as the standard.

TABLE 1—COMPARISON OF THE STANDARD THIN WING-SECTION WITH TWO THICK WING-SECTIONS

	Wing-Section		
	RAF-15	USA-27	USA-27-C
Efficiency at High Speeds	100	72	87.5
Efficiency at Cruising Speeds	100	81	86.0
Efficiency at Climbing Speeds	100	93	95.0
Front-Spar Depth	100	152	220.0
Rear-Spar Depth	100	184	256.0

Both of these sections are suitable for internal bracing, and are being used now by various designers on airplanes that give promise of being superior to any yet produced for the military or commercial purposes for which they are designed. The Cloudster, an airplane designed and built by the Davis-Douglas Co., of Los Angeles, Cal., has one Liberty engine and uses the USA-27 wing section. It is expected to carry fuel enough for a continuous transcontinental flight.

The development of the thick wing has made possible the design of large monoplanes in which the wings are thick enough to house the engines, thus reducing further the parasite resistance on airplanes mounting more than one engine. A recent German commercial airplane is a large thick-wing monoplane with four engines mounted in the leading edge of the wing.

The third method of improving performance is the increase in the speed-range. For airplanes that have a constant area and a fixed wing-section, the ratio of high to landing speed rarely exceeds 3 to 1, and in most cases it is nearer 2½ to 1; that is, if the speed range is 3 to 1 and the landing speed is 50 m.p.h., the high speed will be generally not more than 150 m.p.h. Two ideas for increasing the speed-range have been worked on for some time, but there has been no practical solution of either until recently. One is the variable-area wing; the other is the variable-camber wing.

Variable-Area and Variable-Camber Wings

The most notable of the variable-area-wing airplanes has been developed by a French designer. The wing is constructed so that large flat plates are carried under the leading and trailing edges, forming the bottom of the wing. These plates are extended when taking-off or landing, forming a wing of considerably increased area. The normal area is doubled in this particular airplane; consequently it supports the same weight at a much lower speed. The plates are drawn in for high speed and form a very efficient wing of small area. Results of complete trials on this airplane are not available, but it is understood that it was not entirely successful due to trouble with the operating mechanism. A satisfactory speed-range of from 37 to 125 m.p.h. was obtained; a ratio of 3.4 to 1.

Many so-called designers submit ideas for variable-area wings but, before building an airplane according to their ideas, it is necessary to determine whether the actual gain in lift afforded by their devices more than offsets the additional weight and complication of the apparatus necessary to control it safely. The idea of the variable-camber wing consists in changing the section of the wing so that it can be made to give a high lift for take-off or landing, and then change to a section that is very efficient at small angles of incidence for high speed. The matter of the extra weight of the operating mechanism versus the increased lift is again the determining factor in the feasibility of the idea. The most successful of the designs so far proposed is the racer built by the Dayton-Wright company for the Gordon Bennett race. In this case the leading and trailing edges of the wing were hinged so that they could be lowered for landing or taking-off, thus forming a high-cambered wing with a high lift-coefficient. They were raised for high speed so that the wing was comparatively flat, and efficient at high speed; also, the parasite resistance was decreased for high speed by the use of a retractable landing-gear.

Glenn L. Martin, an American designer, has developed recently a thick wing-section with the trailing edge hinged at a point 70 per cent of the way back on the chord, and the leading edge hinged at 15 per cent. With this combination and with the hinged flap down at a large angle, he has obtained a lift coefficient double that of the RAF-15; and, with the flaps set up at small angles so as to streamline the section, an efficiency is reached at high speed that is about 20 per cent greater than that of the RAF-15. We have every reason to believe that the tests are reliable and, unless there is some structural consideration that will bring the weight of the operating mechanism up to a prohibitive figure, this new section will permit a greatly improved performance.

The Handley-Page company, of England, has done a considerable amount of experimental work with an idea not altogether new but not so well known among designers as variable-area and variable-camber wings. It is to modify the main wing-section by making the leading edge a small detachable airfoil, shifting this small wing a few inches forward of the main wing by using a series of hinged brackets, or fitting it closely in place as the leading edge of the wing, as desired. The small wing is pushed forward for taking-off or landing, which gives the effect of an increase in area and improved lifting power; for high speed the small wing is brought back to its position as leading edge of the main wing, which provides a smaller area in conjunction with an efficient high-speed wing. Several modifications of this idea have been experimented with and flight tests have been made but, to date, we have received no reliable information as to the details of the device. These three devices and their modifications for improving the speed-range are still in the experimental stage, and have not been incorporated as yet in military types.

The fourth method, that of improving airplane performance by a decrease in total weight, has progressed steadily. The decrease in engine weight lies with the engine designer, but the structural weight is not reduced so easily. It requires long and careful research and tests to effect a combination of efficient design and

the most suitable materials. The three materials that have been most investigated are plywood, duralumin and alloy steels. Excellent progress on rib and spar design has been made and wing structures averaging 1.0 to 1.2 lb. per sq. ft. are easily attainable, even for internal bracing, but since the increased factors of safety that are required now for military airplanes require heavier structures, the two have practically offset each other.

All-Metal Construction

Metal construction in aircraft is not a postwar development. Ever since the first airplanes were built designers have been making attempts to use metal in various parts of their structure. Only recently, however, has the development of the high-strength alloy steels and duralumin made possible the design of an airplane structure that compares favorably, weight for weight, with the wood and wire construction that was formerly employed.

Metal construction has the following distinct advantages:

(1) The designer is assured of an homogeneous material, particularly when steel is used
(2) Metal is not subject to such variations in strength or size with changing atmospheric conditions as wood is; consequently, it is better for working under different climatic conditions. From the standpoint of war materials, metal is better for storage
(3) The use of metallic covering on the wings and the body gives a much longer life to the average airplane, and does not require the care in handling that a fabric covering requires
(4) With efficient design and planning, metal construction for production work can be turned out more cheaply than wood and wire construction

The Fokker type of fuselage, made of welded steel-tubing, and the Breguet fuselage, made of duralumin tubing and steel fittings, were the outstanding departures from wood construction during the war. Toward its close, however, the Junker duralumin construction made its appearance.

Since the close of the war all countries have been investigating the question of metal construction. The Germans have built some large flying-boats and land machines for passenger carrying, using duralumin throughout except for some of the more important members which are made of steel. Fokker, however, is using wooden wings, having plywood covering in his commercial machine. The French designers have not taken up metal-construction development very extensively, but the English have been doing considerable work on it. Their experimental stations have investigated a large variety of steel spars and wing constructions and obtained some structures that appear to be simple, light and effective. The Short company has built an entire biplane, including wings and tail surfaces, of duralumin. Apparently it is a success. The Vickers company has done some extensive work with duralumin, while the Boulton and Paul company is handling a large amount of metal construction. One of the English designers states that he has "reached the conclusion that for airplanes exceeding 2000-lb. gross weight metal construction is superior from every angle." It can be made lighter, stronger and at less cost than wood, and is more durable under all conditions. A large part of the design of metal airplanes is composed of efforts to simplify and reduce to a minimum the operations necessary to obtain the required parts and assemble them.

In the United States metal construction has been confined to experimental work. To date nothing has been done toward putting an all-metal airplane into production. Several airplanes of all-metal construction are now under consideration by the Air Service. They embody entirely new features as regards the construction of ribs, spars and coverings, and give promise of being distinct advances over anything now being built in Europe.

So far as actual construction is concerned, little has been done since the armistice that represents any appreciable advance over the work that was completed up to that time. The experimental stations have, however, made considerable progress in the investigation of the four methods of improving performance, and it is believed that, when the necessity for the strictest economy in Government expenditures shall have been obviated, larger appropriations will be made and the construction of aircraft carried along on a scale that their importance warrants.

VAPORIZATION OF MOTOR-FUELS

(*Concluded from p.* 319)

Average gasoline, when so handled in the intake, causes the charge-temperature to stand at substantially 60 deg. fahr. above that of the entering air, for a 15-to-1 mixture. Thus, to realize Professor Wilson's minimum dry-charge temperature, the entering air will be required to have a temperature of about 33 deg. fahr. Since the temperature of the entering air is almost always higher than this, such an intake system can be taken to give invariably dry charges to the engine.

Professer Wilson's dry-charge temperature for a 15-to-1 kerosene mixture at about atmospheric pressure is given as 230 deg. fahr. In the intake system just discussed, the temperature rise with average kerosene is 125 deg. fahr. The temperature of the entering air would have to be about 105 deg. fahr. to give dryness. Ordinarily, this latter temperature is much less than 105 deg. fahr., with the result that some of the kerosene vapor is condensed upon mixing with the air. Professor Wilson's charts show to what extent this condensation will occur with different temperatures of the entering air.

Internal-Combustion-Engine Power for Road-Grading Machinery

By C. O. Wold[1]

Minneapolis Section Paper *Illustrated with* Photographs

After dividing road building into three stages the author describes different types of road machine and the methods that apply to their usage, with a view to affording a basis for decision regarding the types of automotive vehicle best suited for furnishing power to these various road-building machines for the different stages of road work.

The subjects of large versus small tractors and the most suitable type of tractor are discussed in some detail, and engine requirements also receive attention. Such subjects as drawbar pull required, the use of multiple-unit road machinery behind one power unit, combination tractor and grader units, general utilization of power units, and the power requirements of concrete road construction also are commented upon at some length.

Road building can be said to divide itself into the three main stages of the construction of the grade, the surfacing of the road and road maintenance. Several methods are involved in building road grades under differing conditions. One is used in covering territory that is fairly level; another, used where the territory is hilly and where filling is required, is called the cut-and-fill method. A road machine of the blade type is used more particularly where the road has level stretches. The blade type of road machine is made in different sizes, usually designated by the length of the blade. For building a grade with one of these machines, shown in Fig. 1, a 10 to 12-ft. blade is, as a rule, the most economical up to a point where the grade is beyond a normal size.

To build a high grade, the elevating grader is used also. It is a machine with a plow on one side and an endless rubber belt, 36 in. wide, that carries the dirt from the plow and delivers it on the opposite side, as shown in Fig. 2, into a dump wagon, which hauls it away. The elevating grader is used commonly in such localities as the Red River valley. The grades are made very high there for two reasons; one is that this section needs the drainage and high grades not only drain the roads but also help to drain the country. A grade higher than the level of the surrounding country is very desirable where the latter often is covered with water. A low grade would be damaged by the standing water; a high grade remains above the water.

Grass or sod is not a desirable material in a grade and most highway commissions maintain that it should be taken out entirely. If sod is plowed from the side of the road and piled up, it creates a levee and prevents drainage, but it is very expensive to haul it away. The next best thing is to put the sod or rubbish into the middle of the grade and bury it where it will do the least harm. Rubbish, grass or sod has a tendency to hold moisture. For that reason it is not wanted in a grade, especially near the surface. Anything that holds or has

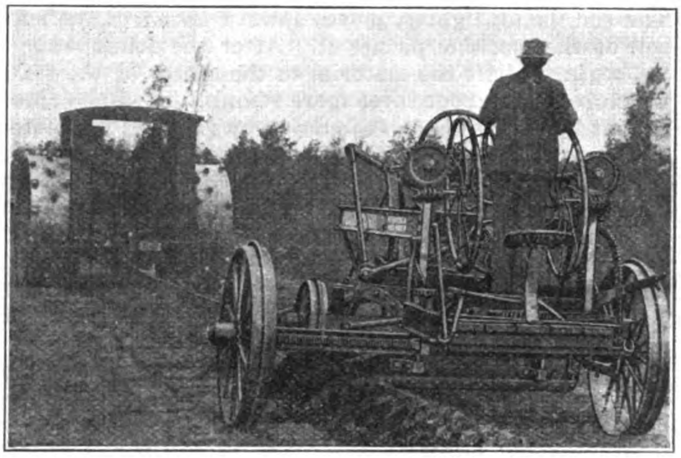

Fig. 1—A 12-Ft. Machine Building a Road in Heavy Gumbo

an attraction for water is detrimental to the road, because we grade a road for the purpose of shedding the water and keeping the grade high and dry. The first cut taken with the blade machine removes the sod. That cut is made as shallow as possible; it is probably only 2 or 3 in. deep or just sufficiently deep to cut soil enough to scour the blade and lift the material toward the center of the road. The usual method is to go up one side and come back on the other side of the road, taking the first furrow where the bottom of the ditch would be and following the grade stakes. After the first furrow, another round is made just inside, shaving off some more sod and bringing it closer to the center of the road. Usually the

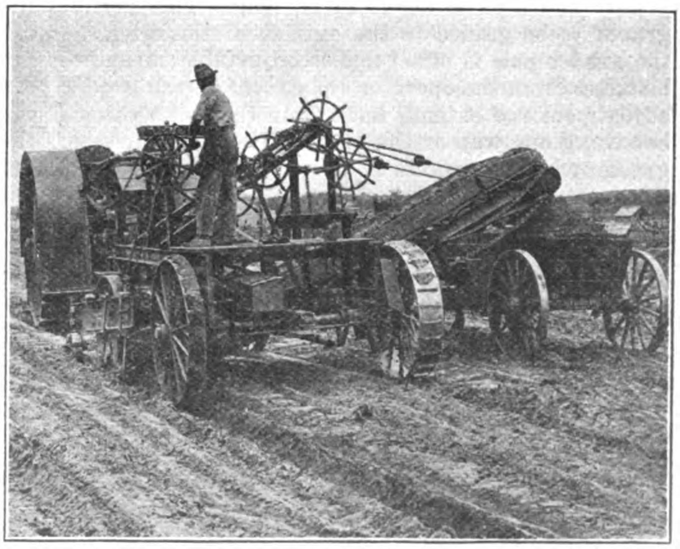

Fig. 2—Elevating Type of Grading Machine Discharging Material Removed from Road into Wagon

[1] Secretary, Russell Grader Mfg. Co., Minneapolis.

roadbed is about 36 ft. wide between the center-lines of the side ditches and two rounds will not bring the material into the center of the road. The roadbed width varies in different localities and States. Iowa, for instance, calls for a wider road than Minnesota.

When these two furrows have been taken off, the machine is run on the outside and shaves off the outer edge or slope that is to incline from the outside into the ditch, often referred to as the back slope of the ditch. Another thin cut of sod is then taken and deposited near the center-line of the ditch that is to be. On the fourth round, the machine cuts 12 or 15 in. deep. It moves in a large volume of earth and with it the windrow of sod. The sod, being lighter, moves inward ahead of the subsoil as the machine pushes it. After the fourth round, we begin to drift the material to the center of the road, which requires about three more rounds. It will be noted that at this stage we drift four furrows toward the center of the road. One reason for this method is to allow the tractor to run in the center of the road as long as possible, where it is on solid ground, before bringing loose dirt into the center. The tractor can go back and forth in the center of the road while it is making these four or almost five rounds, before it travels on the loose soil. This is one reason why it is more practicable to use one machine back of the tractor than two.

These seven rounds result in a moderate grade and a ditch probably about 12 in. deep, depending upon how deep a cut the machine can make and how much power is available. The process is repeated and a second cut is brought in from the ditch and, after five rounds more, the earth has reached the center fully spread. About 12 rounds represent the requirement of an ordinary grade over an ordinary stretch of country, using a 12-ft. machine. This work should make a road having an 18 or 20-in. ditch and a crown of a corresponding height above the grade. With a 10-ft. machine a few more rounds are necessary because the machine cannot drift the material so far each time. With the 12-ft. machine about 25 to 35 hp. is required at the drawbar; the 10-ft. machine requires about 20 to 25 hp. at the drawbar. More than 25 to 35 hp. would be an advantage for the 12-ft. machine, but such extra power might be superfluous from a commercial standpoint since 25 to 35 hp. is sufficient.

The tractor is run in the center of the grade, and the draft is transmitted through the 10 or 11-ft. pole on the grader and a cable about 24 ft. long. This permits the grader to be guided to the outside of the ditch, because the grader pole is offset and is adjustable; usually it can be offset from the operator's platform on the grader. An adjustment rod extends back from the pole so that it can be swung one way or the other. The operator steers the grader where he wants it to travel, regardless of where the tractor travels.

Elevating graders are used wherever possible in cut-and-fill work. In certain localities, steam shovels are used. Steam shovels can be used where a grader cannot, but, where the work is such that an elevating grader can be used, I believe the work can be handled more cheaply with an elevating grader than with a steam shovel. The elevating grader requires about 25 hp. at the drawbar. When it is used to cut down a knoll or hill, the dirt usually is loaded into 1½-cu. yd.-capacity dump wagons and hauled away by teams of horses.

Large and Small Tractors

I have not seen much dump-wagon work done with tractors, but perhaps it could be done to advantage in most cases. On one job in Kansas City, Mo., I saw a 10-ton tractor used with three 3-cu. yd. wagons hooked back of it. These wagons had been built especially for tractor purposes. The tractor cannot very well be used for hauling dirt away because of the necessity of running it into low places and through loose material at the dump, or perhaps to a dumping-off place having a descent of from 5 to 25 ft. The tractor must then go down the grade that is being built and return on the outside of it. This is not very desirable work for a tractor, although the outfit I observed in Kansas City was operating well. One objection to the use of a 10-ton tractor that hauls three large-sized wagons is the length of the cut required. An elevating-grader cut is very short, usually, and the run one way would not be long enough to fill three wagons of large capacity. It could be used to advantage when working on a long cut and a long haul. With a 10-ton tractor, where three wagons can be filled without turning around, such an outfit will replace six to nine teams of horses in removing the dirt to the dump.

The small creeping type of tractor has been used for hauling the dirt wagons and it seems to be practical, but I have not seen a wheeled tractor used for this purpose. The various types of tractor have their proper place in road work, inclusive of caterpillar, four-wheeled and two-wheeled tractors. Any one of these types may have an advantage over the others under certain conditions. Most tractors are somewhat weak for road-building work, especially with respect to their wheels. The ordinary farm tractor is hardly suitable for road work and is not spring-mounted. Several tractor-building companies are making special tractor wheels for road work, but that matter is overlooked generally and the ordinary farm tractor has been used with poor satisfaction. Strong wheels are required for road work; the ordinary farm-tractor wheel is not nearly strong enough.

Tractor Engines and Tractor Types

Tractor engines for road work should be of the heavy-duty type. Most users of road-work tractors think that the engines are far from being of the heavy-duty type. In the West we have observed that many tractors that have been tried for road building are somewhat lacking in power. I have heard it said often that an engine having a large bore and a short stroke is more desirable in a tractor used for this purpose, but I cannot vouch for the truth of this. However, it seems to be the popular opinion. A tractor engine loses 3 per cent of its power for each 1000 ft. of elevation above sea level. In Western territory we are working at altitudes of 3000 to 6000 ft., which causes an appreciable reduction in engine power.

The track-laying type of tractor seems to have gained much favor with road builders and it is especially suitable for cut-and-fill work. Tractors used for road work should be slow-moving. Slow-speed tractors give much better satisfaction and last longer on road jobs than those having higher or variable speeds. With variable speed capacity at his command, the operator often misuses the high speed and, if the tractor weighs 10 tons or more and is put over the road too fast, it destroys the journals within a short time. The large tractors that have had the greatest success in road building have been those having a speed less than 2 m.p.h. Another reason that the track-laying type of tractor has found favor in many places for construction work is that such a tractor can be run safely over small bridges. We do not find that consideration so important in Minnesota, South Dakota, Nebraska or even in Iowa, because the small bridges there are strong enough for heavy threshing rigs to

cross them. But in many districts of the South and in other parts of the Country where such heavy rigs are not transported from place to place, the small bridges would not be strong enough to carry wheeled tractors that are large enough for road construction work.

Road construction involves much scraper work. Work on the approaches to bridges and a considerable amount of the road finishing work must be done with scrapers. A few attempts have been made to use tractors for hauling scrapers. In some instances the small types of tractor have been used for hauling one or two-wheel scrapers behind the tractor with good success, but this usage is still in the experimental stage.

I think it is better to use one large road machine behind a large engine than to use two smaller graders. If two graders are hauled by one engine, one of them must idle to a great extent and the two are a nuisance in turning. In the process I described for bringing in two or three of the outside furrows of sod and waste material to the center of the grade, the second trailing grader is not doing much work. About the most that this would accomplish is to shorten the process by two or three rounds and more time would be lost in trying to do that than could be saved. Another man is needed to operate the second grader, and one does not obtain the desired efficiency. I have seen one grader operated on one side of the road and one on the other side, simultaneously, but this is inefficient because the graders will interfere with each other at the center of the road. The idea of using two tractors in front of one grader has been advocated and I believe it is better than that of using two graders behind one tractor.

Construction work is only a short job when compared to that of road maintenance. When the building of the grade is completed, there is very little more of that kind of work to do, unless it be to change the road or straighten it. If the two-tractor plan of road construction can be used, and I see no reason against it, the expense of a large tractor for construction work would be avoided and the small tractors would be available for road maintenance. Many townships and some counties cannot afford to buy tractors and are dependent upon some one else to furnish them.

Some agitation for the building of a grader and tractor in one unit for construction work has developed, but we do not think this can be done with any degree of satisfaction. The expense of combining a large tractor and a large road machine into one unit is not warranted for the purpose of construction work only. It would have only a limited use during the building of a grade, and an expensive machine of that kind entails a loss during the time it cannot be used. Another bad feature of a combined grader and tractor is that the grader must travel in the ditch and that is no place for a tractor. It cannot get proper traction and the tractor must work on an incline, causing excessive wear on the bearings. The tractor works better in the road.

General Usage

Gravel or crushed rock is commonly used in surfacing a road. Minnesota has much gravel and good road material that can be taken out of a bank almost anywhere and put upon the road, but in other sections road-surfacing material must be shipped in. Crushed rock and shipped material must be placed in a loading bin and hauled from there to the road. Tractors with trailers and motor trucks are being used for such hauling; it is an open question which will prove to be the most practical. Tractors used for this purpose must be equipped with wide wheels so that they will not injure the roads in hauling material to that part of the road which is being surfaced. This is essential. In surfacing roads with gravel or rock, it makes little difference whether this work is started from the near or from the far end. If it is started from the near end, the road is greatly worn down before it is completed. If it begins at the far end and the grading is in the direction of the source of gravel supply, the contractor finds that his tractors or conveyances are digging deep ruts and packing the material down in the center. These ruts require from 20 to 25 per cent more gravel than if they did not exist and, although this may not be detrimental to the road, it is more costly. The same thing is equally true regarding the use of tractors or trucks. The 5-ton trucks used for this purpose encounter the same trouble. Most trucks tear up the road too much during hauling. It seems clear that we must come to the use of a smaller rather than a larger truck. A 2½ or 3½-ton truck equipped with pneumatic tires will cause less wear on the road.

A fleet of trucks for hauling gravel probably is more expensive and from that point of view the advantage is with the tractor, but certain contractors have abandoned the tractor and trailer system and installed fleets of trucks. They claim they are benefited thereby. Such a case is reported from one of the Western cities, where a contractor replaced tractors and trailers and has now had trucks in use for about a year. However, he was graveling the streets in the outskirts of the city, and for such use trucks might be more advantageous than tractors and trailers. The experience of a certain contractor in northern Wisconsin is interesting. He hired his graveling outfit. He had been using trucks but found that they cost him too much. They injured the roads when he tried to gravel them from the near end, as he was forced to do, and there was a great amount of repair work. He declared his intention of installing tractors and trailers the next year. This is a matter that must be worked out. Possibly both systems will be used because of the varied conditions; they are competing now for the business, which is still in its infancy. However, more tractor power will be required for surfacing roads than for all of the other work.

Concrete Road Requirements

Much hauling will be required in building what are called permanent roads with a concrete foundation. There seems to be a demand at present for a conveyance to transport the wet concrete from the mixer to the work. The present method is to use a large mixer equipped with a boom and a bucket to bring the concrete from the mixer to the point where it is to be poured. Many contractors are not satisfied with this method, because they think the concrete mixer should not be moved so often. They believe it should be brought a little farther ahead and have the material brought to it as well as transported from it. They often use trucks for bringing the material to the mixer, the concrete being loaded into small tractor conveyances and transported to the work. The smaller types of caterpillar tractor are used also to take the material to the mixer and haul the concrete to the road. There is a large opening for small tractors for this purpose.

Smaller tractors are more useful for road maintenance, but wheeled tractors are used to a larger extent for this purpose. Blade machines of the type shown in Fig. 3, equipped with 5, 6, 7 or 8-ft. blades, are used in maintenance work. A tractor is used with these machines

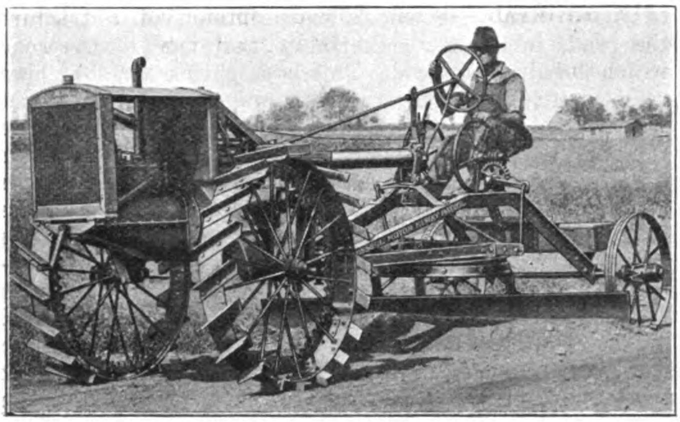

FIG. 3—A 6-FT. MAINTENANCE MACHINE THAT IS EMPLOYED FOR HIGHWAY PATROL DUTY

that varies from 8 to 10 hp. at the drawbar. There is room for another piece of equipment that will run with a somewhat larger blade, a larger machine for more extensive repair work rather than maintenance. This requires from 15 to 18-hp. at the drawbar. There is use also for a lighter machine with a long blade which takes a wider swath for maintenance. Such a machine is shown in Fig. 4. It has a 3-section blade that is adjustable and can be conformed to the contour of the road. Each section is 5 ft. long, making about 15 ft. of blade. It is a machine weighing about 3500 lb. and can be handled with a small tractor having about 15 hp. at the drawbar. The popular demand for road-construction work is for a tractor having from 20 to 25 hp. up to 30 to 35 hp. at the drawbar. For maintenance work the demand is for a tractor having 8 to 10 hp. or one with about 15 hp. at the drawbar.

There seems to be an idea that a combination tractor and grader can be used to good advantage for road maintenance work. We have built a new machine that seems to take well, but we have not tried it out to any great extent. Maintenance work is much different from construction work. In maintenance work the machine naturally travels on top of the grade. The conditions are more favorable; the machine does not need to go into the ditch. Therefore, a tractor and grader combined into one unit may prove desirable.

THE DISCUSSION

E. R. GREER:—You mentioned less than 2 m.p.h. as being probably the best speed for grading purposes and simply specified a certain size of grader for a certain horsepower at that speed. At what speed would you expect it to require that amount of horsepower?

C. O. WOLD:—I draw a distinction between road construction and road maintenance. A tractor doing the heavy work of construction should run slowly; a maximum speed of about 2 to 2½ m.p.h. is fast enough.

R. S. KINKEAD:—What tractor speed should be used for maintenance work?

MR. WOLD:—They try to run 3 to 4 m.p.h.

A MEMBER:—What type of wheel lug is used on a tractor for road work?

MR. WOLD:—For road construction large lugs must be used. Those of the angle type are most popular. For road-maintenance work the lugs must be of a type that will not deface the road too much. A cone-shaped lug is suitable for that purpose, rather than a cleat or angle-type lug.

A MEMBER:—Are solid rubber tires used on tractors?

MR. WOLD:—Rubber-tired wheels would be fine for maintenance work and they are used on certain types of tractor, but the expense might be too great. Some four-wheeled tractors use them.

MR. KINKEAD:—Is it likely that a four-wheel-drive type of tractor will be developed?

MR. WOLD:—That type would have a large field. The cost would be an element. The ordinary user might endure certain disadvantages rather than pay a higher price for a tractor that is built to overcome them. What has been said about a four-wheel-drive truck might apply to a four-wheel-drive tractor. The four-wheel-drive truck is undoubtedly very efficient under certain circumstances; for instance, it will climb out of a gravel pit with a load much more easily than a two-wheel-drive truck.

MR. KINKEAD:—Do you feel that a light tractor is able to get sufficient traction in most cases?

MR. WOLD:—Yes, if it is not used under unfavorable circumstances. It should not be used when the road is muddy; the road should be fairly dry before beginning the work. Under such conditions it apparently has traction enough.

J. A. SKARNES:—What drawbar pull is required for the removing of a certain projected area of soil, for the different kinds of soil? If a 14-in. plow requires an average of about 700 lb. drawbar pull to pull it through the average soil, it will require much less in sandy soil and much more in gumbo.

MR. WOLD:—We are not in the tractor business, and we have not compiled such data. We do not attempt to advise the customer what horsepower he should have to run the grader. We find out what power he has and advise him what grader to use for that particular tractor with that particular power.

MR. SKARNES:—At what angle are road scrapers run so that they will scour most efficiently?

MR. WOLD:—That depends upon the nature of the soil. They are adjusted to an angle that will cause them to scour freely; it would in all cases be less than 45 deg.

MR. SKARNES:—As the angle of the scraper becomes greater that of course means a greater side-draft on the grader.

MR. WOLD:—Yes, but that strain does not come on the tractor; it reacts on the four wheels of the grader. The grader is on the side of the road and the tractor is in the center. The side-draft from the pull of the engine on the grader is very small. At first we had difficulty in convincing users of graders that the side-draft was not too great. We even put an ordinary lumber wagon between the tractor and the grader and hooked the grader

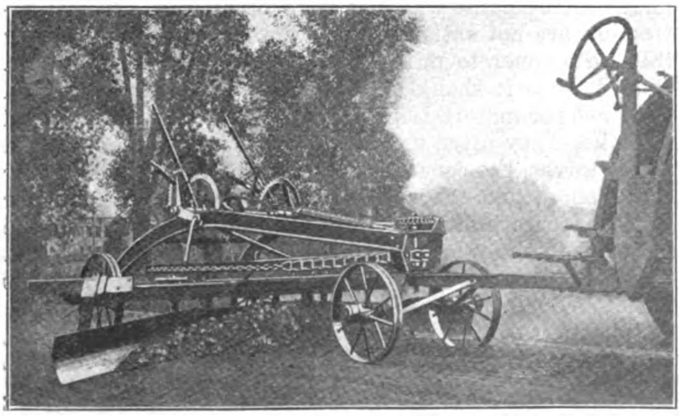

FIG. 4—A 15-FT. MAINTENANCE MACHINE

cable to the rear of the lumber wagon. We could go right along with a good-sized load on the graders without pulling the lumber wagon sidewise. That demonstrated the small amount of side-draft.

A. F. MOYER:—What success have you had in using two engines to pull one road machine?

MR. WOLD:—That has not been tried to any great extent.

MR. MOYER:—As most tractors are equipped with governors it would be necessary to have two engines with exactly identical road speed, or perhaps resort to a special governor adjustment to avoid having one engine take all of the load while the other one idled.

H. J. MAYER:—In maintenance work is it practicable to use more than one grader at a time?

MR. WOLD:—Our former maintenance machine was designed to take about a 15-ft. space on both sides of the road center, covering the entire road in one swath. We discontinued building it because it interfered with road traffic. In very few localities can one go out on one road and return on some other; one generally must go and return on the same road. If one must travel back and forth on the same road, one might as well confine the work to one side going out and to the other side coming back. On one occasion we sold six two-winged machines called planers to certain county commissioners in Iowa for use on heavily traveled roads. The commissioners were not satisfied with stopping occasionally to allow the traffic to pass. They wanted to get the roads fixed and kept on working, which raised a storm of objection because they held up traffic. The same objection applies to using two graders.

J. L. MOWRY:—Regarding the danger to gravel roads because of the use of trucks and tractors, the trouble is with the specifications under which the road is being constructed. If the road is damaged before it is completed, it cannot be expected to stand up afterward. These machines can do the work and the road will stand up if the specifications are all right.

MR. WOLD:—You must remember that the road is in process of construction. If you had a finished road with such a surface that it would carry tractors, there would be no need to put the gravel on it. On the other hand, when gravel is used, it is done largely the first time while the road grade is fresh; it has not been driven on to any great extent and has not had a chance to settle.

MR. MOWRY:—That should be a part of the road construction.

MR. WOLD:—That might be true from one standpoint but not from another. You might say that a dirt road should be built so that it can be driven on as soon as it is completed. That may be true with a more permanent road, but even in that case one cannot drive on it until it is dry and has set. The gravel or crushed-rock road also must set; as a matter of fact, it is a water-bound road.

A MEMBER:—Miles and miles of graveled road were ruined by driving the wagons in one track up and down the road in a certain locality of which I know, and it had to be plowed up afterward. The travel made two ruts that were as hard as adamant; the remainder of the road was loose and never worth anything.

MR. WOLD:—Using large tractors with trailers, when they are driven on different tracks along the road, tends to make a good road; it would be harder to do this with trucks or single units.

MR. KINKEAD:—What is your opinion of a tractor built along the lines of the Gray tractor with a drum? Would that be beneficial?

MR. WOLD:—The wagons would be needed just the same.

A MEMBER:—In the case I cited the gravel was being hauled 3 miles with trucks. I watched the operation very carefully for about 4 days; the trucks were ruining the road by hauling the gravel over it.

MR. WOLD:—For that reason I would recommend lighter trucks and pneumatic tires; they would not injure the roads so much. In one case the drivers were instructed to drive different places on an avenue under construction, so that they would not ruin it.

A MEMBER:—The trucks I mention were heavy 5-ton trucks.

MR. WOLD:—There is an apparatus for weighing the load on each truck tire. It is used in connection with the truck load that can be carried on a given width of tire. It is made in the form of jacks. They are set up under the axle, one at each wheel, jacked up and the number of pounds on each wheel is indicated. The pressure per inch of tire width is computed to determine how much the truck is taxing the road.

A MEMBER:—Would not the proper method in most of the road building in the Minnesota section be to drive on the side of the road? That is done in road building in California, except in maintenance work.

MR. WOLD:—No, it is impracticable here. It might be all right in dry weather but, as a rule, it cannot be done after rains.

A MEMBER:—But it is not necessary to work on the road at such times.

MR. WOLD:—It is necessary. We build roads at all seasons of the year.

MR. GREER:—Will Mr. Wold tell us something about snow-removing machines?

MR. WOLD:—We are developing a snow-plow for installation on a truck. We have built a snow-plow for clearing sidewalks and one for installation in front of tractors. Their use has not been extensive enough to be much of a guide. A snow-plow will work very satisfactorily on a truck. I understand, however, that truck builders object to it because it is too hard on the truck. A snow-plow would be very good on a tractor. We think it is a coming thing and that something of the kind will be required as the roads become all-year roads. The snow must be cleared from country roads the same as from city streets. Snow-plows on trucks are being used in New York City, but that is not practicable under our conditions because we usually have sleet in this section of the Country and generally the snow freezes before we can begin its removal.

MR. SKARNES:—What is the proper kind of road to build in Minnesota? We are appropriating much money for road building. Are we to build roads that in 10 years will still be roads, or will they be what we have today?

MR. WOLD:—I believe that there is no such thing as a permanent road. What are called permanent roads are no doubt more desirable for the traveling public; but we cannot build all roads that way because the amount of money available is insufficient. The next best thing is ordinary dirt roads with a gravel or a crushed-rock surface, and I am inclined to believe that they are, all things considered, the cheapest.

The Fundamentals of Internal-Combustion-Engine Design

By L. H. POMEROY[1] CLEVELAND SECTION PAPER

THE factors affecting the performance of four-stroke-cycle engines are specified and the thermal efficiency and power obtainable from a given engine are stated to be dependent upon the compression-ratio, other things being equal. These features are elaborated upon and the subject of efficiency is treated at considerable length. Engine speed and friction losses are treated in some detail.

Mechanical-construction problems are enumerated and commented upon, inclusive of the respective merits of the long- and the short-stroke engine, the author favoring the latter. Three cardinal points in the design of an engine for automotive use are stated and discussed, and detailed considerations of the mechanical aspect of engine design appended.

A NEW engine design is practically determined by commercial considerations relating to manufacturing economy and public prejudice as visualized by the sales department, but there are certain fundamental problems that are common to all engines. The proper realization of these fundamentals is essential, to satisfy the commercial principle of getting the greatest value for the money expended, whatever the type of the engine may be.

The factors affecting the performance of four-stroke-cycle engines can be grouped as follows:

(1) Compression-Ratio
(2) Mixture Proportions
(3) Heat Losses
(4) Volumetric Efficiency
(5) Mechanical Efficiency

The thermal efficiency and the power obtainable from a given engine are dependent upon the compression-ratio, other things being equal. This does not imply that an engine with a high compression-ratio is necessarily more efficient or powerful than the same engine with lower compression. Much controversy would have been avoided if there has been a clearer understanding of the various practical considerations underlying the phenomena occurring in engine cylinders, particularly those of limited dimensions as used on automobiles, but the above statement holds true if the other things are equal.

The compression-ratio is the deciding factor as to what fraction of the fuel supplied is actually turned into mechanical energy. This fraction is wholly independent of whether the amount of heat supplied is large or small; that is, whether the throttle is open or partly closed. To avoid confusion due to variation in fuel composition, the air-cycle standard was adopted many years ago on the assumption that the working fluid was dry air. The air-cycle efficiency for an engine of a given compression ratio is

$$E = 1 - (1/r)^{\gamma - 1}$$

where

E = the air-cycle efficiency
γ = the ratio of the specific heat at a constant volume to that at a constant pressure or 1.408
r = the compression-ratio

[1] M.S.A.E.—Consulting engineer, Cleveland.

Substituting the value given for γ we have

$$E = 1 - (1/r)^{0.41}$$

This is the foundation of all engine calculations relating to efficiency and power. It will be seen from Table 1 that the efficiency increases rapidly from low to medium compressions and thereafter much more slowly. For this reason there is no great incentive to adopt higher compressions than those at which everyday fuel can be used conveniently.

TABLE 1—EFFICIENCY AT DIFFERENT COMPRESSION-RATIOS

Compression-Ratio	Efficiency, per cent
2 to 1	0.242
3 to 1	0.356
4 to 1	0.426
5 to 1	0.475
6 to 1	0.511
7 to 1	0.540

Unfortunately, the mixture used in internal-combustion engines does not behave as a perfect gas and a somewhat formidable series of deductions are necessary before the air-cycle criterion of efficiency can be applied intelligently. The addition of heat to the working fluid due to the explosion increases the temperature of the mixture very greatly, but not to the extent that is given by simply calculating the temperature increase of the mixture by knowing its mass, specific heat and the total heat supplied. Accounting for this discrepancy has been the task of able scientists for many years and it is only now that anything like definite knowledge is being obtained. Broadly, the effect is as if the specific heat of the mixture were increased at high temperatures so that the increase in temperature is less for a given heat supply than would be the case otherwise. Actually part of the discrepancy is accounted for by direct radiation of heat to the cylinder walls and part of it by dissociation. The important matter to the designer is that the loss due to so-called change of specific heat is tolerably constant and fortuitously independent of the nature of the fuel or the mixture strength. It is unavoidable and must simply be accepted. Its mean value can be taken as 16 per cent of the air-cycle efficiency.

The necessity for maintaining cylinder walls, pistons and valves at a sufficiently low temperature to insure lubrication and adequate cooling reduces the available heat still further by something between 12 to 20 per cent of the air-cycle efficiency, the amount depending upon combustion-chamber design in general. A small deduction also must be made on account of incomplete combustion and early opening of the exhaust-valve. This cannot be reduced to much below 3 per cent. The efficiency attainable is less for these reasons than that indicated by the air-cycle efficiency and amounts to some 63 to 69 per cent thereof. The loss due to the change of specific heat is outside the control of the designer, but that due to direct cooling and incomplete combustion

merits the most careful consideration and its reduction is the essential thermodynamic problem in engine design.

Relative Engine Efficiency

The relation between the efficiency actually obtained and the air-cycle efficiency is known as the relative efficiency of an engine. It may vary, owing to the losses previously mentioned, from 60 per cent in a good engine of standard design to 70 per cent in a very well designed engine in which every care has been taken to suppress the said losses. When the relative efficiency is known, the indicated thermal efficiency is calculated by multiplying the air-cycle efficiency appropriate to the compression-ratio used, by the relative efficiency. The extent to which any given engine falls short of this in indicated thermal efficiency is a measure of what development can be made without fundamental alterations.

The mixture proportions that can be used in any standard type of gasoline engine vary roughly from 70 to 130 B.t.u. per cu. ft. of mixture supplied. Actually, the mixture strength used in normal working is about 100 B.t.u. per cu. ft. and, on this basis, the indicated mean effective pressure can be calculated. For example, taking the compression-ratio at 5 to 1; the air-cycle efficiency, 47.5 per cent; the relative efficiency, 62 per cent; the indicated thermal efficiency, 29.4 per cent; 100 B.t.u. supplied per cubic foot of mixture; and 77,800 ft-lb. per cu. ft.; the number of foot-pounds available per cubic foot is $(77,800 \div 100) \times 29.4 = 22,800$ or 13.2 ft-lb. per cu. in. The number of foot-pounds per cubic inch of mixture obviously represents its energy. The pressure per square inch or indicated mean effective pressure is obtained by multiplying this number by 12. In this case it is 158 lb. per sq. in. which, moving through a distance of 1 in., is equivalent to the energy per cubic inch. However, this calculation involves the assumption that the mixture burned is at 32 deg. fahr. and a pressure of 14.7 lb. per sq. in. at the end of the suction stroke. Unfortunately, this is an impossible condition in the normal engine. The mixture on passing into the cylinder is heated by contact with the hot wall of the cylinder, the piston and the valves. It is impossible also for the mixture to flow into the cylinder without a certain pressure-depression relatively to the atmosphere.

Volumetric Efficiency

The extent to which an engine can avail itself of the mixture supplied per revolution is known as its volumetric efficiency and is primarily dependent upon the pressure and temperature of the mixture at the end of the suction-stroke, excluding the rise of temperature due to the residual exhaust-gas in the cylinder from the preceding exhaust stroke. So long as the pressure of this residual exhaust is not in excess of the incoming mixture, its temperature has absolutely no effect upon volumetric efficiency because it is obvious that this exhaust can be considered as being entirely separate during the suction-stroke and only mixing with the incoming mixture at some later period of the cycle. If, then, 100 per cent volumetric efficiency is that obtained when a charge volume equal to the piston displacement is drawn in at 32 deg. fahr. and 14.7-lb. per sq. in. pressure, it is possible to estimate what the volumetric efficiency may be under practical conditions. If the atmospheric temperature is 60 deg. fahr. and the resistance due to valve-port restriction is 0.5 lb. per sq. in., the only other factor required is that of the rise in temperature due to contact with hot cylinder-walls, pistons, valves and the like. In general, this temperature-increase will not be less than 115 deg. fahr. The volumetric efficiency attainable is, therefore, inversely proportional to the absolute pressure 14.2 lb. per sq. in. and an absolute temperature of 333 deg. cent. (599 deg. fahr.) of the mixture at the end of the suction-stroke; that is, $(491 \div 599) \times (14.2 \div 14.7) = 79.5$ per cent. There are many published figures in excess of this, but they can be disregarded without risk. The figure previously given of 158 lb. per sq. in. indicated mean effective pressure must, therefore, be reduced by multiplying by the volumetric efficiency to bring it into line with actual working conditions. It thus becomes $158 \times 79.5 = 125.6$ lb. per sq. in. This can be considered the maximum attainable under the prescribed conditions.

Engine Speed

The indicated mean pressure already mentioned presupposed a pressure-drop in the induction system of 0.5 lb. per sq. in. At high engine speeds this drop is considerably larger and, according to Ricardo, the volumetric efficiency expressed in terms of inlet gas velocity becomes less than 60 per cent when this is over 250 ft. per sec., as is shown in Table 2. My own experiments do not corroborate this but indicate that the effects due to inlet-port and valve friction reduce volumetric efficiency from 79 to 75 per cent, over a gas-velocity range of from 100 to 300 ft. per sec. Ricardo's deductions were made from an examination of many engine tests and fortuitously represent the facts. It would appear, however, that the effect of internal temperature on volumetric efficiency has been neglected completely and the whole loss related to port and valve restriction only. If this point can be established, as experience seems to show, it is possible to use smaller valves than hitherto considered necessary, with the very important advantages of better turbulence and a more compact shape of combustion-chamber.

TABLE 2—RICARDO'S VALUES FOR GAS VELOCITY AND VOLUMETRIC EFFICIENCY

Gas Velocity, ft. per sec.	Volumetric Efficiency, per cent
100	80.5
120	79.0
140	77.0
160	75.0
180	72.0
200	69.0
220	66.0
240	62.0

Turbulence is the necessary state of commotion in the mixture to obtain complete combustion at the speeds used in automotive engines. It is dismissed usually by the statement that a gas velocity of from 130 to 160 ft. per sec. is essential for adequate turbulence, but nothing is said about the engine speed at which this velocity should be attained. It is obvious that, for automobile engines working over a wide speed-range, the specification of a desirable gas-velocity to insure turbulence is meaningless unless the engine speed also is specified. Fortunately, the gas velocity that is appropriate to adequate turbulence seems to be lower than is prescribed by general conditions of design; that is, it needs a large departure from accepted practice to make valves large enough to obtain such low gas-velocities as to affect turbulence adversely.

My experiments on this point[3] indicate that there is

[3] See *Proceedings of the Institution of Automobile Engineers*, vol. 13, p. 163.

substantially no difference in turbulence at 800 r.p.m. between gas velocities of 56 and 150 ft. per sec., but no difference could be detected at 2000 r.p.m. between gas velocities varying from 167 to 200 ft. per sec. Consideration of the point that high gas-velocities can be used without great reduction in volumetric efficiency indicates that the line of attack in improving existing engines to make them capable of delivering the power that is undoubtedly in them is to study combustion-chamber and piston design to reduce direct heat-losses and internal temperature. It should be mentioned also that neither gas restriction nor internal temperature affect the indicated gas consumption. Indeed, the only effect, even upon the brake gas consumption, is in respect to reduced mechanical efficiency. The reason for this is that the thermal efficiency of an internal-combustion engine depends upon the compression-ratio and not upon the temperature or pressure of the mixture at the beginning of compression.

The indicated mean pressure is a measure of the torque exerted by the engine before overcoming the internal friction thereof. The brake mean pressure is obtained from the indicated mean pressure by subtracting the engine friction expressed in terms of pounds per square inch of piston area. This internal friction due to the sum of gas and mechanical friction and over a speed range of from 400 to 2800 r.p.m., from reliable tests of two engines of very different design, is as shown in Table 3.

TABLE 3—INTERNAL ENGINE-FRICTION

Engine Speed, r.p.m.	Engine Friction per Square Inch of Piston Area, lb. Four-Cylinder $4\frac{1}{4}$ x $4\frac{1}{4}$ in.	Six-Cylinder $3\frac{3}{8}$ x $5\frac{1}{2}$ in.
400	6.0	7.0
800	8.4	8.9
1,200	11.2	12.0
1,600	14.9	14.9
2,000	17.6	17.5
2,400	20.9	20.8
2,800	23.0	24.0

It is of interest to note that the weight of the six-cylinder-engine piston complete, with wrist-pin and rings, was 1.5 lb., and the weight of the four-cylinder-engine piston in a similar condition was 2.4 lb. The total reciprocating weight was, therefore, as 9.0 is to 9.6 and the piston friction in each engine is such that the difference between the two is probably less than the difference that would be noticed between a series of engines of either type. These are good average figures for well-made engines with forced lubrication and aluminum pistons. Subtracting these piston-friction values from the indicated mean pressure as previously obtained, the brake mean pressure and, consequently, the power obtainable at any given speed can be determined. The indicated mean pressure already calculated is that obtainable under the conditions of minimum gas-restriction and internal cylinder-temperature. The effects of these two taken together are given by Ricardo's figures as shown in Table 2 and, in the present state of knowledge, these are a safe guide.

Internal Engine Friction and Summary

To determine the brake mean pressure over the complete speed-range it is therefore necessary to use the volumetric efficiency applicable to the gas velocity at each engine speed in calculating the indicated mean pressure and then to deduct therefrom the appropriate friction losses. I have indicated the pitfalls in this procedure but, in the present state of knowledge, it is the best that can be done and such a calculation will usually show the extent to which most engines fall short of what can reasonably be expected.

Attention is drawn to the great importance of minimizing internal friction in any engine that works at very partial throttle, as is the case with all automobile engines. The internal friction can easily equal the horsepower actually developed; that is, the mechanical efficiency may not exceed 50 per cent under running conditions, so that even a small reduction in the engine friction may have a very powerful effect upon the gas consumption.

Summing up, the brake thermal efficiency of an automotive engine is dependent upon the compression-ratio, the mixture strength and the mechanical efficiency. The possibilities of improvement lie in the use of higher compression and the reduction of direct cooling and mechanical friction losses. The power obtainable, from the thermodynamic viewpoint, depends upon the compression-ratio, the volumetric efficiency, the mixture strength and the mechanical efficiency. Considerable possibilities of improvement lie in the study of internal engine-temperatures, combustion-chamber shape and the reduction of internal friction.

The question of mixture strength is not of great practical importance as the mixture proportions that can be used are limited and, incidentally, can easily be made adjustable; also, the best mixture for maximum torque is about the same as that for maximum efficiency. In the automobile engine, however, there are many considerations other than thermal efficiency and maximum torque. It is absolutely essential to possess the ability to accelerate without hesitation; the vast majority of users judge power by the capacity to accelerate, although these two qualities are distinctly separate. It is in this aspect of the question that the carbureter expert steps in and, by the larger wisdom arising from an extended experience, supplies the knowledge without which an automobile engine may possess all the virtues but be bereft of charm. This is indeed a field in which progress has been made by practice rather than by mathematical theory. The whole literature of practical carburetion is a mist of vague speculation, contradictory theories and intensive groping for the light. As such, I will leave it with the one remark that he who designs automotive engines should consult the best carbureter expert he knows before putting his pencil to paper. This concludes the summary of what may be called the general theory of design with respect to efficiency and power.

Mechanical Construction Problems

After many years of experience, I believe the short-stroke engine is the best for all purposes when maximum high-speed torque is unessential. The last ounce of power at very high speeds involves a sacrifice of many other things for the sake of a compact combustion-chamber. Although justifiable in a high-speed racing engine, it will not be warranted in the automobile engine of commerce until high car speeds are much more important than at present. It must be remembered that while racing-car engine-speeds have increased some 50 per cent in the last few years, touring-car engine-speeds have remained stationary. Even now gear-ratios show signs of being increased, thus reducing the engine speed. The racing engine and the touring engine become more sharply differentiated year by year and, like race horses and racing yachts, racing cars are becoming a distinctly

separate breed with less than ever in common with the species from which they have sprung.

The cardinal points in the design of an engine for automobile use are:

(1) Light weight
(2) High brake thermal efficiency at light loads
(3) Low internal-friction losses

That the obsession of the long-stroke engine is passing is indicated by some of the later aircraft-engine designs such as the Napier Lion in which the bore is considerably greater than the stroke, this being a fair example of a large high-speed engine in which lightness is, of course, a sine qua non while high specific power and economy are essential. A large part of the weight of any engine is incurred through the geometrical disposition of its elements. For example, the necessity for maintaining a reasonable relation between the length of the connecting-rod and the crank in a long-stroke engine imposes upon the designer either a high crankcase or long cylinders whose only function, so to speak, is to fill up space. Further, the additional "crankiness" of a long-stroke crankshaft calls for more metal to maintain the desired stiffness. The short-stroke crankshaft is in this respect infinitely easier to deal with.

The argument for the inherent lightness of engines in which the bore and stroke are approximately equal is demonstrated easily by considering any standard design from the viewpoint of increasing the cylinder capacity thereof. It will be found that increase of bore will produce a corresponding increase of cylinder capacity with far less disturbance than that entailed by increasing the stroke. The only arguments against the short-stroke engine are that it is more difficult to obtain a compact combustion-chamber and to keep down piston temperature than with the same cylinder capacity but with a smaller bore. The aluminum piston can overcome the latter difficulty easily. My latest L-head engine of 4¼-in. bore and 4¼-in. stroke, having a compression-ratio of 4.5 to 1, is showing a brake fuel-consumption of 0.55 lb. per b.hp.-hr. at any speed from 600 to 2400 r.p.m. This is a figure that is by no means bad for any engine, regardless of its bore and stroke.

To obtain high brake thermal efficiency at light loads it is essential that the indicated thermal efficiency also shall be good; that is, that the direct cooling losses be reduced so far as possible and that the combustion-chamber be designed with the object of presenting the minimum surface per unit volume. At first sight, this seems inconsistent with ratios of bore to stroke that approach unity, but skilful design in placing the bulk of the compression space over the valves in an L-head engine will give results very comparable indeed with some of the best obtained in long-stroke overhead-valve engines. With this condition fulfilled, the question of reducing the internal friction-losses becomes of dominating importance.

The chief source of friction in automotive engines arises from the piston. Piston friction is about 50 per cent of the total friction in an automobile engine and obviously deserves attention as to whether it can be reduced. The whole question of internal losses has so far been the subject of more or less unsatisfactory guess-work, particularly in the statements on piston friction, and there is room for very valuable research work on this subject. My view is that piston friction is a function of inertia loading and oil-film shearing resistance, and that pistons showing a low friction-value are characterized by large clearances and small areas of contact between the piston and cylinder walls, or means whereby the oil-film can be "rolled up" without restriction. A familiar method of achieving the last is by the use of drilled pistons. Whatever the cause, the short-stroke engine seems to possess considerably less internal-friction than one of long stroke.

This does not seem unreasonable when it is borne in mind that the surface of the oil-film subjected to shear per stroke is a function of the circumference of the piston multiplied by the stroke, while the friction expressed in pounds per square inch of piston area is a function of the piston diameter squared, multiplied by the stroke. The optimum condition evidently is reached when the friction due to inertia loading, as such, plus that due to oil-film shearing, is a minimum. In this connection, due to exigencies of foundry work, the weight and hence the inertia loading produced by pistons of the sizes used in automobile engines do not increase according to the dimensional theory; that is, as the cube of similar dimensions, thus minimizing the inertia loading effect in the short-stroke engine compared with one of the same capacity but with a long stroke.

The essence of the mechanical aspect of engine design lies in this minimizing of inertia loading effects. From the familiar miles per gallon viewpoint it should never be forgotten that, taking for example a 3000-lb. automobile, it requires about 4 hp. to propel the vehicle at 20 m.p.h., while, at the engine speed corresponding with this car speed, a 180-cu. in. engine may easily absorb 2 hp. in mechanical friction alone. The results to be obtained by cutting down internal friction are thus indicated very clearly.

The use of materials of high specific-strength, so that the inertia loading is reduced, is important. Of such materials, the various alloys of aluminum possess unique properties owing to their low specific-gravity. The specific strength of a material is a number obtained by dividing the tensile-strength by the density, and it is interesting to note that from this viewpoint cast aluminum is equal to a 0.20-per cent-carbon forged steel, while forged aluminum has a value equal to that of some of the finest alloy-steels. Since the stresses induced in the piston and connecting-rod are very largely a function of the inertia of these parts, the importance of this consideration is evident. In addition, the reduction of such inertia stresses obviously reduces the inertia loading on the various engine bearings. They can either be reduced for equal life or have their life prolonged by such reduction of the inertia loading.

The virtues of the aluminum piston are now generally accepted from the viewpoints of lightness, heat conductivity and mechanical strength. Following upon this is the logical corollary of a forged aluminum connecting-rod. The loading on the connecting-rod big-end bearings is composed of (*a*) that due to the gaseous pressures arising during compression and expansion and (*b*) that due to inertia forces arising solely from the mass of the moving parts, namely, the piston and the connecting-rod. If the loading of the crankpin during a complete cycle be studied, the overwhelming importance of the inertia loading compared to that arising from the gaseous pressures will be manifest. In fact, it is invariably found that crankpin wear in high-speed engines occurs on the inner side of the crankpin; that is, that nearest the crank axis and not on the outer side of the crankpin, as would be the case if the wear due the pressures during explosion and compression predominated.

It is found that the use of an aluminum connecting-rod as against a steel connecting-rod, both having aluminum pistons, reduces the loading on the big-end bearings

by some 30 to 40 per cent, thus allowing the designer either to speed his engine up for a given bearing-endurance capacity or to increase the endurance capacity of the bearings in cases where these are already overloaded. In addition, the mass of the big-end of the connecting-rod plays no inconsiderable part in the torsional oscillations on long crankshafts, so that the use of an aluminum connecting-rod is entirely beneficial in engines where periods exist, arising from the above stated cause.

After considerable experience, I believe that the aluminum connecting-rod is one of the most important recent detail developments relating to internal-combustion-engine design. It is economical to manufacture and it is readily susceptible of being direct-babbitted, with all the advantages this method of construction carries with it. Now that the limiting factors of engine speed are being determined with something like precision, the importance of weight reduction in respect to rotating and reciprocating parts becomes more pronounced than ever. It seems highly probable that the use of aluminum for such parts will enable designers to reduce the bearing surface, thus cutting down the overall length of engines and making savings from the viewpoints of engine weight and cost of production that can be appreciated only after a careful investigation of the whole subject. In other words, the statement of a famous automobile engineer that "An ounce off the piston is as good as a hundredweight off the car" may be soon far more than an expression intended to emphasize the importance of weight reduction in respect to the rotating and reciprocating parts.

THE DISCUSSION

H. C. SNOW:—What should be the size of exhaust-valves?

L. H. POMEROY:—So far as exhaust-valves are concerned, the limiting diameter seems to be determined by cooling difficulties. I think it is not practicable to use an exhaust-valve of over 2-in. diameter on any engine running at more than 2000 r.p.m. It usually is possible to run with exhaust-valves that are very much smaller than the inlet-valves, without any loss of power or economy.

A MEMBER:—In connection with aluminum connecting-rods, have any alloys of aluminum been tried without using any babbitt metal in the bearings?

MR. POMEROY:—Several American firms have experimented with aluminum rods running directly on the crankshaft. In one instance a case-hardened crankshaft was used, but so far such experiments have not been successful and the recommended practice is to direct-babbitt the connecting-rod. There is, of course, no more objection to the use of babbitted bronze-shells than is the case with steel rods but, in view of the extra oil-film through which the heat generated by the friction of the bearing must pass, the use of babbitted bronze-shells deserves careful consideration as to whether the advantages they present in regard to replacements do not make the replacements necessary.

A MEMBER:—I understand that trouble has been experienced with the alloyed aluminum in making the babbitt metal adhere to it.

MR. POMEROY:—There is no more difficulty in successfully direct-babbitting an aluminum connecting-rod than in performing the same operation on a steel rod. In fact, it is considerably easier to make a firm union between babbitt and aluminum than between babbitt and steel.

A MEMBER:—You stated that the fuel-mixture is usually 15 to 1. What are the limits that can be used?

MR. POMEROY:—Under proper conditions of temperature and absence of deposition, the richest mixture that can be burned has about 10 parts of air to 1 of gasoline; the weakest mixture has about 20 parts of air to 1 of gasoline. In the richer mixture, combustion is not complete, but because the after-explosion volume of the mixture is greater than its volume before explosion, it will give an increase in mean effective pressure at the expense of economy.

A MEMBER:—What is the definite ratio of valve-lift to diameter of port? You referred to changing the diameter of the valves. It is easier to change the valve-lift in the ordinary engine than to change the diameter of the port.

MR. POMEROY:—All textbooks show that the lift of a valve should be one-fourth of its diameter, so that the area round the valve is the same as that of the port. This is, however, merely a geometrical equality and takes no account of the different orifice-coefficients arising from the fact that the gas is issuing through an annulus which is very different in shape from a circular port. The interesting experiments published in the Clarke-Thompson research[8] indicate that there is an increase in valve capacity up to a lift equal to 0.7 of the valve diameter and that the amount of gas that can be passed through a valve is proportional to its periphery multiplied by the lift. My own experience with two engines of which the valve area was in one case controlled by the height of the lift, keeping the valve diameter constant, and in the other case with varying valve diameters, keeping the lift constant, indicates the truth of this statement. Considering the compression-pressure at any given speed as a measure of the volumetric efficiency, the results obtained on these two engines were the same within 2 per cent, irrespective of the nature of the valve-opening. The difficulty in connection with the high valve-lift is to arrange for cam profiles and valve-springs to take care of it. The valve-spring seems to be the nearest metallurgical equivalent to dynamite and the difficulties in working with high speeds and high valve-lifts can be thoroughly appreciated only after bitter experience.

A MEMBER:—What is the general tendency of internal-combustion-engine speeds in Europe? Has there been any increase in them?

MR. POMEROY:—The only alteration in engine speeds in European practice is that due to reductions of gear-ratios, thus increasing the average speed of the engine. On the other hand, the maximum speed has not been altered greatly except in some of the new sport-type cars where high car speeds on indirect gears are desired. Possibly the most interesting developments in this respect occur with the so-called light cars that have engines capable of maintaining speeds of 2500 to 3500 r.p.m. for hours on end.

A MEMBER:—In regard to your experiments with the reduced inlet-valve, how much value do you attribute to turbulence in the results obtained?

MR. POMEROY:—Turbulence is a function of gas velocity. On the other hand, it seems rational to consider not only gas velocity but gas velocity and revolution speed as being correlated. It is obvious that a gas velocity of 100 ft. per sec. will give all the necessary turbulence for complete ignition at 500 r.p.m., but be entirely inadequate at 2500 r.p.m. The relationship between gas velocity and revolution speed is of great interest, although I am strongly of the opinion that it is possible to use higher

[8] See National Advisory Committee for Aeronautics Report No. 24.

(Concluded on p. 354)

Notes on Motor Trucks

By Cornelius T. Myers[1]

Pennsylvania Section Paper *Illustrated with* Photographs and Charts

AFTER pointing out that the publication of articles in the trade and technical journals, to the effect that very considerable weight-reductions in motor-truck construction with consequent savings in gasoline and tires are possible, works an injustice to the motor-truck industry and is misleading, the author outlines some of the reasons why such weight-reductions are very difficult to effect, as well as the possibilities of standardizing axle details. The use of aluminum to effect weight-reduction is commented upon and the various advantages claimed for metal wheels are mentioned. In the latter connection the author points out that, while these claims may be true, they are unsupported by reliable data.

The greater part of the paper is devoted to an account of a series of tests conducted by a large coal company to determine the relative merits of wood and metal wheels on its trucks. Four trucks, each equipped with wood and metal wheels on diagonal corners so as to secure, as far as possible, an equalization of conditions were employed. The tests lasted over a year, and at their conclusion it was found that the average wear of the tires mounted on metal wheels was about 13 per cent greater than that of those mounted on wood wheels.

The question of unsprung weight is discussed, as is the importance of reducing chassis and body weights to a minimum, but it is pointed out that a reduction in these weights does not necessarily mean a resultant saving in the gasoline consumption or the tire expense. Lubrication of the various parts of a motor-truck chassis also receives attention, the annual cost of truck repairs due to poor lubrication of the chassis being given as from $15,000,000 to $20,000,000, in addition to which there is a loss of earnings while the trucks are being repaired of from $30,000,000 to $35,000,000. The superiority of oil over grease as a chassis lubricant is emphasized.

In conclusion it is pointed out that the next few years will see bitter competition in motor-truck service and that refinement of detail and simplification of operating features will be emphasized. The need for the unexperienced buyer to secure expert advice regarding the design of the vehicles offered him is stressed, it being stated that this practice would clarify the situation for both the buyer and the seller.

ARTICLES keep appearing from time to time in various publications to the effect that our motor trucks are far too heavy and that they can be lightened materially if we but use aluminum and alloy-steels in this or that form; also that these weight-reductions will effect big savings in gasoline and tires. These articles prophesy a Ford passenger-car descendent in a motor truck, and are not at all just to the motor-truck industry in that they lead purchasers to believe that the near future holds out the fulfillment of such a promise. There are also statements in regard to unsprung weight, quantity production, wheel construction and similar subjects that are likely to be misleading to those not in close touch with the subject. Doubtless, in time, great advances in construction will be achieved, but this will be rather by a large number of short steps than by a revolutionary change in design.

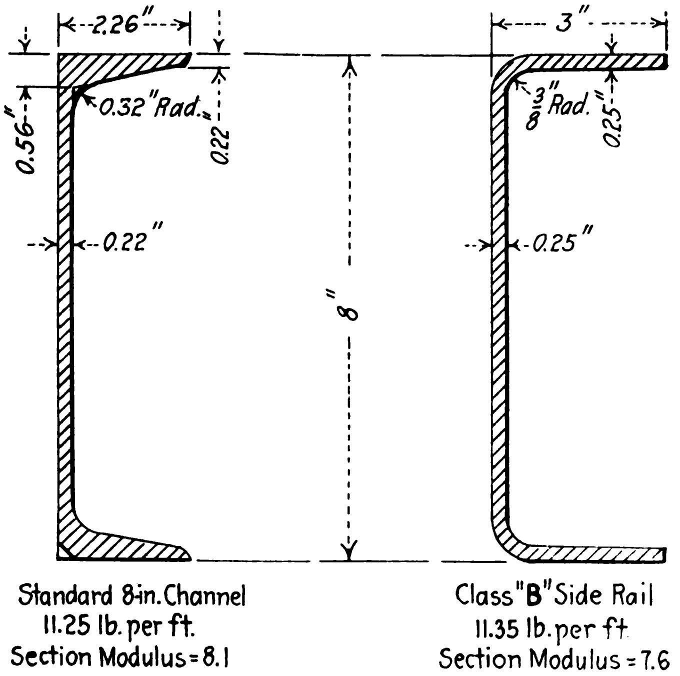

Fig. 1—Rolled and Pressed Steel Frame Sections

The Standard Rolled Channel Is 1 Per Cent Lighter and 7 Per Cent Stiffer Than the Pressed Channel. The Latter Permits the Use of Large and Easily Made Fillets in the Frame Bracket, Although the Rolled Section Is Often Beveled Off to Give the Same Advantage. Bolts Through the Flanges of the Rolled Channel Require Special Wedge-Shaped Washers to Hold the Heads True

We often hear the alloy-steel man talk of much lighter frame-sections, etc., but we have reached a point today in motor-truck construction where the possibilities for chassis weight-reduction are approaching a safe limit, regardless of application of higher-grade steels, because the limiting features in many parts of the chassis design are those of deflection and of metal thickness sufficient to hold rivets and avoid buckling, rather than tensile stress. One magazine article states, "In frame construction the old heavy charcoal-iron has been almost entirely superseded by the pressed steel which, with about half the weight of the former, possesses nearly double its strength." This would result in making each pound of steel in the frame about four times more effective. But this is not the case. Presuming that "charcoal iron" refers to the rolled standard structural section, it is worthy of note that several very widely known and highly rated truck builders still use frame steel in this form, and also that their frame weights are, if anything, heavier today than 8 or 10 years ago for similar load capacities. Further, the weight of frames made of pressed steel for similar load capacities by well-established competing truck builders averages about the same as the weight of frames made from rolled sections as is brought out in Fig. 1.

[1] M.S.A.E.—Consulting engineer, Rahway, N. J.

The same principle applies to axles. Lighter front-axles can be made of high-tensile heat-treated steel, but under working loads the deflections would be such as to seriously misalign the wheel spindles and steering connections. Also, such an axle, when accidentally bent could be straightened only with difficulty and would have doubtful value thereafter as a load carrier. Rear axles, too, could be lightened in the same way, but at the expense of misalignment under load deflections that would be fatal to their driving mechanisms.

Possibilities in Rear-Axle Standardization

A comprehensive paper on Rear Axles for Motor Trucks was presented last month before the Metropolitan Section. A casual glance through the paper by a member of the Society's Standards Committee will cause him to contemplate with awe the enormous investment in axle spare parts it must entail to service the different makes of motor truck already in operation. Further contemplation increases this by the amount of different manufacturing tool equipment necessary to produce them, and the terrific expense it must entail to keep the many little armies of salesmen each fighting their twenty-sided battles over unimportant detail. Eighty million dollars is spent every year to sell trucks, a large part of it going to support fruitless detail arguments. The truck industry now generally recognizes that it must sell transportation and not details, and there seems to be a great opportunity to standardize on axle details. Everyone at interest will benefit by this, for all the well-known types of rear axle are firmly established in the trade and give very good results in use. The user is not interested in axles if they carry his goods without disturbing his peace of mind. When accidents happen or wear takes place, however, he wants repairs made quickly and cheaply. This not only saves him money but keeps him in the frame of mind to spend money in buying another truck. I will put before you certain features of the various axle designs that seem to offer more or less attractive fields in which to pursue the well-defined and enormously profitable policy of standardization, that has been steadfastly fostered by the Society of Automotive Engineers.

(1) For each of seven truck sizes, standard spring center-lines and spring widths can be established; giving a reasonable variation, for the time being, by using spring-pads that are wider than necessary. The spring centers will depend on frame widths, a subject that is already being attacked with every reason to expect success. These established, the spring clips and nuts will fall readily in line. Even the spring-pad drilling and spring-seat height offer possibilities of being reconciled between chain-drive and internal-gear axles, and also among single-reduction, double-reduction and worm-drive axles

(2) Standard brake-drum diameters, widths and metal thicknesses are now being discussed in the Standards Committee and tentative dimensions have been suggested. Once these are determined, the door will be open to the crying need for standardizing the dimensions of brake-linings, rivets, hinge pins, cams or toggles, diameters of camshafts and bushings, fits for camshaft levers, clearances and movements, release springs and all the minor details that are so annoying when replacements are needed at an inopportune time or in an out-of-the-way place. There will be double paths to follow here by those who differ as to having more than one brake inside the drum, and also by the adherents of the external brake

(3) Once the brake-drum width is selected, the wheel spoke is located. The wood wheel can fall in line readily here and the steel wheel also. Our tire sizes are standard and also their fits on the wheels. Standard practice with reference to front-wheel hubs has already been adopted. The rear hubs also should yield to the same analysis and attack, though the solution will not be so simple. The hub details depend upon the type, size and location of the bearings used

(4) The wheel bearings cause a considerable diversification. Either ball or roller bearings can be employed in each of the five types of axle most used. But of these five types single-reduction, double-reduction and worm-drive can be grouped in both the fixed-hub or full-floating designs; while chain-drive and internal-gear drive also can be grouped. Thus, for each size truck axle 16 variations in bearing sizes, types, fits and spacing may be reduced to six. Possibly a thorough study of the situation would reduce these still further, as was found possible in the front-hub bearing standardization, in which over 200 hub variations were boiled down to 10, which are served by only 14 bearings

(5) Fits for hubs, flanges, hub-caps and wheels can be standardized correspondingly, once the bearings are fixed

(6) The fits at the outer ends of spindles and drive-shafts can be standardized in accordance with the classification given under (4); and if the wheel tracks are held close the spindles and drive-shafts themselves offer possibilities. Items 4, 5 and 6 will react on the spring-pads mentioned under (1), and help reconcile some discrepancies that at first may occur there

(7) Differentials can be standardized. The subject is now under consideration. A reasonable number of sizes to cover the range of torques to be transmitted, is all that is necessary. It should make little difference in what type of axle they are used. There would be a series for ball bearings and a series for roller bearings.

(8) The above accomplished, the fits for the inner ends of drive-shafts would fall in line

(9) Bevel, spur and worm gears all offer great opportunities for standardization. This would lead to a similar accomplishment for the bearings and bearing fits on worm spindles and bevel-pinion spindles, but these can be standardized whether or not the gearing is standardized

(10) Adjusting devices for differential bearings and spindle bearings also offer a field

(11) The spindle fits for universal-joint flanges have already been standardized

(12) Clevises and pins for brake-lever connections have been standardized. Possibly the levers themselves are susceptible to this work

Here, then, are 12 features in rear-axle standardization, some of them having been solved. All types of rear axle are now so perfected and perform so well that a standardization of all assembly dimensions and common details is the next big move in the progress of the industry.

Alloy-steel crankshafts are negligibly lighter for the same stiffness than those made of less expensive carbon-steel. The alloy-steel will resist wear somewhat better than carbon-steel if the latter is not case-hardened. However, wear can best be reduced by improved lubrication.

Weight-reduction by means of aluminum is not new in engine crankcases, gearboxes, radiators and other smaller parts. Temporarily, cast iron is being used to a large

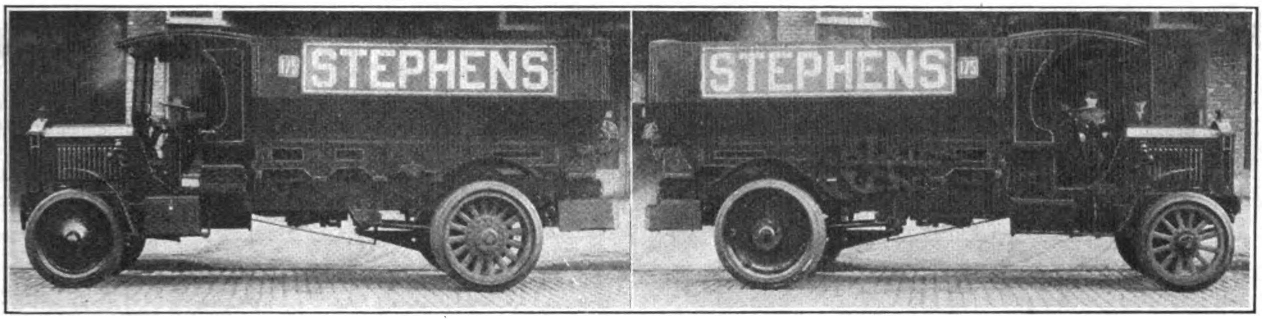

FIG. 2—VIEWS OF OPPOSITE SIDES OF ONE OF THE TRUCKS TESTED

extent, because of the high cost of aluminum. Aluminum has been used successfully in some parts of passenger-car rear-axles. Some experimental truck-axles have been made of it. But this metal has a structure too coarsely crystalline to make it readily serviceable under the heavy and uncushioned shock loads a truck rear-axle must endure. Some alloy of aluminum may solve the problem, however.

A few truck wheels have been made of aluminum but with unsatisfactory results. The most promising truck wheel today is the wood-spoke type but of lighter construction and better detail than those made in the past. There is much misunderstanding of the functions performed by motor-vehicle wheels, and much misinformation has been circulated in the past few years. The average man's idea of a wheel covers a device that revolves on an axle spindle and carries a tire. The common opinion is that the stronger a wheel is, the better. This accounts to a considerable extent for the development of various types of metal wheels, cast steel, malleable cast iron, rolled or pressed disc and even forged steel, that are offered and championed mainly on two contentions that (*a*) they are stronger than the wood-spoked wheel; and (*b*) they are more accurately made and more true than the wood-spoked wheel. Both of these contentions are true. It is claimed, therefore, that

(1) Tires mounted on them will last longer
(2) Gasoline consumption will be less
(3) The truck will be less liable to be put out of commission due to accidental damage to the wheels

It is claimed also that cast-metal wheels weigh less than the conventional wood wheel and therefore decrease the unsprung weight, 1 lb. of unsprung weight being variously estimated in these claims as equivalent in load effect to 5 to 10 lb. of weight above the springs.

Wood and Metal Wheels

For several years there has been considerable controversy as to the relative merits of motor-truck wheels made of wood in the usual manner and those made of cast steel or malleable iron. The following are typical statements, taken from manufacturers' circulars or from the advertising pages of trade papers with reference to cast-metal wheels:

> The permanent true roundness of .... wheels makes for lower fuel bills, and increases the mileage obtained from tires. Users tell us of increases of from 10 to 30 per cent
>
> They increase tire life; they decrease fuel consumption
>
> To move a pound of weight in the wheels requires as great an expenditure of tires, power and fuel as is needed to move 10 lb. above the springs..........
>
> .... wheels are as much as 100 lb. lighter per set than other types of wheels
>
> Their trueness and roundness add 10 to 25 per cent to the mileage obtainable from tires
>
> They reduce unsprung weight and lower tire costs

Repeated inquiry among those who made these statements convinced me that they were made on a basis of supposition and hope, and not on established data. Possibly the claims were true, but no data supporting them were forthcoming. Neither was there any sound basis for the statements made in regard to unsprung weight, or the inferences drawn therefrom.

While it is true that the manufacturing limits for roundness of wood wheels are several times as great as those for steel wheels, the variations allowed the wood-wheel manufacturer are so small as to be negligible in the operation of the truck. The inequalities of the road surface are so much greater and the cushioning value of the tire is so large that the effect of the relatively minute wheel variations cannot be measured in any operating expense. The greater accuracy of the metal wheel seems, therefore, to be but gilt on the lily.

In checking up wheel weights it was found that in commercial production the cast wheels were in most cases heavier than wood wheels built for the same load capacity, especially in sizes up to those suitable for 3½-ton trucks. In sizes for 5 to 7½-ton trucks wheel weights in some instances more nearly coincided but the wood wheels still averaged somewhat less in weight. In comparing weights the weight of the hub, flange and brake-drum or sprocket assemblies were included in the weight of the wheels. It did not seem likely that the heavier metal wheels would give greater tire mileage; but our absolute knowledge of the effects of varying amounts and proportions of sprung and unsprung weight is still very limited. In spite of the fact that the superior tire-mileage claims were based on *lighter weights* for metal wheels, it seemed that some comparative data should be secured, rather than leave the truck user to conjecture on a point so important in the operation of his vehicles. The motor-truck user has to spend his good money for everything he gets, and it is important for him to know whether the extra price he has to pay for cast wheels really comes back to him in tire savings. A practical test under every-day conditions, but with proper safeguards, is necessary to secure this information.

Tests Made With Four Trucks

A company that trucked large tonnages of coal in one of our big cities, was interested in this subject, equipped four of its trucks and conducted such a trial. It operates a large fleet of trucks, mostly of 6½ and 7-ton capacity. It keeps a careful record system that includes mileage, tonnage and operating costs. Its trucks are well kept-up. The drivers are well trained. The operation of the trucks is very capably supervised. Altogether the conditions were as near ideal as could be expected.

On trucks of 5-ton capacity and up, wood wheels are thought to be at more of a disadvantage than on trucks of smaller capacity, and 6½ and 7-ton trucks were therefore used in the test. The trucks carried full loads on each trip and operated on regular schedules and predetermined routes. Half of the trucks were equipped with

TABLE 1—WHEEL AND TIRE EQUIPMENT

Truck	Front		Back	
No.	Wood	Metal	Wood	Metal
162	*Left*	*Right*	*Right*	*Left*
	Interlock spoke	Hollow spoke	Interlock spoke	Hollow spoke
	36 x 6 S	36 x 6 S	40 x 6 D	40 x 6 D
168	*Right*	*Left*	*Left*	*Right*
	Mitred spoke	Hollow spoke	Interlock spoke	Hollow spoke
	36 x 6 S	36 x 6 S	40 x 6 D	40 x 6 D
172	*Left*	*Right*	*Right*	*Left*
	Mitred spoke	Cast disc	Mitred spoke	Cast disc
	36 x 6 S	36 x 6 S	40 x 7 D	40 x 7 D
175	*Right*	*Left*	*Left*	*Right*
	Mitred spoke	Cast disc	Mitred spoke	Cast disc
	36 x 6 S	36 x 6 S	40 x 7 D	40 x 7 D

wood wheels on the right-hand end of the front axle and the left-hand end of the rear axle; the wheels at the other diagonal corners of the truck being cast-metal as is shown in Fig. 2. All the wheels were equipped with new tires, of not only the same make, but made at about the same time with the same compound and cure. All tires contained approximately the same thickness of rubber. The rated loads on the tires were within commercial limits of good practice per inch of width, and these loads did not vary widely for the different trucks used.

Two chain-drive trucks, Nos. 162 and 168, and two worm-drive trucks, Nos. 172 and 175, were selected for the test. The wheel equipment was as set forth in Table 1.

Each wheel was carefully stamped for identification. The metal wheels for trucks Nos. 162 and 168 and the wood wheels for trucks Nos. 172 and 175 were new, but

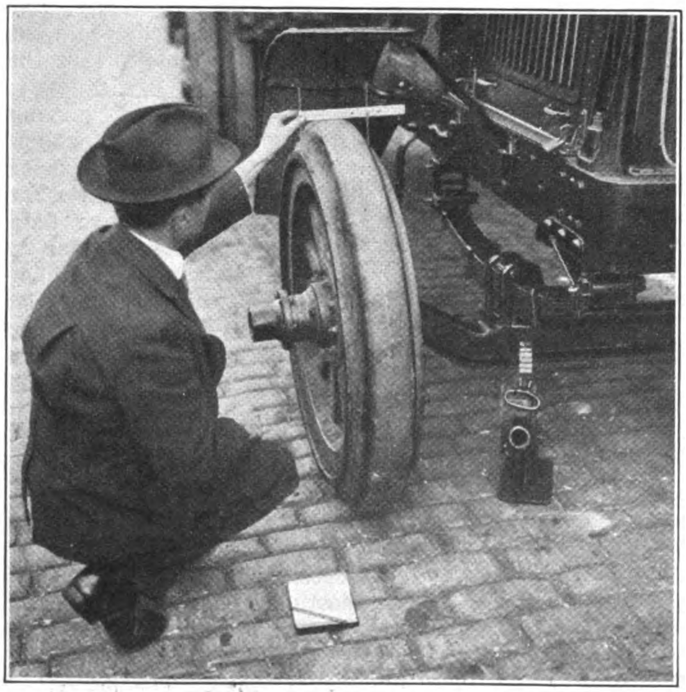

FIG. 3—TAKING MEASUREMENTS OF THE HEIGHT OF THE RUBBER ABOVE THE TIRE BAND

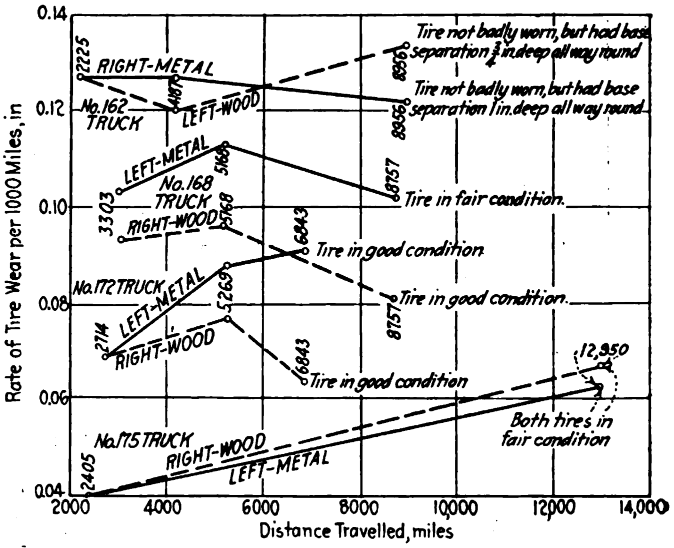

FIG. 4—RELATION BETWEEN THE RATE OF TIRE WEAR PER 1000 MILES OF TRAVEL AND THE DISTANCE COVERED FOR FRONT WHEELS

the wood wheels, for trucks Nos. 162 and 168 and the metal wheels for trucks Nos. 172 and 175 had had considerable service on these trucks. This arrangement, it will be noted, placed the wood wheels and also the metal wheels at diagonal corners of each truck, but alternated them as to rights and lefts on the two trucks of the same make. This assured that the average wear would take place under the same conditions for both wood and metal wheels. The superintendent said that his records showed that in general the tires on the right-hand side of their trucks wore-out somewhat more rapidly than those on the left-hand side. This is probably due to the slight slope of the road toward the right, which loads the tires on the right-hand side somewhat more than those on the left-hand side.

The front wheels on all four trucks carried 36 x 6-in. single tires. The back wheels on trucks Nos. 162 and 168 carried 40 x 6-in. dual tires, while the back wheels on trucks Nos. 172 and 175 carried 40 x 7-in. dual tires; 24 tires in all being included in the test so as to get a fair average.

Before purchasing the tires the matter of furnishing a uniform quality was taken up in detail with the tire factory, and after careful inquiry tires were selected that had all been produced at about the same time, and which the factory stated could be depended upon to be practically uniform and give comparable results in this test.

Each steel tire-band was stamped for identification so that damage to the serial number on the tire would not cause confusion. Each tire-band was then marked at six places, equidistant around the rim. At each of these places the height of the rubber above the flange of the band was measured carefully with a gage made especially for that purpose as shown in Fig. 3. The average of these measurements was taken as the initial height of the rubber in the tire. All measurements were checked by two persons. Tires selected were applied at random on the wheels, and it was impossible to distinguish any difference in them.

As each truck was equipped with new tires, records of mileage and tonnage were carefully compiled. The starting dates of the tests are given in Table 2.

In addition to the regular operating records of these trucks (they were operated entirely in accordance with the business demands and not given any special routes or service) the tires were given a special inspection about

TABLE 2—STARTING DATES OF TESTS

Truck No.	
162	Nov. 11, 1920
168	Nov. 12, 1920
172	Nov. 19, 1920
175	Nov. 18, 1920

once a month. This was to keep track of any unusual wear, bad cuts, base separation, etc.

Due to the mild winter of 1920-1921 these coal trucks did not cover a very great mileage for the first month or so. Each load carried, however, was a capacity load, as can be seen from Table 3.

TABLE 3—CAPACITY AND LOAD PER INCH OF TIRE WIDTH

Truck No.	Pay-Load Capacity, tons	Average Weight per Inch of Tire Width, lb.
162	6½	720
168	6½	743
172	7	702
175	7	696

On March 4, 1921, inspection of the tires showed but little wear, and all of them were in good condition. Measurements were taken. The rates of wear in inches per 1000 miles and the mileage at that date are shown on

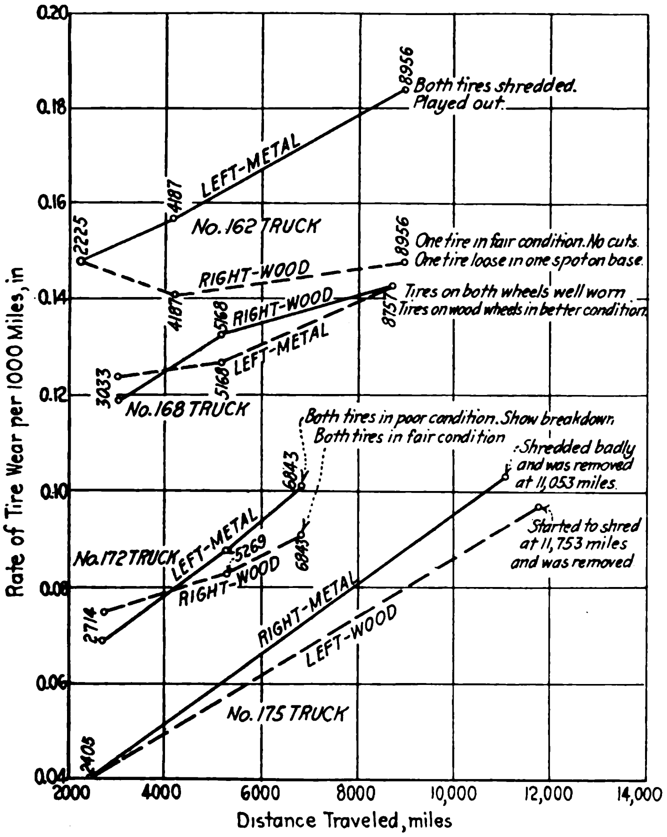

FIG. 5—RELATION BETWEEN THE RATE OF TIRE WEAR PER 1000 MILES OF TRAVEL AND THE DISTANCE COVERED FOR REAR WHEELS

charts reproduced in Figs. 4 and 5. The average rates of wear show that for the front wheels the tires on the wood wheels were doing about 2.75 per cent better than those on the steel wheels. On the rear wheels the tires on the steel wheels averaged 0.5 per cent better than those on the wood wheels.

In considering the results given herein, it must be borne in mind that the differences in rate of wear between individual trucks is not directly comparable, because they all operated under somewhat different conditions. For instance,

(1) Their routes and mileage varied

(2) Their tare-weights were different, as is indicated

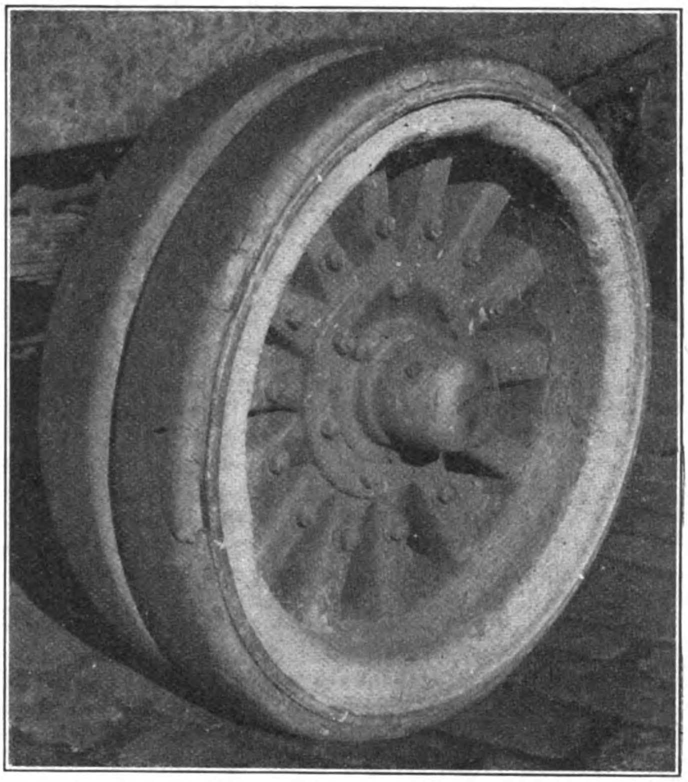

FIG. 6—REAR WOOD WHEEL ON THE SAME AXLE AS THE WHEEL IN FIG. 7 AND ITS TIRE AT THE CONCLUSION OF THE TEST

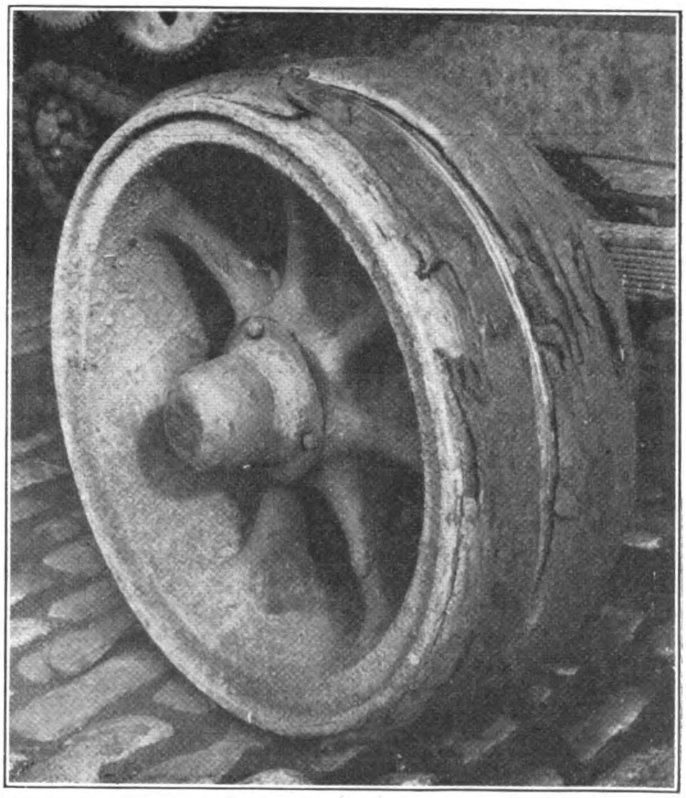

FIG. 7—REAR METAL WHEEL ON THE SAME AXLE AS THE WHEEL IN FIG. 6 AND ITS TIRE AT THE CONCLUSION OF THE TEST

by the weights per inch of tire width given in Table 3
(3) They had different drivers; often an important item
(4) The size of the back tires was different
(5) The load per inch of tire width varied
(6) They operated at different average speeds, though at the same maximum speed
(7) The springs of the chain-drive trucks were not the same as those on the worm-drive trucks

To compare, therefore, the results in tire wear of trucks Nos. 162 and 168 with those of trucks Nos. 172 and 175 is misleading on account of the variables. True comparisons are possible only where all essential conditions are the same. It was realized that this difficulty would confront any attempts to get the comparisons in tire service on wood and on cast metal wheels; hence the use of four different trucks and the alternate diagonal arrangement of the wheels. Here the average results are an indication of the relative service, because both front tires and both sets of rear tires on each truck were subjected to the same operating conditions, with the exception that possibly the tires on one side of each truck might have slightly more load to carry than those on the other side. To counter this exception the "hand" of the wheels was alternated on similar trucks; wood wheels being placed on the right-hand end of the front axle on one chain-drive and one worm-drive truck, but on the left-hand end of the front axle for the other chain-drive and worm-drive. This arrangement was followed also on the rear axles, as will be seen by referring to Table 1.

During the next 10 or 11 weeks the trucks were in active service, and on May 20 another series of tire measurements was taken. The inspection showed that all the tires seemed to be wearing uniformly. There were no large cuts, no hangnails, and no base-separations. Even from a casual glance, however, the tires on some of the metal wheels showed more wear than those on the wood wheels on the same truck. Figs. 4 and 5 give the individual rates of wear and mileages up to May 20.

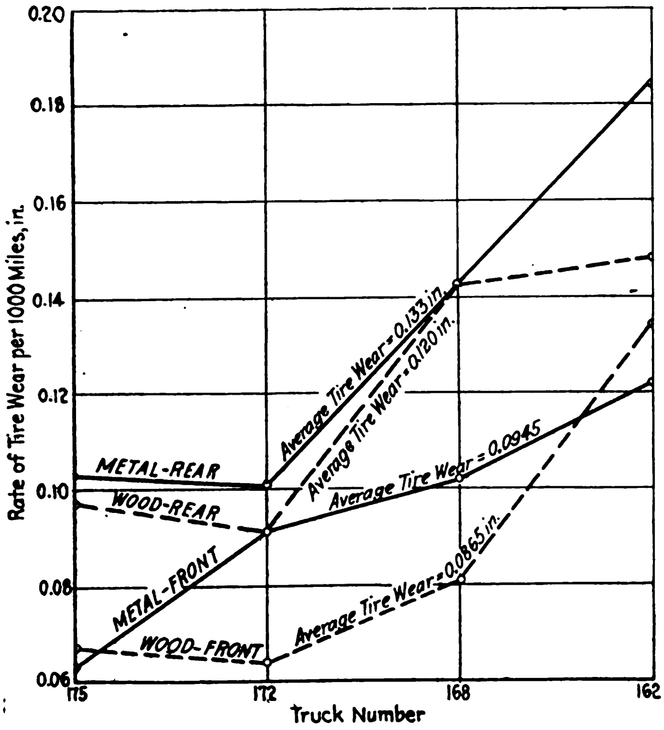

FIG. 8—RELATIVE RATES OF WEAR OF FRONT AND REAR TIRES MOUNTED ON METAL AND WOOD WHEELS PER 1000 MILES

Through the late summer and early fall there was comparatively little for these trucks to do, and it was not until some of the rear tires showed signs of disintegration that the next measurements were taken, on Dec. 1, 1921, a little over a year from the time the tires were new and first measured. These measurements showed that for the eight front tires the average wear of those mounted on metal wheels was 9 per cent greater than the wear on the tires mounted on wood wheels; also that for the 16 back tires the wear of those mounted on metal wheels was nearly 11 per cent greater than the wear of the tires mounted on wood wheels. These percentages are based on the average rates of wear per 1000 miles over the entire period. The charts show the cumulative results at the end of each period when measurements were taken.

Of the tires on rear metal wheels none exceeded the tires on the rear wood wheels in mileage. On truck No. 168 they were equal, but on the other three trucks the tires on the wood wheels gave superior results. Not only were the tires worn more on the metal wheels but they showed signs of disintegration to a much greater extent than those on the wood wheels. Late in the fall of 1921 the superintendent in charge of the operation of trucks, who had previously thought the metal wheels were the more desirable, stated that there was no use taking any more measurements, for anybody could see that the tires on the wood wheels were giving better service.

On the front wheels none of the tires had worn-out, but those on truck No. 162 showed that base separation had started. On the back wheels all the tires were well worn; some of them were shredded and about to be replaced, as can be seen from Figs. 6 and 7.

As the tires wore down and their cushioning effect diminished, it was noticeable that small failure cracks appeared sooner and developed into larger cracks in almost every case in the tires on the metal wheels. As the steel wheels in every case exceeded the wood wheels in weight it might be thought that this was due to the greater inertia of the heavier wheels. This may have been true to some extent, and it appears reasonable but, if that was the case, how can the differences in the tire mileage between the chain-drive trucks and the worm-drive trucks be explained? The worm-drive trucks gave the better results and the unsprung weight of their axles considerably exceeded that of the chain-drive. Neglecting wheels and tires the weight of the worm-drive rear-axle parts was about 1400 lb. greater than that of the chain-drive axle. These results need careful interpretation as will be indicated later.

UNSPRUNG WEIGHT

It is my opinion that the relative effect of a difference in unsprung weight is modified to a considerable extent by the presence of the springs above the axle and, of course, the speed at which the truck operates, as well as the thickness and hardness of the tire itself. In other words, against vertical impact there is the cushioning effect of the springs, and to some extent of the frame and load itself, as well as of the tire. The tire therefore does not have to withstand the full force of the road impact when delivered in a vertical plane. When the blow is delivered in a horizontal plane, however, the springs are not able to cushion the blow and the tire must absorb the full impact if the wheel is rigid. Neither is the tire designed to withstand sidewise or skid loads as efficiently as the vertical loads.

One of the important functions of a wheel is the abil-

ity to act to some extent as a cushion to the savage side-thrusts of rut and curb. The wood wheel is several times as resilient as the metal wheel and affords this protection, while the metal wheel does not flex to any appreciable extent. Thus with wood wheels the tire has some assistance from the resilient spoke at the time it needs it most. In the extreme case of a heavy side-skid against a curb, the wood wheel breaks, acting like the fuse in an overloaded electric circuit, and protecting more expensive parts.

Tire Wear Per Thousand Miles

Figs. 4 and 5 show the results of the test in terms of mileage and rate of wear per 1000 miles. As the results for the different trucks vary so much, it may be thought that the test was not a fair one because the individual tires themselves must have varied in composition. In this connection it should be noted that the rates of wear

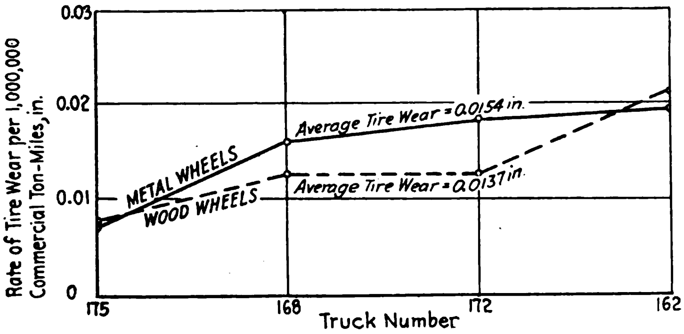

Fig. 9—Rate of Tire Wear per 1,000,000 Commercial Ton-Miles for Front Tires

of the tires of each truck form a distinct group. The differences in the rates of wear, therefore, are not due to differences in tires, but to differences in trucks, their drivers or operating conditions. Figs. 4 and 5 show this plainly. Fig. 8 shows the rates of wear per 1000 miles on the front and rear tires, plotted to show the relative difference between the tires on the wood and the metal wheels.

Figs. 9 and 10 show the rates of tire wear per 1,000,000 commercial ton-miles. Here we see that trucks Nos. 168 and 172 are not very far apart in tire wear when certain allowances are made. The front tires on truck No. 168 show 7 per cent less wear than those on No. 172. The rear tires on truck No. 172 show 15 per cent less wear than those on No. 168. It should be noted, however, that truck No. 172 had the advantage of 4 in. more in width of tire on the two rear wheels, which probably accounts for a great part of this difference. On the basis of commercial ton-miles we find that for the front wheels the rate of wear was about 12½ per cent greater for the tires mounted on metal wheels, while at the rear the rate of wear was about 13 per cent greater for the tires mounted on metal wheels.

Truck No. 175 was in charge of a very careful driver and in every way was kept in as nearly 100 per cent of first-class condition as was possible. During the last few months of the test this truck was hired out to some contractors, as not enough coal was being delivered to keep it in that service. On this contract work it covered greater distances than on the coal deliveries, and making extra trips it rolled up a big commercial ton-mileage. This is a case of exceptionally good performance by a carefully groomed truck, and a very good driver.

Fig. 11 gives a final comparison between the rates of wear per 1,000,000 commercial ton-miles of tires on wood

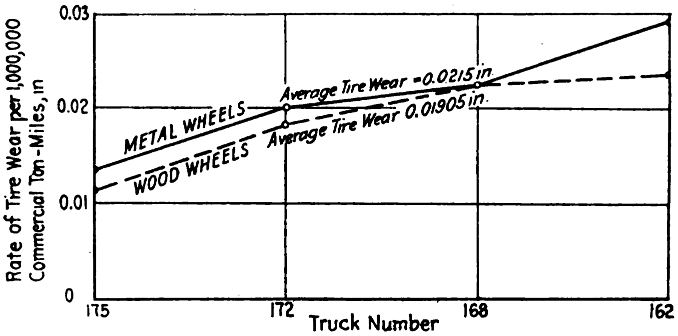

Fig. 10—Rate of Tire Wear per 1,000,000 Commercial Ton-Miles for Rear Tires

and metal wheels. It averages the rates of wear of the six tires on each truck, or 24 tires in all. Twelve of these were on wood wheels and 12 on metal wheels.

In setting forth the results of this test it is not intended to convey the impression that solid tires on wood wheels will give uniformly from 10 to 15 per cent better performances than on metal wheels. The number of trucks tested was far too small to justify such a conclusion. The results are strongly indicative, however, and cast grave doubts on the unsupported statements of those who have sold us steel wheels. It seems safe to say that the claims for metal wheels, cited early in this report, are by no means justified. Those who make such claims should demonstrate them beyond doubt by concrete performance.

The logic of lighter and more resilient wheels would indicate a better performance by the wood wheels, even in the event of no great amount of confirmative data. The claim for better performance by a metal wheel because it is more nearly round and true is really a point of minor importance, because the road inequalities so vastly exceed the very small variations that occur in a well-made wood wheel that even a worn tire will practically nullify them.

It has been claimed also on many occasions that solid tires wear less on metal wheels than on wood wheels, because the former "radiate" the heat generated in the tire at a more rapid rate than the latter. This is another theory that has not by any means been established. The test described above shows that the difference in the rate of wear was greatest in favor of the tires on the wood wheels during the heat of summer, directly contrary to

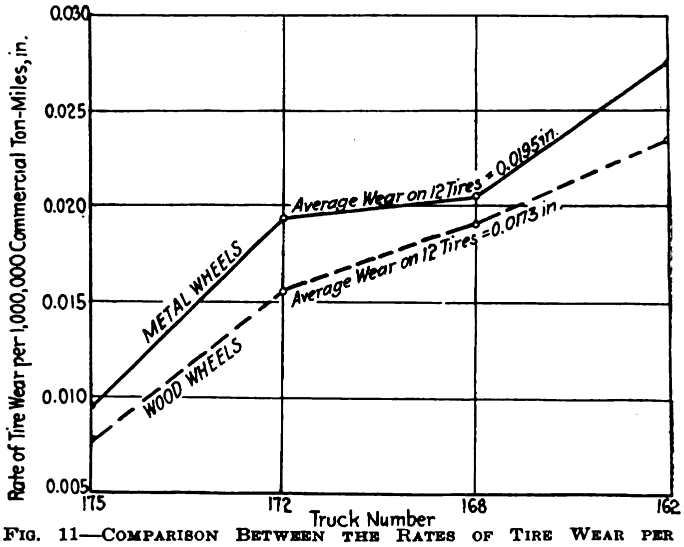

Fig. 11—Comparison Between the Rates of Tire Wear per 1,000,000 Commercial Ton-Miles of Tires on Metal and Wood Wheels

the above theory, when the metal wheels should have had the greater relative cooling effect and saved the tires. The theory is defective in that it assumes that the heat due to the rolling of the tire causes more rapid wear, and that the radiating factor of the wheel is a large factor in cooling the tire. However, the air surrounding the rubber is probably of major importance in cooling the tire; and if the metal wheels are painted, as is almost universally the case, the paint film probably offsets the ability of the metal itself to transfer heat from the tire-band to the air around the spokes.

It is true that during 1917, 1918 and 1919 some very poor wood wheels were produced and assembled under trucks, and these have been held up as horrible examples. These were produced under extenuating conditions, and are by no means typical of the millions produced prior to that time and still giving good service if they have had reasonable care.

In regard to weights, the cast-metal wheel is seldom as light as the ordinary wood wheel, which is really heavier now than it need be. This item also touches on the much bandied question of the relative importance of unsprung weight to the total load carried. This is most often cited in discussions of different types of rear axle. It is reasonable to suppose that the lighter the axle the less will be the force of the blow transmitted through the springs to the frame when the wheels encounter bumps in the road surface. The relative effect of heavier blows by heavier axles is not readily discernible on the load carried, however, if the truck has well-designed springs, especially if these springs are well lubricated. Then, too, the heavier the axle the greater the proportion of the blow absorbed in the tire, due to the greater inertia of the axle. So, it would seem that this would be accompanied by a greater tire wear under the heavier axle. But again there are no reliable supporting data.

Of the four trucks used by the coal dealer in the tire-mileage test already mentioned, two had worm-drive rear-axles, the heaviest type, and two had chain-drive rear-axles, the lightest type. The rear-axle loads per inch of tire width were about 7 per cent greater for the chain drive than for the worm drive. Leaving out all reference to unsprung weights, one would expect the tires on the chain-drive trucks to show somewhat more wear, perhaps 10 per cent, per commercial ton-mile. The total weight of the worm-drive axle, wheels, tires, etc., was 70 per cent in excess of the weight of similar parts on the chain-drive axle. If 1 lb. of weight below the springs is equivalent to 5 lb. above the springs, then the tires on the worm-drive axle were loaded 31 per cent more per inch of width than those on the chain-drive axle. If the more fanciful ratio of 10 to 1 is used, the excess becomes 68 per cent. In any event, if the proponents of the unsprung weight theories are to be believed, the tires on the worm-drive axle should get the worst of it to a substantial degree, but, as a matter of fact, careful records show that average rate of wear of the rear tires on the chain-drive trucks was 53 per cent in excess of those on the worm-drive trucks.

Certainly these results do not indicate that a light axle saves tires. However, neither do they prove the opposite, because the conditions of the test were made such as to afford comparative data on the wheel service, and were *not* such as to enable one to draw fair comparisons on an unsprung-weight basis. They are cited as an example of data that are being submitted repeatedly in advocating one type of design over another in making truck sales, and buyers should be on guard against such statistics and comparisons.

Nothing in this paper should be construed as deprecating the importance of reducing chassis and body weights to a minimum. For every idle pound of such weight removed, a pound of useful load can be added. Weight-reduction, however, does not always mean a proportionate gasoline or tire saving. Other factors enter into these items of expense, and those who generalize expose themselves to the same error as those who have allowed their imaginations to over-emphasize the importance of a minimum unsprung-weight. To date there are no data on which to make safe comparisons, and a full exposition of the subject may establish a different feeling than that which exists at present.

Engine weights can be reduced if improvements in fuel and lubrication allow us to use higher compressions and maintain oil-films on rubbing surfaces. The air-cooled engine will effect a considerable weight-reduction if perfected for motor-truck service, and this is not at all impossible.

For years the intensive study given to the main units of the chassis, engine, radiator, gearbox, axles, frame, springs, steering-gear, etc., has absorbed nearly all the efforts of designers. The details by which these units were assembled in the chassis, kept in proper relation, and controlled, have suffered by comparison. As a consequence they have been responsible for many breakdowns and a heavy upkeep expense. At present these details are receiving closer attention but there is still room for improvement.

CHASSIS LUBRICATION

An outstanding subject at the present time is that of chassis lubrication, the lubrication of the numerous small pins and bearings connecting the working units of the chassis. All mechanical motion, in the millions of forms that surround us on every hand, depends upon good lubrication for its continued existence. The lubrication of steering pivots and connections, universal-joints, spring-bolts, pedal-shafts, brake-shafts, radius-rod pins, clutch-release bearings, unit support pivots, etc., of motor trucks, in most cases has been cared for crudely in the chassis design. Consequently these parts have been subject to severe wear and entailed heavy upkeep expense. Lack of good lubrication in engines, gearboxes or rear axles is quickly attended with an ominous noise and dire disaster; hence the studious development for these units of lubricating devices and systems that are practically automatic in operation and require infrequent attention. Engines will go from 200 to 500 miles on a charge of oil. Gearboxes and axles will go 3000 to 5000 miles on a charge of lubricant.

When we turn to the consideration of the chassis bearings mentioned above we find most of them equipped with lubricating devices of the crudest sort, in spite of the fact that in many cases, on spring-bolts and universal-joints, the pressures per square inch are as high as those encountered on the crankshaft bearings of the engine. In addition, the ends of the bearings are, for the most part, exposed to the dust and mud of the road, while atmospheric moisture can enter almost at will. As a result these bearings deteriorate rapidly, entailing gradual loss of power, hard riding, difficulty of control, spring breakages, squeaks, rattles and a general looseness that adversely affects other parts of the chassis. The fact that the chassis bearings can function at all under these conditions, and neither produce a startling outcry nor actually breakdown for some time, has been responsible for much of the lubricating neglect on the part of both the manufacturer and the user.

The user pays a big bill for this neglect, however, and the truck builder feels the reflection of it in sales resistance. In this Country the estimated direct cost of truck repairs due primarily and secondarily to poor chassis lubrication is between $15,000,000 and $20,000,000 per year, a truly staggering bill. On the other hand, at only 50c. per hr. the labor charge to lubricate properly our trucks with the devices now furnished, would cost over $40,000,000 per year. It can be seen readily, therefore, that the truck user has actually been justified to some extent in his neglect of these parts, for, even if he gave them the attention that the builder desires, he would still have some repair bills, say $5,000,000 per year, and the total direct economic loss would exceed $25,000,000 annually.

The indirect loss to the user, the loss of earnings while the trucks are being repaired, runs from $30,000,000 to $35,000,000 annually. This is not always taken into consideration by owners; otherwise they would pay more systematic attention to chassis lubrication, and make a more insistent demand upon truck manufacturers for means to allay this heavy expense.

Except at one or two points, grease as a chassis lubricant has been acknowledged by close students to be inferior to oil and it is becoming less popular. Grease is a dirt collector and carrier; it is not uniform and is only part lubricant; it will not spread over a bearing by capillarity; it can be forced over the slack side of an oscillating bearing without reaching the load side, and as it exudes from the ends of the bearing it gives the misleading impression that the bearing is lubricated; it entails a high labor charge for application. Oil is a much better lubricant, for it carries no inert matter; even if dirty it can be filtered as it is fed; it spreads completely over a bearing by capillarity; it can be fed automatically; it can be led from one point to another in series; it is cheaper. The average oil-cup, grease-cup or fitting, however, is an excrescence, easily damaged or broken off. It admits dirt and moisture to the bearing, and its repeated filling calls for a heavy labor item.

An early attempt to embody thorough and automatic chassis lubrication was made in 1916 by a Pacific coast truck builder who embodied small oil-reservoirs, or magazines, in the brackets that held the chassis bearings. In this system the oil is filtered and fed by wicks to the bearings whenever the motion of the bearing calls for it. About once a month the reservoirs are filled from a spout can. This magazine system was so satisfactory that the 20,000 Class B Trucks for the United States Army were so equipped. After 3½ years of hard service in the New York district 300 of these trucks were overhauled, and the inspection showed that less than ½ per cent of all the chassis bearings fed by the magazines showed appreciable wear. This is a vast improvement over the usual condition of such bearings, which in many cases must be renewed entirely, both pins and bushings, after 2 years' service. The surplus oil from the spring-bolts spread down the springs, kept them soft and flexible, and practically eliminated spring breakage. A magazine type of lubricator for spring-shackkles that is equipped with a rapid fill opening is illustrated in Fig. 12.

By piping the various magazines to a few convenient points at the side of the chassis, they can be filled in less than 10 min. The first applications of this system had no provision for regulating the feed, which was much more than sufficient. Later improvements regulate this so that the magazines need not be filled oftener than once in 2 months. The national labor charge for truck chassis lubrication can be reduced in this manner to $500,000 per year. The chassis-bearing repair-bill can be cut to an insignificant figure, as can also the lay-up loss due to these repairs. A conservative estimate of the direct and indirect savings on our 1,010,700 commercial cars is $50,000,000 per year. This alone would buy 15,000 new chassis. It would buy 20,000 chassis if these were built in one plant each year.

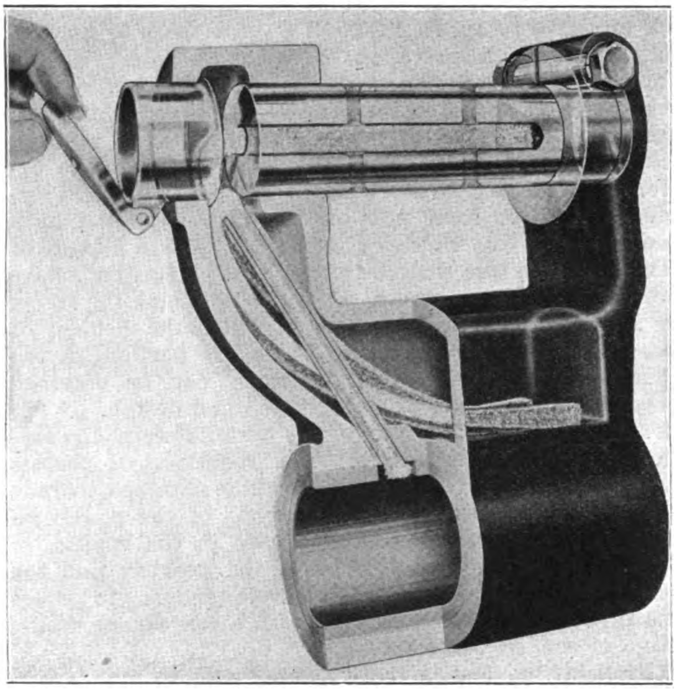

Fig. 12—A Magazine Lubricator for Spring-Shackles with a Rapid Fill Opening

The next few years will likely see much active competition in motor-truck service. Refinement of detail and simplification of operating features will be emphasized.

Many buyers today are not sure as to what features are essential and which ones are non-essential. Few buyers have any fundamental knowledge of motor-truck design, or the compromises necessary to produce a successful design. In purchasing they stress this or that point without absolutely knowing its value or the effect of it on the cost and operation of the truck as a whole. This makes for differences of opinion among builders as to what the trade wants. Neither buyer nor seller gets any permanent benefit under these conditions.

Frequently the buyers' experience does not qualify them to pass judgment on the propositions that are offered. Once the full picture is before them, there are few executives who are not qualified to judge the situation and make a proper decision. Such decisions would soon clarify the situation for both buyer and seller.

In the long run motor-truck design should tend toward such standardization as has been adopted to so great an economic advantage by the Master Car Builders' Association, which permits railroad rolling-stock to be repaired quickly and cheaply in any part of the Country. This need not restrict initiative or halt improvement, but will make any innovations conform to good practice, as determined by careful analysis of cause and effect, and long study in the application of the results.

THE DISCUSSION

President B. B. Bachman: — Regarding the brief analysis that Mr. Myers made as to the relative stiffness and strength and the uses of higher grade material, I

am prepared to agree, although I think there is a possibility of misinterpretation of what he said. We all know that the technical points that he brings out regarding the form of a structure and its effect upon stiffness, and the lack of effect of the quality of material, are well taken; however, I feel that Mr. Myers has gone to extreme lengths to a large extent in this paper in endeavoring to clear up some possible misstatements that have been made in the endeavor to market certain classes of product. I believe that the advantages of elasticity have not been fully realized and, while I do not think the frame or the front axle are members in which the element of elasticity is a desirable factor, there are portions of the chassis structure in which elasticity can be obtained properly by the form of construction and design.

The Hotchkiss drive, in itself, permits weight reduction in the chassis design. The reduction of chassis weights and the advantages that can be obtained thereby are a result of study and an analysis of the power requirements and the engine size used in the vehicle, in conjunction with the gear-ratios, the gearbox and the rear-axle. Such considerations have a tendency to reduce the tare weight of the chassis, with a consequent realization of economy in operation.

Probably no one is more sympathetic to standardization than I am. I have had the honor of being Chairman of the Standards Committee for 4 years, and I have been associated with the standards work in one capacity or another for many years, practically since the inception of that work in the Society. But I do not concur fully with what Mr. Myers says with regard to the desirability of the standardization he outlined. If I interpret correctly the closing paragraphs of his paper, he is making a plea for interchangeability of essential components of motor vehicles, that would undoubtedly be detrimental to initiative and the development of design.

He uses as an analogy the work accomplished in railroad circles by the Master Car Builders Association. In railroad practice a freight car may leave New York City with a load and, without having its load changed, cross the Continent, traversing an air-line distance of 3000 miles, and possibly going many hundreds or even thousands of miles more than that, dependent upon the methods of routing. In that time it would pass over a number of different railroad systems. It is evident that there should be a considerable degree of interchangeability. The most obvious standard is that of the gage of the tracks. In the matter of repairs in the various roundhouses the stocks carried must be interchangeable. In spite of what we have seen in the past few years, under abnormal conditions, of long truck-transport, it does not seem to me that we will ever approach a condition entirely analogous to that of the railroads. While the benefits of interchangeability that Mr. Myers points out have a certain foundation and can be defended along logical lines, I believe there is a reverse side of the picture.

There is room for reasonable differences of opinion on the matter, without any actual antagonism from either side. It is possible that, as the motor-truck industry gets out of its swaddling clothes, some of the things Mr. Myers has outlined will come to pass. Certainly, history tells us that the views of men of vision are needed to point out things that some of us who are more closely tied to details think cannot be accomplished.

In regard to the subject of wheels, Mr. Myers has reported the results of some tests with the idea, as I understand it, of refuting certain sales statements that have been made. The data he has brought forward would not sway me one way or the other. I feel that the results which he points to, and which he has definitely qualified, as secured from a small number of trucks, really do not tell us anything. At the same time the claims he is attempting to refute have not, so far as I know, been established otherwise than from a controversial standpoint.

On the other side of the case, Mr. Myers refers to the fact that during the last few years an unsatisfactory quality of wood wheel was manufactured and sold with disastrous results. That factor alone is, I believe, responsible for the rise of the metal wheel to the quantity production it has attained. The only fear I have had with regard to the wood wheel is from the standpoint of the felloe stock. I have had no fear of the spoke stock. The felloe stock, particularly in the case of large-sized wheels, should be improved. Designs have been proposed, and some of them have had a considerable amount of practical trial. That appears to eliminate the difficulty of wooden felloes and offers the possibility of satisfactory service from substituting steel while retaining the essential characteristics of the wood wheel. I believe that the serviceability and satisfactoriness of the wood wheel and of the metal wheel are dependent more upon honest workmanship and good quality of material than upon anything else. Either construction will give good service under proper conditions.

The questions and theories that have been raised with regard to flexibility, radiation of heat and the like are all matters of conjecture and possibly matters for future research. So far as I know, none of us has had time to investigate and obtain data to substantiate any one of these various thoughts or theories.

C. T. MYERS:—Our chief problem in truck design is to secure a construction that will offset impact blows. The more flexible and more resilient the construction is, the better it will resist impact blows. I agree with Mr. Bachman that the frame weights and all other weights should be cut down to the limit. The point I tried to make is that there are many people who do not know what they are talking about at all and never had to spend months and years in getting a truck on the road and keeping it there, but make claims of the possibility of vast weight-reductions that, so far as we can see at present, are not tenable. Some of them claim to reduce the weight of a motor truck by one-third. That is absurd. Accomplishing a 10 to 15-per cent weight-reduction in the average truck is doing exceedingly well and it takes extremely good designing to do that without an excessive increase in cost.

As to standardization, I have gone the full distance of what I think are the possibilities. There is considerable conjecture as to the probabilities. I believe that there is a fair analogy between the Master Car Builders' standards and those of the motor truck; not that many motor trucks will cross the Continent in a week or two and demand service in San Francisco, but that trucks of a dozen different makes will be built and sent to San Francisco, to Canada, to the South, to Australia, South Africa, India and elsewhere. Why should each truck builder be called upon to maintain his trucks at every point where he sells a truck? Each manufacturer must do this if he is to keep his trucks in service there and hold his business. Why should each one be called upon to maintain wheels that fit no truck but his own, to maintain bearings in those wheels that vary only slightly from those made by the other manufacturers, different steering-pivots, bushings and a hundred and one little details that do not make a particle of difference if they can be picked out from a series of standard sizes when he designs and builds his truck? I am a designer myself

and I dislike to be hampered, but I have been up against the practical side of designing in the past several years, particularly from the motor-truck user's side. The user is certainly in a serious predicament when out of touch with satisfactory service parts; it is maddening to find that one cannot replace some particular part because it differs by a few thousandths of an inch from a similar part that is available.

As to claims concerning wheels, I particularly state that what I have set forth does not prove the case finally. I have used data from carefully made tests, those of a man that intended to equip all his trucks with steel wheels. I realize that all sorts of questions and doubts can arise in the minds of the people who are interested in this matter, because I have done much research work for people who make wood wheels; and it was stated to them several years ago that the steel-wheel claims would not hold water. At about that time I was acting in a consulting-engineering capacity for three different truck builders and this question of wood or steel wheels came up repeatedly. The statements of most steel-wheel salesmen simply cannot stand up against careful analyses. There are some things about wheel design and use that are not fully appreciated in the industry. The same condition exists with regard to a great many other details in a motor truck. But we should know these things. The buyer must know them; otherwise he cannot express his desires. He is the one who pays the bills, not the motor-truck builder.

RUSSELL HOOPES:—Other than the tests described by Mr. Myers, I do not know of authentic data of this kind. As Mr. Myers says, we have heard many statements but have not seen them supported by facts.

We hope that the good work started with the standardization of axles and hubs will be carried on and that the wheel standardization will follow, so that a truck user can stop at any service station and get a new wheel that will fit his axle and hub equipment, and tires that have been standard for many years.

Considering some of the recent remarkable results secured with aluminum alloys, I believe that we will use aluminum for parts of trucks that heretofore have been under too severe a strain and too expensive. Aluminum will save decidedly in weight, but it may be too expensive to use extensively. In regard to the question of unsprung weight, I hope that in some way additional information will be available.

The paper gives the result of tire tests comparing the cast-steel wheel and the wooden-spoke and wooden-felloe wheel. The company in which I am interested makes a wooden-spoke and wooden-felloe wheel of the best possible construction and has found that the wood felloe is a source of weakness. To overcome this it has used a comparatively simple method of inserting a flat-plate washer between the spoke-end and the counterbore in the wood felloe, and has never had the least trouble with this. The same results have been obtained by the Pierce-Arrow company, in its form of spoke sockets or shoes, and also by the White Motor Co. in making its wooden-spoke and wooden-felloe wheels. The latter company accomplished the same results by using about twice as much timber in the spoke at the felloe shoulder as was necessary to make the head. This gave the spoke ample bearing so that it did not become felloe-bound after hard service. Our company has felt that the metal-felloe truck-wheel has altogether eliminated the wood-felloe.

The matter of chassis lubrication is an essential feature that is commonly neglected. As Mr. Myers shows, an enormous saving can be made by proper practice. Oil is much superior to grease.

Concerning the weights of wheels in connection with unsprung weight, the wood-felloe 36 x 12-in. wheel such as we used to make weighed 216 lb. The weight of the 36 x 12-in. metal-felloe wheel we make is 166 lb. The weight of a 36 x 8-in. metal-felloe wheel, with 2½-in. spokes, is 101 lb.; whereas our type of wood-felloe 36 x 8-in. wheel would weigh 140 lb. In our 36 x 12-in. wheel the metal-felloe makes a saving over the wood-felloe wheel of 50 lb. The metal-felloe 36 x 8-in. wheel saves 39 lb. over the same size of wood-felloe wheel.

These facts demonstrate that the S.A.E. Standard wood-felloe depths have been unnecessarily great since the advent of the metal-base tire that has superseded the fabric base. The tire base strengthens any wheel on which it is placed so much more than the old fabric base that the S.A.E. Standard dual-felloe depths probably could be reduced 25 per cent and the single-felloe depths 50 per cent; in fact, where the single S.A.E. Standard bands are ⅜-in. thick, they probably are strong enough to stand the work without any wood felloe, when a pressed-on tire is mounted.

The wheel weights I have stated are correct, but different wheels will vary in weight on account of the difference in the specific gravity of the various kinds of timber used in the wheel; also, there is a variation in weight of the different pieces of the same rolling of steel used in the felloe band, in the drop-forged sockets and in the plate washers.

MR. MYERS:—In regard to the point that the S.A.E. Standard wood-felloe thickness could be reduced by 25 per cent in some cases and 50 per cent in others, it seems to me that it is advisable to use the thickness specified in the S.A.E. Standards at present, or else use no wood at all. In assembling a wheel, if we have too thin a wood felloe it will not withstand the pressure of the press in which the wheel is put together. It seems to me that it is somewhat risky to advocate cutting 50 per cent off of the thickness of the felloes, and that we should use the present standard until we shall have experimented further with light felloes.

J. H. WAGENHORST:—My experience has been in connection with the substitute for the wood-felloe wheel that Mr. Bachman mentioned. I started on this work in 1913. It consisted of placing steel-felloe wheels instead of wood-felloe wheels on passenger cars, principally on the large-sized No. 66 Pierce-Arrow cars. The result of that development today is that practically all of the wheels that are used on passenger cars now have steel felloes. Our production alone has been about 2,000,000 sets in the last three years, and that may represent one-half or probably somewhat more than one-half of the steel-felloe wheels being used by automobile builders. The Packard, Cadillac, Hudson, Buick, Nash, Overland and Ford companies are using steel-felloe wheels exclusively. Our own factory is making steel-felloe wheels entirely. The wheel itself is considerably lighter than the wood-felloe type, speaking of demountables, ranging from 6 to 16 lb., depending on the size. The average increase in strength is approximately 25 per cent. There is very little change in wood lengthwise with the grain; there is a considerable change across the section. We have taken spokes and kiln-dried them to 4 per cent of moisture, measured them very accurately, immersed them in water for 2 weeks and found, on the second measurement, that there is practically no change in length overall; whereas the cross-sectional change runs as much as ⅛, or possibly ⅞ to 1 in. The crushing strength across the grain is about one-sixth to one-eighth as compared to lengthwise strength.

One of the important questions that was brought up

originally in regard to steel-felloe wheels was that of shear-off of the spoke. All of these spokes are supplied with a fillet, that is the natural result in form, and a socket in a steel-felloe base, so we have it somewhat smaller in section, although at the point of shear in the steel it has a considerably greater section, giving a greater endwise area as well as shearing strength.

A MEMBER:—In looking over Fig. 11 of Mr. Myers' paper, I note that truck No. 175 averaged about 0.008 in.; truck No. 172 about 0.017 in.; truck No. 168 about 0.019 in.; and truck No. 162 about 0.026 in. of wear per 1,000,000 commercial ton-miles. Mr. Myers pointed out that truck No. 175 was driven by an extremely careful man. I wonder if there is any significance in connection with that. It seems to me that possibly an analogy can be drawn there between a very careful man's driving and the well-known behavior of a certain car on tires. I know that it is impossible to slip or spin the wheels in starting a certain car from a standing start under any condition that I have encountered. Does he not think that the tire wear is more the result of braking and accelerating than of the load imposed on the tires themselves?

MR. MYER:—It is not at all fair to compare the wear of the tires between trucks; the conditions were not the same. I give several different variables in my paper. The tests were made as carefully as possible to get some comparative data on wheel service. If the steel wheels were better on account of being more nearly round, it ought to show up as an average on 20 front tires, particularly when we were so careful about the tires. I cannot say that those tires were all exactly the same; I do not believe they were, but I do believe they were very nearly the same. As I pointed out, a comparison of the diagrams shows that all of the tires wore in separate and distinct groups, evidently controlled by the trucks on which they were operated. Truck No. 175 gave the best results; truck No. 172 was next, and truck No. 162 was last. There were marked differences between the worm-drive and the chain-drive trucks on which the rate of wear was greatest, and it is not at all fair to compare the tire wear. What I have given in the paper was just an indication that minimum unsprung weight is by no means all of the story.

Mr. Bachman will acknowledge that as soon as he tries to change the design of anything in his two-cylinder truck very much, he has to change the whole truck. I have seen a change in the size of a bolt in a truck modify 28 other different parts, some of them 6 to 8 ft. distant from the bolt that was changed. The inter-relation of parts in a motor truck is very intimate, the compromises are manifold and good judgment and long experience are needed before one is able to make a compromise. Many conclusions are being drawn from insufficient evidence.

W. B. BUTTERICK:—Has Mr. Myers seen the steel wheel used on the London General Omnibus Co.'s vehicles? It is not by any means so bulky a steel wheel as that which he illustrated. The London General Omnibus Co. operates about 4500 buses on the streets of London and all have steel wheels. I think one will not find a wood wheel manufactured in Great Britain; there are practically none on any truck in London. I find that the London General Omnibus Co. has reduced its tire costs 2d. (4 cents) per mile over those obtaining with wooden wheels, since they began using steel wheels. In the case of the omnibus, the wheel is built especially light in weight for that purpose only. The vehicle has an ash frame between two steel clinchers or flitches. Possibly that is the reason why they get a better tire mileage, with the resiliency in the frame, not at the wheels.

MR. MYERS:—The wheels used by the London General Omnibus Co. are very light steel wheels, but the difference in the tire cost cannot be attributed entirely to steel wheels. When it changed from wood to steel wheels it also changed a number of other things. The Fifth Avenue Coach Co., in New York City, uses a design very similar to that used by the London General Omnibus Co. It uses a steel wheel that is very well designed and very light.

There is this difference to be considered in regard to the steel wheels used by these two companies and the steel wheels used by the average motor-truck builder. The Fifth Avenue Coach Co. runs its vehicles on a smooth street and straight ahead all the time; but the London General Omnibus Co.'s vehicles run on rougher roads. Some of the streets have cobblestone pavements and the vehicles have more side-thrust to withstand, but not anywhere near the amount of side-thrust that a truck in regular service has. The tire-mileage cost of the Fifth Avenue Coach Co. was reduced greatly between the time it used wood wheels some 6 years ago and the present; but tires have been improved within that time to an extent where they will give from 70 to 80 per cent greater mileage regardless of the type of wheel, and they have also been reduced in price, so that one cannot get any direct comparison on the wheels themselves. The prime reason so many steel wheels are used in truck construction abroad is that they have great difficulty in getting good wood. It was very difficult to get good hardwood in this Country a few years ago; the Government has requisitioned practically all of the hardwood. However, the Saurer Co., a leading foreign motor-truck builder, still uses wood wheels; some of the wheels illustrated in my paper were made by that builder. That company also uses fewer spokes in its wood wheels than we use in this Country, to obtain the maximum resiliency. This company makes its own wheels. When a wood wheel is made properly, it is superior to the steel wheel, because of its greater resiliency. The French and the Swiss had a sad experience with the steel tire a few years ago, when they used motor trucks over cobblestones. The steel tire produced so severe an impact all the time that buildings deteriorated from great cracks in the foundations, all along the line of travel, and they had to stop using the steel tire. The Saurer Co. appreciated the fact that great resiliency is essential in tires and was the only builder that did not use a steel tire.

MR. BUTTERICK:—Why do not the White and the Pierce-Arrow companies use the wood wheel? I believe they said they could not get any suitable wood.

MR. MYERS:—That is exactly the Pierce-Arrow company's reason. This company made its own wheels and they were fine ones. Records showed wheels that had been in service from 1 to 5 years in the Arizona and New Mexico districts where some wheels dry out in a few months and go to pieces, but the Pierce-Arrow company's wood wheels had practically a perfect record of service. But those wheels were properly dried, finely finished, excellently fabricated and thoroughly painted. Two coats of primer were put on them as soon as they were fabricated and before the wheel went out for test. Before that truck got away from the plant it had five coats of paint. It was a good truck and the man who bought it took pride in it and painted it from time to time. All material must be used with respect to its characteristics, and moisture must be kept out of a wood wheel. It can be done without any trouble if one recognizes that need and cares for it. I seem to be a champion of the wood wheel, but I do not want to take that attitude. I am trying to face facts. Let the man who has the facts

of the matter in his possession present them and let us use them as we can.

J. E. WOLFF:—Very little good hickory is grown abroad. I believe this is the chief reason that more wood wheels have not been used in London. Aside from that, we have supplied several hundred sets of wood wheels to foreign companies. These have been very satisfactory, as there has not been one complaint; so, there are wood wheels abroad as well as steel wheels.

ERWIN L. SCHWATT:—Mr. Myers' paper states that "the average rate of wear of the rear tires on the chain-drive trucks was 53 per cent in excess of those on the worm-drive trucks." It seems that, when the chain is under tension, the wheels will slip a little when the loaded frame changes its position relative to the axle due to the roughness of the road.

MR. MYERS:—The figures are not truly comparative between trucks, or for unsprung weights; but if a man says that the truck with the light axle will show decreased tire wear, how does he reconcile that statement with this condition where it does show increased tire wear? They did not measure any clutch performance or check up on the drivers. They sent each driver out on a truck equipped with a wood wheel on one end of each axle and a steel wheel on the other. The driver did not know what was going on at all, and operated the truck just as he thought he ought to operate it.

MR. SCHWATT:—That skidding action probably is the reason for the excessive wear, and the relative weights of the rear ends do not enter into it. In the case of the worm drive we have the possibility for a skid, and also in the chain drive there is an actual skid.

MR. MYERS:—That *might* be true, but that is just one of a number of variables that can enter into this matter. How can anybody make flat statements that this wheel or that wheel can accomplish all these economies when so many other variables can more than offset any possible saving?

MR. BUTTERICK:—The Albion truck people were using the method of automatic lubrication with a wick in 1908.

MR. MYERS:—Possibly. However, they did not make substantial magazine brackets, but used oil-cups which are excrescences. When one got knocked off, it was forgotten that its pin needed any oil.

OUR HIGHWAYS

THE United States is founded upon the possibility of maintaining a sufficient and efficient transportation system.

In 1921 our estimate of the accumulated investment in highways for the 11-year period was $2,526,000,000, exclusive of the amount we have spent for maintenance each year. The estimate of rolling stock values is $8,790,000,000. That is, there has been over three times as much investment in rolling stock between 1910 and 1921 as we have expended during the same period for highway construction.

I estimate that it costs the American individual, each man, woman and child in the United States, about 1 cent per day for the highways. Our estimate of the amount that the Federal Government contributed last year toward the total bill for highways was only about 14 per cent, and the portion of the cost that the automobile paid in direct taxes we figure at about 19 per cent. That is, the Federal Government and the automobile paid about 33 per cent of our total bill. That leaves 67 per cent to be paid from other sources, largely taxes, partially those due to the floating of issues of bonds and partially direct levies.

If our estimate as to the investment in motor vehicles is somewhere near right at $8,790,000,000, a 20-per cent depreciation of the motor vehicle is over $1,600,000,000 per year. I do not know how much of that depreciation we can prevent by improving the roads, but if we can cut the bill for repairs by $600,000,000 that would be enough to pay our total road bill for last year.

It will not be increasingly difficult to carry the traffic over the highways. It will be increasingly easy if we study out the scientific principles that we must know and apply in order to carry heavy loads upon soils such as we have and which vary from place to place in the United States. If we were building all roads over good gravel or over good sand subgrades, we would not need to worry about how heavy the motor vehicles become, because all we would need is a thin slab on top to take care of the wear under the wheels; the subgrade would do the rest.

I want to contradict the oft-heard remark that our highways are going to pieces now. During the war practically all maintenance was withdrawn from the highways during the period when we increased our rolling stock over them more than 1200 per cent. It has taken the whole time since the war period to bring those highways back. Generally the highways are in better condition to-day than they have ever been before, notwithstanding our rolling stock over them has increased more than 1800 per cent.

The Federal-Aid Highway system will not for many years extend beyond the system of main highways.—T. H. MacDonald.

FOREIGN TRADE

THE inability of the cotton planters and the farmers of America to sell their produce abroad last year induced the American Government to reconstitute the War Finance Corporation and to create export credit to facilitate exports sales, and the ability of Continental Europe to buy cotton and other produce in recent months in part arises from the assistance afforded to them by this corporation. But the assistance granted is only of a temporary character, and this method of meeting the difficulty offers no permanent solution of the problem. Until steps are taken to induce the American people to buy foreign goods more freely, so that American exports and interest can be paid for by imports of goods, the problem will, of necessity, remain unsolved.—Sir George Paish.

Practical Road Construction

By R. C. Schoen[1] Minneapolis Tractor Meeting Paper

THE viewpoint from which the paper is presented is that of the engineer-contractor. Since mobility of a road-construction outfit is of prime importance, the camp equipment described was designed to meet this requirement in an unique manner, dependence for power being placed upon the track-laying type of tractor.

Because of having used both tractor and horse-drawn outfits on the same work, the author has had an exceptional opportunity to make comparisons between them. This he does in considerable detail, daily cost data being presented in tabular form, and finds that the evidence favors the tractor for use in practical road construction. The subjects discussed include finishing road work, tractor operation, daily output of work, repairs, wheeled tractors, lubricants, earth removal and grading machinery.

THIS paper on practical road construction is presented from the viewpoint of an engineer who entered the field as a contractor in that line. It includes some of the impressions acquired during 3 years of experience in the operation of outfits on such work. I entered this field after nearly 20 years of experience as a construction engineer, having had active charge of railroad and highway work during a large part of that time. Naturally, I felt that I was familiar with all the intricacies of operation; but, I am willing to admit that within a short time I found there were numerous things I did not know, and I have been brought to a realization of that fact many times since.

Camp Equipment

Camp equipment must be the first consideration in the organization of a practical and efficient road-building outfit in which tractors are to be the chief power. Highway-construction conditions are very different from those surrounding railroad construction. There are transportation advantages in railroad work, especially on reconstruction, that do not appear in road work. A road job may be, and probably is, located several miles from a railroad. This means an overland move. In all likelihood the next road job will be located several miles distant, perhaps on another line of railroad; so, one cannot afford to ship an outfit a long distance, especially when the tariff is high. Therefore, the mobility of the outfit must be one of the chief considerations. The units must be handled easily and arranged for coupling into a train. Tents for horses and men were much used formerly, but these are high priced and are quickly destroyed, especially in the prairie country where the wind blows almost continually. I have not used tents in the last 2 years. The men are housed and fed in cars. These are generally about 12 x 24 ft. in size, built fairly light and well braced. The cars are mounted substantially on separate trucks and, when placed in a train for moving, are hooked to a heavy cable which runs underneath the entire string. No strain due to pulling one car by another is placed on the individual vehicles.

Up to the present, I have not motorized the earth-hauling equipment. Tractors are used only on the graders; horses on the wagons. Horses, of course, need barns. The designing of barns that could be easily moved presented one of our greatest difficulties. During the first two seasons we used sectional barns mounted on skids. On short moves, a tractor would drag these barns, one at a time, to the new location, straining them badly in the process. On long moves they were knocked down, having been built in sections which could be handled by four men. They were then reassembled and set up at the new location. This method was unsatisfactory and the buildings were practically useless after one season.

We then designed a barn that could be mounted on wheels and have since made one that I consider thoroughly satisfactory. I now have barns mounted on wheels that can be prepared for movement in 10 min.; they can be reset almost as quickly. These barns are 26 x 32 ft. in size and can accommodate 12 horses comfortably. A half section of the roof on either side is made to drop down. This makes the width while moving less than 12 ft. and avoids obstructing the highway; it also permits loading and shipping on flat cars if so desired. They can be coupled together and placed in a tractor train, one being hitched to another, without danger of being pulled apart. Thus every building on the camp can be placed in a train behind the tractor and moved at one time, leaving the lighter equipment and odds and ends to be brought up by the horses. Last season with such an outfit I made one move of 32 miles in 1 day. Later, I made two moves of 24 miles each in 1 day and experienced no trouble. The same outfit could not have been moved in the same time by horses. The power in this instance was furnished by a tractor of the track-laying type.

Tractor versus Horse-Drawn Equipment

During the last season, I was well situated for making a comparison in the operation of outfits handled by tractor and by horses, having one outfit that was handled each way. These outfits were operating in the same district all summer and, for a time, they were on the same job, so that the soil conditions were identical.

Aside this one job, the tractor had the most difficult work; in one place the soil was very sandy and stony and in another it was heavy clay containing a large percentage of boulders. The month of June was extremely hot and dry and we were operating in the sand during that period. The wagons suffered severely here, because they sank deeply into the sand and were a heavy drag even when empty. The heat and dust made breathing difficult and there were times when the machines were enveloped in such a heavy cloud of dust that they could not be seen a short distance away. Nevertheless, the machines kept going and would have worked at maximum efficiency throughout if the wagons could have stood up under the strain; but we were compelled to slow up. I have often thought that a smooth running, completely motorized outfit would have finished this job in one-third the time.

On the two jobs mentioned, we encountered many boulders deeply embedded in the soil. Some of these boulders weighed several tons and, without adequate power on the job, it would have been necessary to break them with explosives in order to handle them. They were nearly all removed by the tractor by the simple process of digging around them, putting on a heavy chain and snak-

[1] Highway contractor, Ortonville, Minn.

ing them out of the way. Boulders which could not possibly have been handled by horses were thus removed with a minimum loss of time.

On one of the jobs the road passed through a small lake which was about 1200 ft. wide. The water reached a depth of 10 ft. or more for a large part of that distance. A special bulldozer was designed and built to push the material ahead. This contrivance had a blade or pusher 8 ft. wide, was heavily built and performed its work well, with four heavy horses for power; but, in my opinion, a small tractor would have been better in such a case than horses and I would like to see a practical machine built for use with a small track-laying tractor.

The soil on the job where the horse-drawn outfit and the tractor both worked at the same time was a heavy gumbo. The cuts were light and much of the material was taken from the sides and hauled long distances. Here I had a chance to make a comparison of performance, and that comparison was very favorable to the tractor. On this class of work, which consisted of a series of short cuts and long light embankments, turning the machine was an important item. On several occasions, I found that the average time consumed, from the time the plow was raised until it was set in the furrow ready to go, was 45 sec. for the tractor and 1.5 min. for the horse-drawn machine. This means something throughout a season's run. There is also a marked saving of time in starting to work. For instance, one day I held the watch on the tractor when the machine was working about 800 ft. from camp. From the time the foreman called "hitch up" until the first wagon was loaded just 10 min. elapsed; with the horse-drawn machine, under similar conditions but with about 100 ft. farther to go, 18 min. elapsed. This was a clear saving of 8 min. These items may sound small, but they are important when considered as a part of the daily saving over a period of 7 months.

Another item that should not be overlooked when comparing tractor and horse-drawn outfits is road finishing. With the tractor outfit, we made a practice of blading up the completed work after hours or occasionally on Sunday though, generally speaking, I do not approve of Sunday work. For this purpose I used a heavy grader having a 10-ft. blade. I have found from experience that a machine having a 10 to 12-ft. blade is the only one that will finish a roadbed in heavy soil properly, and that such a machine must be drawn by a tractor. As stated, this work can be done after hours where a tractor is employed as a part of the regular equipment. It was our custom to rough in from 0.5 to 1 mile and then go out for 2 or 3 hr. after supper and blade it down. By keeping up with the finishing in this way, the amount of sand-papering required at the completion of the job is minimized and the highway engineer and the traveling public are kept in good humor. With an all-horse outfit, the preliminary blading is done with a machine having a 7-ft. blade and drawn by six horses, but a light machine of this kind cannot do the work as well as the heavy machine. On the completion of a job by the horse-drawn outfit, it was always necessary to swing one of the tractors and a large blade grader over and do the final blading. This sometimes meant moving a tractor several miles, loss of time when it should have been working and loss of time in moving it back. Under present road specifications and system of inspection, a tractor for finishing is almost indispensable.

TABLE 1—DAILY OPERATING COST OF TRACTOR AND HORSE-DRAWN OUTFITS

Tractor Outfit

Operator, at $175 per month and board	$8.75
Gasoline, 28 gal. at 23 cents per gal.	6.45
Crankcase and Transmission Oil, 2 gal. at $1 per gal.	2.00
Track Oil, 3 gal. at 20 cents per gal.	0.60
Track-Roller Grease	0.20
Interest on Investment	3.00
Repairs for Season, including overhaul	4.00
	$25.00
Depreciation, subject to question	15.00
Total	$40.00

Horse-Drawn Outfit

Rental, 8 teams at $35 per month	$10.80
Hay and Feed	16.60
Machine Driver	6.00
Gig Driver	4.00
Push Driver	4.00
Extra Pay for Wheel-Stock Man	1.00
Harness Repairs	2.50
Horseshoeing	1.10
Veterinary Charges and Medicine	0.50
Total	$46.50

Operating Costs

The cost of operation, given in Table 1, is of interest. It affords an opportunity for a comparison between tractor and horse-drawn outfits and also for the comparison of actual costs with the data furnished by the builder. It shows a difference in favor of the tractor outfit of $6.50. It states only the cost of power, as the remainder is practically the same, and the costs of operation differ very little.

In making comparisons between tractor and horse-drawn outfits, the winter expense should be considered also. If a contractor owns his stock and is fortunate enough to get into a lumber camp for the winter, all well and good; but, if such work is not offered, as has been the case this past winter, and he is compelled to winter the stock himself, he piles up a bill of expense that would not be incurred with a motorized outfit. Another item is the smaller expense of a motorized outfit when tied up by rainy weather. Last fall we had bad weather that lasted 2 weeks and could work only 2 days out of 14. The expense of the tractor outfit during that time was only slightly more than one-half that of the horse-drawn outfit.

There is a wide difference among operators in the manner of handling a tractor, but there is only one right way. I had two operators early in the season. One came well recommended and apparently knew his business; he remained about 6 weeks. Afterward, I got a young man who was a real operator who reduced the daily gasoline consumption from 42 to 28 gal. This of course reduced the consumption of oil and also was much easier on the engine.

Daily Output and Repairs

There was, in my experience, very little difference between the daily output of the tractor and that of the horse-drawn outfit under similar conditions. Heretofore I have used the contractor size of elevator with a 36-in. carrier. Next season I am planning to use the heavier machine with the 40-in. carrier, as there is ample power to handle such machines. The best working gear for the 10-ton tractor is the low gear, but I have found this to be rather too slow a speed for ordinary work. In my opinion, 2 m.p.h. on low gear would be about right. This is about the speed at which a machine is drawn by 16 heavy horses.

When starting to use the track-laying tractor with the elevator grading machine, I feared the grader would not withstand the strain of continually being stopped and started. I have found this fear to be groundless. The tractor starts the grading machine more easily than horses do, there being no lunge forward but an easy pull that throws no great strain on any particular part. Fewer repairs were required on the machine operating behind the tractor than when it was drawn by horses.

I can say very little on the subject of repairs for the track-laying machine, as I have had this tractor but one season and the repairs have been comparatively light, consisting of small items such as valve-springs, gaskets, magneto parts and the like. Only one section of track broke during the season and no pins were replaced. No delays whatever were caused by breakdowns but this, of course, should be true of any high-grade machine for the first season.

Although I have been discussing tractors of the track-laying type entirely, this does not mean that I am altogether ignoring the wheeled tractor. I have one large wheeled tractor that has been in operation for seven seasons. Last season it traveled a distance of nearly 3000 miles on work, which consisted of turnpiking township roads, heavy maintenance of State roads and finishing new work. The grader used had a 12-ft. blade. This is good work for an old machine, but a comparison of costs between this tractor and the track-laying tractor cannot be made consistently because the old machine naturally used much more gasoline and oil than the new one. The wheeled tractor worked in long stretches, which called for very little turning. I gave up trying to use this tractor on wagon work in short cuts where frequent turning is necessary, because too much time was lost; but, on long, straight-away work, I have found it very satisfactory.

I have been particular to use the best oil lubricants, obtainable, regardless of price, and have adhered strictly to the instructions of the builder in this respect, although salesmen wished to sell me oils at a much lower price that were said to be just as good as those I was using. Other oils may be as good as those the builder recommends but it seemed unwarranted to change to some other oil for the sake of saving a few dollars. As a result of persistently using the oils indicated, I have had no bearing trouble. All parts of the machine are in excellent condition and the expense of overhauling after a full season's work will be very light. I have had the cylinders of the engine of the new tractor rebored and new pistons fitted. This would not have been necessary under ordinary conditions, but the excessive wear was due to the fact that we operated much of the time in fine sand, which worked in through the air-cleaner.

Earth Removal

The records show that the actual cost of moving earth with the tractor outfit on wagon work during the past season averaged about 1.5 cents per cu. yd. less than with the horse-drawn outfit. These figures are no doubt disappointing to those who expect the tractor to make a better comparative showing but the tractor outfit was operating largely on jobs where the machines could not work at their maximum efficiency at all times. Part of the time they worked in very loose dry sand, freely interspersed with boulders, while the horse-drawn outfit encountered very little rock during the entire season and was operating in soil that was ideal for an elevating grader. If the same outfits were placed on work where the soil conditions were the same, the actual cost of moving earth with a tractor outfit would be from 3 to 4 cents per cu. yd. less than with the horse-drawn machine.

Motorization of earth-loading machinery has been accomplished. The next step is to motorize the earth-hauling equipment in such a way that material can be moved more cheaply. Much time and money have been spent but no machine thus far designed for this purpose seems entirely practicable. Gravel hauling has been revolutionized by the motor truck; so, why not apply mechanical power to the moving of earth?

Many grading contractors are prejudiced in favor of horses for all purposes. After 2 years in business I decided that it might be well to adopt the plan of trying anything once. I am satisfied thus far; so much so that I shall add tractors to my outfits as fast as conditions will permit, being firmly convinced from experience that we are only beginning to realize the value of tractors on construction work.

I believe the contracting field offers the greatest opportunity for engineers in the history of the Country. Highway construction has only just begun and railroad work will be opening up again within a short time. The number of contracting firms in this line will be wholly inadequate. The engineer who enters the contracting field is not sacrificing his prestige; rather, he is adding to it. The day of the old-style, hard-boiled contractor is passing and the educated man is taking his place. There was a time when the man bidding for a job worked out his proposal by rule of thumb, and then wriggled and squirmed his way through the job, trusting that the "customary settlement" would bring him out all right in the end. Much litigation resulted. Now, a proposal must be made intelligently, with a real knowledge of costs, and the contractor is paid on the basis of work actually performed. The engineer knows how the work should be done and will do it, and he will do it in the most economical way, because the education of the engineer is always toward economy and efficiency. I do not mean to say that all engineers will make good contractors. There are many who are lacking in initiative or who have no confidence in their own ability; but the confident, practical engineer can put his talents to better use as a builder than in any other way. He can not only plan, but he can carry out such plans to completion and have the supreme satisfaction of knowing that the work is well done.

THE DISCUSSION

CHAIRMAN W. G. CLARK:—I am sure it has been of great interest to hear the practical side of road building from a man who has been through it. It is easy to theorize on how a thing should be done, but the man who has to go through it and do it is the man who really knows the thing from one end to the other.

L. C. HILL:—What grading is done to prepare the subsoil for carrying the loads that motor trucks impose on the roads? As I understand it, the roads under discussion are principally gravel roads.

R. C. SCHOEN:—The work is grading entirely. We have done gravel work, but my paper pertained almost entirely to grading and not to gravel work. We build right up from the virgin soil.

MR. HILL:—Is nothing done other than grading? Is nothing done, for instance, in the way of special drainage, laying of tile to drain the road, or laying of concrete?

MR. SCHOEN:—Wherever special work in the way of drainage is necessary, it is done; we have laid tile in many places designated by the highway engineer, or con-

structed special side-ditches, and the like, but that is all in connection with the road itself to prepare the roadbed for gravel.

O. W. YOUNG:—Where do the truck and dump-body equipment available today fit in with Mr. Schoen's ideas of efficient hauling? I believe he stated that this phase of the work has not yet been motorized satisfactorily.

MR. SCHOEN:—I spoke only of hauling earth. The motor truck is satisfactory for hauling gravel, but the motor truck with a dump body will not work in hauling earth because the earth is carried out over a loose dump down over the end where the wheels do not get traction, and a different type of vehicle must be used. On hard-surfaced roads the motor truck cannot be excelled, but the roads are not always hard-surfaced.

My attention was called recently to a new dump wagon. It is a 6-yd. all-steel wagon, weighing about 6500 lb. empty. It is designed to be hauled behind a small tractor, say about 15 hp. If such a wagon can be made practicable, I think it is what we need to motorize our hauling equipment.

I tried the Maney four-wheel 1-yd. scraper in 1919 in North Dakota. I used at first a 16-hp. steam engine for loading. The engine loaded the scraper all right but wrecked three or four men who happened to be on the machines. It requires a man to unload each machine. If you hit a rock it is liable to throw the man off on his head. This steam tractor was continually breaking down, and I abandoned it. Then I began loading with five heavy horses and found that was too expensive; I could not afford to have five horses loading and four horses plowing, so I went back to the elevator and wagons. I think that a machine like the Maney four-wheel scraper, built heavier and designed to be hauled behind a small tractor, would be a good machine for handling loose earth. I think the Maney four-wheel scraper is not practicable for hard work.

MR. YOUNG:—During the war, track-layers, I think they called them track-adapters, were applied to trucks. Although not used extensively they were operated successfully, at least experimentally. A track-laying unit was applied to the axle of the truck. Does Mr. Schoen consider that this traction equipment might fulfil the fundamental requirements of hauling?

MR. SCHOEN:—I think you are on the right line. What we want is some sort of machine that can easily approach the elevating grader to pick up its load. While it is under the elevator, traction for any kind of truck is adequate as a rule, but when away and going down on the loose, soft dump, the traction is not adequate.

MR. YOUNG:—Steering probably would be the problem then. On loose ground the front wheels of the four-wheel truck would be difficult to control.

MR. SCHOEN:—That would not bother much; it is the traction. A truck will sink into the loose earth when driven over it to dump a load over the end. A team would be necessary to pull the truck out.

MR. YOUNG:—Traction would overcome that.

MR. SCHOEN:—A track-adapter could be fitted to these trucks in a practical way, which would be all right.

A. W. SCARRATT:—A truck somewhat along this line, the Lombard truck-tractor, is on the market and is built, I believe, in Maine. It is a kind of cross between a truck and a tractor. Instead of having the usual round wheels at the rear end it has a pair of track-laying wheels. It was developed primarily for logging operations in the northern part of the Country and is not very different from the one Allis Chalmers worked on about 6 or 7 years ago. Another company at Eau Claire builds a logging engine that I think could also be used for this work. I think that the trade name of this machine is the Centipede.

C. O. WOLD:—There is another that I think would probably come closer to what you have in mind, because the Eau Claire and the Allis Chalmers machines, especially the latter, are very large. I think the Allis Chalmers weighs about 15,000 lb., perhaps more. There is one made in New York State called the Lynn tractor, of which a number have been sold in New York and also in northern Maine. It is often used as a tractor for small 8-ft. graders. There are about 50 of our machines working behind that truck now in road building.

MR. SCHOEN:—A thing that must be considered is the possibility of operating a dump-body.

MR. WOLD:—A truck equipment of that kind has to be made in such a way that it can come to the elevating grader from behind. The cab, as usually put on such a truck, would be in the way.

MR. SCHOEN:—Not at all.

A. W. BENSON:—My scheme of handling the dirt and the gravel is by a trailer to be hauled behind a tractor. The great difficulty of doing this is in controlling the loads on grades. We think we have solved this problem by adopting a rig by which we carry a large tonnage not on four wheels, but divided into small units. For this we have developed a two-wheel rig, that can be controlled by air. We have made experiments showing that with a 30-hp. tractor we can haul 16-yd. up an 8-per cent grade. That, I believe, is the record with such a load under control.

OBITUARY

EDMUND LEAVENWORTH FRENCH, manager of the Halcomb & Sanderson Bros. Works of the Crucible Steel Co. of America, died Aug. 31, 1922, at his summer home in Tully Lake Park, near Syracuse, N. Y., aged 51 years. He was born Oct. 12, 1870, at New York City. Following his preliminary education he attended Syracuse University and was graduated from that institution in 1902, receiving the degree of Bachelor of Science. He then spent 2 years in attendance at the Royal School of Mines, Freiberg, Germany, specializing in metallurgical engineering subjects.

Mr. French was successively chemist, metallurgist and manager of the Halcomb & Sanderson works subsequent to the completion of his technical education and, in addition, was in charge of sales of automobile steels for the Crucible Steel Co. of America during 1905, 1906 and 1907. He was elected to Member grade in the Society, July 23, 1910.

GAGE STEEL STEERING COMMITTEE

A MEETING of the Gage Steel Steering Committee has been held at the Bureau of Standards for the purpose of outlining in detail the first work to be undertaken. It was decided that a special research operator, proceeding under the supervision of the Committee, would be required for the successful prosecution of the work in view, and that the first series of experiments should largely form a study of the effects of the rate of heating on the dimensional changes in hardening and a comparison of the resistance to abrasion of the various steels previously selected by the Committee.

The Storage-Battery as a Mechanical Problem

By CLARENCE W. HAZELETT[1]

CLEVELAND SECTION PAPER *Illustrated with* PHOTOGRAPH

AFTER stating his belief that the storage-battery is a mechanical and not an electrochemical problem, the author reviews some of the storage-battery improvements of the last decade and comments upon the advantages of using thinner plates as a means of increasing their number in a given size of cell and so holding the active material in the positive plates. Plate and separator construction difficulties are enumerated and discussed in some detail, a photograph being shown of a battery having unusual characteristics that are described.

Questions relating to storage-battery capacity and length of life are considered, inclusive of effects due to temperature, initial amperage, discharge rate, amount of active material used, charging rates, internal battery resistance and voltage.

NO unusual improvement has been made in the lead storage-battery since the discovery of the pasted plate. The principal improvements have been mechanical in nature, and I believe that the storage-battery is a mechanical and not an electrochemical problem. Within the last decade we have had the iron-clad type of battery developed in which the mechanical construction is such as to hold the active material in the plates, and the alkaline cell in which materials are employed that will allow a better mechanical construction. Several times the theoretical amount of material actually used in its discharge is present in the plates of a storage-battery. This leads us to think that it is possible to make use of more of the material. It has been proved that the action on the storage-battery plate does not penetrate more than 1/32 in., even at commercial rates of discharge. To-day, plates ⅛ in. thick are practically standard. If it is possible mechanically to make thinner plates and take care of the other difficulties at the same time, it seems that some real improvement might be made.

The life of all storage-batteries is particularly dependent upon holding the material in the positive plates, in preventing mechanical breakdown, short-circuits, the cutting through of separators and other mechanical failures. A pronounced tendency toward thinner plates in storage-batteries has existed for the last 10 to 12 years. Plates ¼ in. thick were standard 15 years ago, but that thickness has been decreased gradually, 1/64 in. at a time, until both starting and vehicle batteries have been standardized, practically, and use plates ⅛ in. thick. The principal reason for this change has been the increase in capacity per dollar of cost, but that is not always the safest rule to follow. The capacity per pound of lead and active material also increases rapidly with a decreasing thickness. The tendency has been to use the same number of plates when their thickness was cut down, which did not decrease the capacity materially but shortened the life of the battery. If abused, the thin-plate battery will fail generally through buckling and short-circuiting in connection with standard construction, but for equal amounts of active material, the active material itself is longer-lived in a battery having a large number of thin plates.

If twice as many plates of one-half the thickness are used, the amount of surface will be doubled, the current density on charge and discharge will be reduced one-half and the heating effect will be decreased considerably; the rate of sulphation on discharge will be decreased and many advantages in charging and discharging will result from these characteristics. Heating is the principal cause of plate buckling and separator destruction; it will soften the active material in the positive plates and, if carried beyond 130 deg. fahr., will almost destroy the negative-plate capacity. So, the thin-plate active-material is longer-lived when used in the same quantity as in thick plates. The amount of gas given off per unit of surface is less, and the erosion of the material from the plates that is due to the gas bubbling up is very much lessened. The thin-plate battery has been tested by engineers of the largest companies and it has been found that it almost always wears out from buckling and short-circuiting, which are mechanical failures. On discharge, a thin-plate battery having a large number of plates does not heat up so much as one having thick plates. The reason is the lower internal resistance when the number of plates is increased, or, in other words, when the number of circuits in multiple is increased. The performance and life of a storage-battery should be improved by thinning down the plates, if this can be done from a mechanical standpoint.

When a battery gives the high capacity obtained by using the thinner plates, the battery is not charged as often in vehicle service. The principal damage to a storage-battery in service is done at the end of the charge when the gassing and heating occur. The plate material is worked more uniformly if a working thickness of the plate of 1/32 in. can be approximated. The unnecessary material in the ordinary storage-battery plate is used as a reserve while the surface material is worked off. In a new battery the superficial material is working and the interior is inactive when the plate becomes old, the outer material is in the bottom of the jar, while the cores of the plates are active. Several practical methods are used for holding the active material in the plates. It is customary to use a rubber separator in contact with the positive plates of vehicle batteries, which assists in holding the material in place. It has been shown that there is a large increase in the length of life by using the rubber separator but as soon as the surface of the plate has been worked off, the core washes out and does not produce any large increase in life. If we save the surface of the plates we save all. In the case of the iron-clad storage-battery, two to five times the ordinary life is obtained by holding the active material in the positive plates. That is the secret of obtaining long storage-battery life. The active

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material can be held in place, and the manufacture of thinner plates is purely a matter of manufacturing ability.

Plate and Separator Construction Difficulties

It is difficult to construct equipment with which plates 1/8 in. thick can be cast successfully. It is an heroic task to make equipment that can cast the 3/32-in. plates, which are used entirely by one manufacturer, without greatly increasing the percentage of antimony. Antimony is put in because it makes the metal more fluid and is necessary to give stiffness to the grid, which prevents it from growing. The handling of a large number of thin plates is expensive and it is difficult to paste thin plates in ordinary practice because this must be done from both sides. The plates become light, they will not lie on the table when being pasted and there are other very difficult complications, such as the holding of the pellets in a thin grid through the handling processes in the current method of manufacture.

The view that the need is for a large amount of surface, an increase in the number of plates, a decrease in their thickness, the holding of the active material in the plates and the development of the mechanical construction so that it will withstand the abuse to which batteries are subject, resulted in the battery shown in Fig. 1, which has some unusual characteristics. It required the development of a system of making cast lead sheets because rolling-mill lead will not work, and the production of a cast-lead antimony alloy sheet at a low cost. The sheet is approximately of No. 20 gage and is made in strips several thousand feet long. It comes from the machine at the rate of 70 ft. per min.; it is then fed into a punch-press and grids are stamped from it continuously. Practically no skilled labor is necessary and the material comes out as one would expect it to come from a machine.

It then becomes necessary to develop a separator to hold the active material in place. Wood veneer 1/16 in. thick was tried first. This worked very satisfactorily in an electric car and no difficulties developed in the batteries. However, it is difficult to treat a wood separator. Wood veneer 1/16 in. thick could be treated by any satisfactory factory process only with difficulty. Further, the veneer separator did not fill the space between the plates fully. The failures of others indicated that if an attempt were made to hold the material in the positive plates it must be held hight in the grid. If it is not held tightly, it will wash off, begin to sulphate and swell. To avoid this long-grained wood was rubbed up on an emery wheel; this split off the fibers, they were treated, and laminated sheets were made from them. This process resulted in a laminated separator 1/16 in. thick of pure wood fiber that can be treated more easily than veneer. It is made into sheets and is mechanically perfect. These separators swell so as to fill the space between the plates absolutely and hold the active material in the plates as is desired; they hold sufficient electrolyte and it is surprising what a large number of cubic inches of electrolyte they will absorb. Regarding porosity, if one of these separators be dried out and a drop of water placed on one side, the moisture will penetrate entirely through the separator in 5 or 6 sec. A battery is never discharged within 5 or 6 sec.; so circulation need cause no anxiety. Several mechanical advantages result from using such a separator. In ordinary storage-battery construction it is not possible to extend the separators down to the bridge and, if they are extended beyond that, they will split. This laminated separator material simply folds over, the separators are extended beyond the bottom of the plates, rest on the bottom of the jar and give added protection; further, wood fiber separators will swell and fill all the space allowed them. If allowed to do so, it is likely that they would expand another 1/32 in., but as that is unnecessary, it is preferred to have them more compact. The wood-fiber separator acts like a wick, draws the electrolyte up and prevents any damage to the plates; it will hold the electrolyte between the plates so that the excess can be poured from the battery, a discharge can be made and the rated capacity obtained at any rate of discharge desired. That feature is very desirable in such portable outfits as miners' lamps or in airplane work.

Observation of the oldest of these cells that have been made indicates that the active material does not and cannot escape from the plates. The insulation is dependable as it now stands because it extends beyond the plates. An important improvement has been made wherein the four edges of the separator are dipped in a hardening acid-proof mixture, which produces an indestructible frame about the entire separator, thus preventing any damage to them. The positive plates are pushed to one side, the negative plates to the other and the projection on the vertical edges is 1/4 in. or more. The fit of an element in the battery jar is very important. Jar breakage in motor vehicles occurs because the jar does not fit snugly in the tray, or the element does not fit snugly in the jar. If the element can move in the jar, a cracked jar is likely to result. These elements fit tightly, will hold their position and come out in good condition after indefinite use.

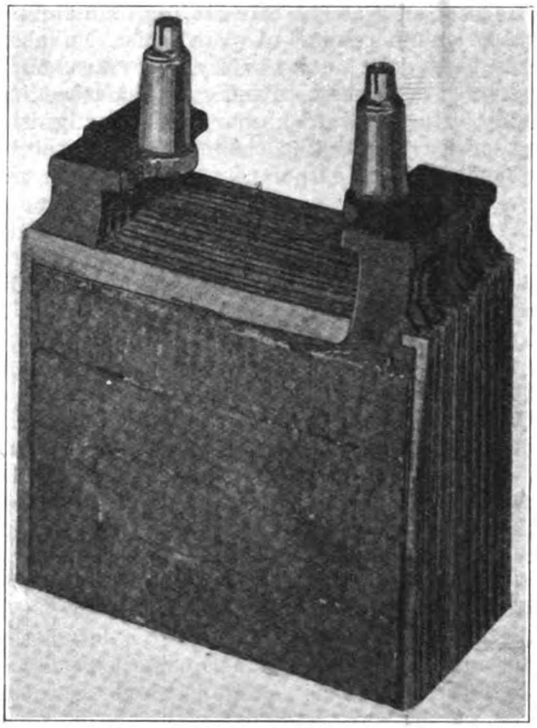

Fig. 1—Storage-Battery Element in Which Wood-Fiber Separators Are Employed between the Plates

This wood-fiber separator is not subject to mechanical chafing. In the usual construction, when an ordinary plate buckles the vibration and the consequent chafing action cause the plate having the pressure to wear through and produce a short-circuit. Practically all of the failures of batteries are caused by buckling and short-circuiting and by the erosion of the material in the plates. This battery uses approximately double the number of plates that are provided ordinarily in a storage-battery. An element consisting of 13 plates replaces the 7-plate standard. The 11-plate standard element is replaced by a 20-

plate element in this battery and, because there is approximately double the amount of surface, an unusual capacity per pound and per dollar is expected.

Storage-Battery Capacity and Life

Storage-battery capacity in ordinary automobile practice is important primarily at low temperatures and at the current value required for initial movement of the engine parts; that is, the break-away current. Initial available current is approximately proportionate to the plate surface exposed. With the type of battery under discussion this initial available current per pound of plate is 2 to 3 times that obtained from batteries having thicker plates, both cases being at +10 deg. fahr. The corresponding capacity at low rates is approximately 50 per cent greater on a per-pound-of-plate basis. No laboratory work has been done on low discharge rates, but other recent observations seem to indicate a marked increase in capacity at the low-rate discharge used for ignition and lighting service also; that is, the increased capacity seems to be something like 50 per cent per pound of plate at very low rates of discharge and at a 5, 6, 7, or 8-min. rate. At the 20-min. rate of discharge the capacity per pound of plate is increased approximately 100 per cent. Less active material and a decreased plate-weight are used in these batteries.

Ordinarily, if a battery plate has a flat side, when it begins to gas two volumes of hydrogen come from the negative plate to one volume of oxygen, and the flat side of the separator is against the negative plate. The gas escapes without difficulty, and it does so in a somewhat similar manner from the battery under discussion. The charging rates it requires while in a discharged condition allow the usual proportion of amperage but, as the battery becomes charged, this amperage should be cut down. The gas does not escape as rapidly, even though the battery has twice as many plates, and it will heat up about as is usual, but the total heating effect is less because of the lower internal resistance due to the large number of plates. On discharge, the voltage is higher at given current rates, particularly if the battery is shaken somewhat or allowed to stand after charge until the gas has escaped more completely. The internal resistance of the battery referred to seems to be 65 per cent of what is usual and that results in a higher efficiency. The separators do not clog or disintegrate. It was difficult to form plates with the separators in place between them, especially when using those made of veneer, without causing trouble at that time, but this difficulty was overcome. At any period in the battery's life, the separator does not clog and comes out as clean as when put in. It does not disintegrate, for the material has no place to be deposited. The separators are made out of the same material as is a wood separator, and there is no mechanical chafing action on them. In reference to whether the gas given off at the positive plate attacks the separators, no such effect on a rib of a wooden separator occurs, as is evidenced by a 2-year experience with the battery. Tests were made some time ago on a vehicle cell. After approximately three times the expected life, an examination of the cell revealed about 1/4 in. of sediment in the bottom. The capacity of discharge was still 12 per cent above ordinary and all the material was in the plates. The voltage on discharge was high. The material came out with the separators when the plates were opened, and that cell had been abused greatly. No difficulty has been experienced with mechanical disintegration of the fiber separators since the use of the frame of acid-proof material.

The ability of the separator to hold the electrolyte within itself results in another advantage. The elements are charged completely and shipped to the service-stations. All that is required then is to insert the elements in the jars, pour in the electrolyte, seal the battery and put it to work. The elements do not dry-out. The separators hold the electrolyte and the elements do not discharge any more than if in a battery jar. The element is wrapped in oiled paper. This allows the service-station to keep elements in stock. If a customer wants a rebuilt job and insists on having one cell repaired, only about 20 min. is required to repair a cell, and about 45 min. to rebuild an entire battery. That allows service-stations to give very rapid service to customers. A customer can come in with a battery and go out in the time required to put in a loan battery or take one out; or he can have a dead cell replaced within 20 min.

When an engine is started with a battery the voltage falls and causes ignition difficulties. The electric starter should give the engine a good kick, and it is necessary that the voltage stay up. This requirement is met by having a battery of this type with a large number of plates.

THE DISCUSSION

C. T. Klug: In making a quick change at a service-station, is the element put into the battery without having been recharged?

C. W. Hazelett: The element does not require charging unless it has been stored on the shelf. The action on the plates is exactly the same as if the element had been installed in a battery. The electrolyte is held within the separators, instead of being present between the plates.

W. R. Strickland: What space is necessary as a settling basin?

Mr. Hazelett: A 1/8-in. space is ample. A dusty appearance on the bottom of a jar is the only indication of sediment. The oldest successful battery of this kind was built in October, 1919, and was returned just recently; the battery case was completely gone and the bottom was out, but the amount of sediment was very slight.

Mr. Strickland: In what type of battery would a settling basin of 1/8 in. be needed?

Mr. Hazelett: In a vehicle battery.

Mr. Klug: With the separator pressed up so tight, how does the sediment descend?

Mr. Hazelett: The sediment does not go down; it is only lead that has been dissolved and that is reprecipitated. It is the same as the moss on the top of the negative plates.

Mr. Klug: What is the recuperative power of this battery?

Mr. Hazelett: The recuperative power is rapid, but very little current is delivered after the first discharge.

Mr. Klug: What is the effect on this battery of holding the starter of an automobile in service too long at a time?

Mr. Hazelett: The battery will run the starter longer than usual and no ill effects are produced.

Mr. Strickland: How many times can the battery be discharged and recharged?

Mr. Hazelett: A large number of times, I dislike to make any statement regarding the life of a battery because the length of life of any battery is an open question. Judging from experience and all available information, the indications are that at least double the ordinary life of a battery can be expected.

A Member: What is the efficiency of this battery?

Mr. Hazelett: About 88 per cent.

A Member: What are the possibilities of using a hard-rubber case for such a battery?

MR. HAZELETT: The hard-rubber case has been tried out; it was a failure on account of its high cost and because it was easily cracked after it had aged. When one crack developed, the case had to be thrown away. The demand for a good substantial storage-battery case is unlimited. Every firm in the country needs something better than the standard wooden box. Much experimental work is being done along that line.

A. M. DEAN: What is the maximum mileage obtainable on good roads from an electric-vehicle storage-battery that requires the same space as is usual, after having had a complete single charge; that is, what limitation is there on such a battery for use on an electric vehicle?

MR. HAZELETT: With some of the lighter electric vehicles 135 to 140 miles should be obtained when driving in the running notch. The type of car, the kind of tires, the weather, the kind of road and the driving notch used make a difference. In climbing hills there is almost no dropping off in the battery capacity because of the high rate of discharge. My own car pulls 100 amp. at times.

A MEMBER: The battery being so well filled up with the separators, how long a time is required, after starting, to charge it so that the electrolyte gives a proper specific-gravity reading?

MR. HAZELETT: In general, about 2 hr. after the voltage indicates full charge. When this kind of a battery discharges at a high rate, hardly any drop in the specific-gravity reading is noticeable. The specific-gravity reading does not follow the condition of the cell very rapidly. During charge, hardly any increase of specific gravity can be noticed until the cell begins to gas. The electrolyte is at a low specific gravity at the beginning of the charge, and it has a tendency to increase in specific gravity as it is charged. The electrolyte of high specific-gravity then settles downward in the cell of its own accord, and no increase of gravity occurs at the top of the cell until the electrolyte becomes thoroughly mixed on account of gassing.

A MEMBER: When a repair unit is shipped, is any allowance made for swelling of the separators after they are put into a cell? Is the element soaked to a maximum before being shipped?

MR. HAZELETT: The elements are delivered just as they are constructed in the factory; the separators are so uniform that the elements have swelled as much as they will swell and all that is required is to bring an element to its proper dimensions by squeezing it. The element is somewhat like a cushion; it does not break a jar when pushed down and yet it does not move with respect to the jar.

A MEMBER: Is there any less danger of breaking battery jars from freezing with this type?

MR. HAZELETT: I do not know; the expansion would be absorbed by the separator to some extent.

A MEMBER: To store this battery in cold weather, what should be done with it?

MR. HAZELETT: It should be stored in a warm place.

W. C. KEYS: What kind of wood is used in making these separators?

MR. HAZELETT: We prefer to use a long-grained wood, such as cedar or cypress. It withstands the action of sulphuric acid better. The process is somewhat along the lines of making paper.

MR. KLUG: Are these separators built up in layers?

MR. HAZELETT: Yes, they are laminated to increase the dependability by eliminating flaws. There is not so much lead in this battery as is ordinarily the case and a 6-volt battery is lighter than is usual by 11 lb. When the batteries are sent out the specific gravity is 1250 to 1260. That provides sufficient acid for an adequate capacity, and the cells will last longer if worked at a lower specific-gravity.

A MEMBER:—How is a battery plate made?

MR. HAZELETT: The grids are pasted, practically by hand, although it requires little hand labor and does not require a machine. One man can paste 10,000 plates per day. It is necessary to apply paste only on one side of the plate. The sheet from which the grids are punched is cast. Handling lead under pressure destroys its crystalline structure. Only a normally cast alloy will work satisfactorily.

H. G. WELFARE: A battery ordinarily has 11 plates and the battery described uses 20; why is one negative plate left off?

MR. HAZELETT: The quantity of plates that fits the standard-sized jar snugly is used and the number happens to be 20. The omission of 1 plate in 20 makes only a 5-per cent difference, assuming that this decreases the capacity of the battery an amount equal to the capacity of an entire plate, which of course is not true. The ordinary battery has a plate of standard thickness on the outside of each element and that plate works only one-half way through; so, there is one-half a plate on the outside of the two sides of each element that is not working, the equivalent of one entire plate per cell. That entails a loss of about 9 per cent. Since the plate in this battery works all the way through, that 9 per cent is not lost.

MR. KEYS:—If, as stated, this battery carries more electrolyte than is usual and has less clearance at the bottom, where does it carry the electrolyte?

MR. HAZELETT: Actually within the separators; in an 11-plate battery there would be 19 separators and the volume of acid is above the ordinary amount.

A MEMBER: If this battery were applied to a solid-tired motor-truck, would it be necesary to go to extreme trouble in mounting it, supporting it on springes, for example?

MR. HAZELETT: It is advisable to do that when mounting any battery, but this type is less affected by vibration.

C. H. PRATT: I understand you to say that the electrolyte will not freeze, but when this evaporates and the solution is replenished with water, what chance has the water to mix with the old electrolyte?

MR. HAZELETT: When water is put in, it will mix only as rapidly as will two fluids of unequal density, unless the battery is charged. The filling of the cells with water and putting the battery out in real cold weather is not recommended.

A MEMBER: Does the same amount of evaporation occur as is usual?

MR. HAZELETT: About one-third the usual amount of water is needed to compensate for evaporation. However, when any battery is charged, a given number of ampere-hours will decompose a given quantity of electrolyte, but as this battery develops less heat, the evaporation is not great.

A MEMBER: What is the rate of heating on this battery under high charging rates?

MR. HAZELETT: Before the gassing point is reached, the temperature does not increase appreciably. Constant-current charging of a set of battery cells at an ordinary voltage will not heat any battery to amount to anything. This battery absorbs the current satisfactorily without increasing the temperature above the limit and it absorbs current at a tremendous rate when the battery is first put on charge. After it is charged completely at the normal

charging rate, the battery will lag about 10 deg. behind usual temperatures.

A MEMBER: This battery having more plates, is it possible to use it for lighting-plant service?

MR. HAZELETT: Yes, it works well in lighting plants. It is not necessary to worry about charging rates or overcharging.

A MEMBER: What is the effect of over-discharging on the plates?

MR. HAZELETT:—If this battery is allowed to discharge for a long period of time and is charged, the charging voltage is high but the battery comes right up because the plate material is accessible to the electrolyte and the crystals formed are not large for the reason that the separators are in contact with the plates.

A MEMBER: Is not the active material broken down as a result of this practice?

MR. HAZELETT: No. The ampere-hour capacity can be decreased slightly by making the material less active, but the voltage will be adequate under those conditions and the battery will work satisfactorily and not decrease to an extent sufficient to prevent it from functioning in ordinary service.

A MEMBER: In charging, will the gas emitted from the separators produce a rising of the level of the electrolyte?

MR. HAZELETT: The rise in level is about the same as is ordinarily experienced. Gas dissolves in the electrolyte to some extent and thus increases the level of the electrolyte during charging.

A MEMBER: What is the 20-min. discharge rate on a battery of the size used on a Ford car?

MR. HAZELETT: The 20-min. discharge rate is 125 to 130 amp.

A MEMBER: What is the 5-sec. discharge rate?

MR. HAZELETT: It approaches the ratio of plate surface; the 5-sec. discharge rate would be 75 to 80 per cent above the usual result. The capacity does not increase in proportion to the plate surface on low rates of discharge; at high rates of discharge it becomes a plate-surface proposition. The plates in this battery can be buckled the same as all storage battery plates, but they can be straightened by pressing one's finger against them.

FUNDAMENTALS OF ENGINE DESIGN

(*Concluded from p.* 332)

gas velocities than is usually supposed, without any loss of volumetric efficiency, in which case turbulence looks after itself.

A MEMBER:—I believe that turbulence has nothing to do with it. If one blows smoke into a test-tube, puts a plug of cotton in the end and pushes it up, there is no mixture whatever. If the plug is pulled down turbulence is produced.

MR. POMEROY:—I think there is no doubt that gas velocity, as such, has a pronounced effect upon turbulence in spite of the experiment described. Sir Dugald Clerk made experiments many years ago by taking indicator-diagrams from a gas engine that had been allowed to rotate for a few revolutions with the ignition off and the inlet and exhaust-valves closed, the ignition and valves then being put into commission and a diagram taken of the consequent working cycle. The result showed that whereas maximum pressures of some 400 to 500 lb. per sq. in. would be produced in the ordinary way with a characteristic gas-engine diagram, when the explosion was delayed sufficiently to allow turbulence to die down, the explosion pressure was very low indeed while the resulting diagram had none of the characteristics previously obtained.

JOHN MCGEORGE:—I gather that the poppet-valve engines are most efficient when they are new, but that a slide-valve engine is easier to work with. Does that mean that in the beginning the poppet-valve engines are much more efficient and that their efficiency becomes reduced, while the slide-valve engines continue their efficiency unimpaired?

MR. POMEROY:—My experience is that from the viewpoints of economy and power a well made poppet-valve engine will maintain these qualities fully as well as engines of the slide-valve type. In fact, it seems difficult to believe that the same volumetric efficiency can be obtained with the slide-valve as with the poppet-valve engine, owing to the necessarily higher internal-temperature reducing the volumetric efficiency due to the heating of the incoming gas. Experiments on slide-valve engines show a much larger proportion of heat rejected to the exhaust than is the case with poppet-valve engines, thus clearly indicating the higher internal cylinder-temperature.

A MEMBER:—Is the fuel in Great Britain of a better grade than we have in the United States? I am told that American cars fitted with American carbureters operated better in Great Britain on a mixture of three-fourths gasoline and one-fourth kerosene.

MR. POMEROY:—The English gasoline is of better grade than the American. The poor quality of gasoline in America has done much to hasten the development of the gasoline engine.

A MEMBER:—What system of lubrication was used for the engine in which the babbitt bearings were installed?

MR. POMEROY:—Forced-feed lubrication with a pump of 7/8-in. bore and 3/4-in. stroke, running at camshaft speed, was used. The delivery of this pump at 1000 r.p.m. of the engine is about 1/2 gal. per min., the important thing being to supply plenty of oil rather than worry about the oil pressure itself, which in this case is about 15 lb. per sq. in.

MR. MCGEORGE:—I have heard the statement that it is better not to use any oil-grooves in forced-feed lubrication.

MR. POMEROY:—That is correct for forced-feed lubrication. The oil, being under pressure, will distribute itself without any grooves.

A MEMBER:—Do you endorse the floating piston-pin?

MR. POMEROY:—The floating piston-pin is advantageous because the wear is distributed over its entire surface and no restraint is imposed upon the expansion of the piston itself.

MR. MCGEORGE:—Do you favor the slitting of the skirt of the piston?

MR. POMEROY:—The piston having a slit skirt certainly allows the use of very small clearances and, given correct general design, it completely eliminates piston slap, the well known hindrance to aluminum-piston applications. It involves, of course, having the head partly separated from the skirt so that cooling takes place through the contact between the rings and the cylinder walls, which is, however, probably where the majority of heat escapes in any design of piston.

Automotive Equipment in Road Construction

By A. C. GODWARD[1] MINNEAPOLIS TRACTOR MEETING PAPER

THE plan adopted by some contractors of renting machinery to the project to assure a return on the equipment, regardless of the status of the rest of the work, involves the discussion of methods of determining depreciation, rental charges and the ratio of operating costs to fixed charges. Comparison is made of the author's system with that used by the Associated General Contractors. The costs of operation of a dump-truck, a caterpillar tractor and a steam-shovel were investigated and the results tabulated.

IN automotive equipment, the vital things are depreciation, maintenance and repairs. In the brief time allotted to me it will not be possible to cover this subject fully. I have chosen particularly the economic side of automotive equipment, to show the value of records of operation, maintenance and repair, to the owner, the designer and the manufacturer. Twenty years ago we thought only of mules and manpower. I remember the first tractor I had was a great disappointment. The tractor today is a joy forever. Through all these years as a personal hobby I have kept records of performance, cost and depreciation; they have shown how the designer has gradually eliminated the undesirable features and made automotive equipment an economic working unit. That being my hobby, it is naturally the line along which I wish to speak.

As engineers, we approve the statement of James J. Hill, the empire builder, that in modern civilization the engineer is the creator. It was not entirely the hope of pecuniary gain but the instinctive desire to create for the welfare of our fellow beings, that drew us into this our chosen profession. Among the members of this diversified calling the automotive engineer ranks high in ingenuity, tenacity and ability in the creative art. His work in a few years has changed the mode of transportation and the mode of living.

By revolutionizing the methods of transportation the automotive engineer has made the highways of yesterday useless, but in return he has given us equipment by the use of which we can economically construct the roads anew. We who during the greater part of these 20 years have devoted our energies to the construction of roads and highways have watched with interest his progress, we have sorrowed at his failures and rejoiced at his successes, and now, more appreciative than the rest of his friends, we bear glad and whole-hearted tribute.

In automotive equipment, the proof of design and mechanical construction is in the operation. Contractors as a group were as vitally interested in the development as were the designers and the builder, and being constantly at the operating end they were in a position to observe and record accurately the performance of the equipment and to give constructive criticism. A few with limited technical knowledge appreciated their position and responsibility and assisted the designer and the serviceman in eliminating faulty features. To them much credit is due. Experimentation, the creation of new alloys, the design of new bearings and appliances, the elimination of unnecessary parts and weight, standardization, better ignition, carburetion, transmission and balance and all the other improvements in equipment and accessories with which we are familiar today were continued until at last the many types approach perfection in design and application.

DEPRECIATION AND RENTAL SCHEDULE

Progress automatically eliminates the contractor who will not see. Today contractors manage their equipment investment in an efficient and economical manner and designers and builders can procure reliable and accurate statements of performance, detailed as to quality, quantity and cost. Competent public officials have learned to treat pieces of equipment as individuals and to study them as economic units. There is now something more than intangible kicks upon which to base study and design. The day is gone when equipment is bought as a one-job investment or at the whim of a public official regardless of economic principles. The trend of the times is shown by the development by the contractors' association of a schedule of depreciation and rental, and by the tendency of contractors to rent the equipment to their own projects at a predetermined rate per hour or day so as to assure a sufficient return of all costs regardless of the status of the project. Road construction is a continuing business and equipment must be designed and purchased accordingly.

Many of the municipal contractors today are following the plan mentioned; instead of letting their machinery go with the job if the job goes wrong, they let the job go where it will but assure a return on that piece of equipment sufficient to maintain, repair and put it into condition for the next job when they are fortunate enough to get one. The Associated General Contractors adopted a schedule in July 1920 predicated upon an expense divided into seven parts. This was made a basis of rental and is illustrated by a typical case shown in Table 1, on the assumption of a 6-year useful life.

TABLE 1—TYPICAL DEPRECIATION AND RENTAL SCHEDULE; LIFE, 6 YEARS

Items	Percentage of Original Cost
Average Annual Depreciation	12.5
Equivalent Annual Interest	4.0
Shop Repairs	6.0
Field Repairs	4.0
Storage and Incidentals	3.5
Insurance	1.0
Taxes	1.0
Total	32.0

Based on 8 months of operation this total of 32 per cent, as shown in Table 1, is a rental per month of 4 per cent of the original capital investment. The average depreciation is based on the economic life of a machine that is supposed to end when the value of the machine

[1] Chief engineer, Minneapolis Park Board, Minneapolis.

shall have depreciated to 25 per cent of the original cost. The annual depreciation is 75 per cent of the original cost divided by the number of years during which experience has shown it can be expected to give service. The equivalent annual interest is based on the prevailing rate each year on the depreciated value. The other charges are self-explanatory, a division being made between shop and field repairs for the purpose of computing different terms of rental. Nearly 100 items of equipment enter into consideration and the annual depreciation varies from 75.0 to 7.5 per cent, depending upon the type of machine and the class of work.

In municipal work we began the practice years ago of considering all equipment requiring an operator as an individual unit, renting it to the project on which it was being used at a fixed cost per hour and at a rate to assure a return sufficient to cover all fixed and operating costs and create a small reserve for emergencies. We based the rental charge on nine instead of seven component parts, five of which were fixed charges and four were operating charges. The fixed charges are (*a*) average annual depreciation, (*b*) equivalent annual interest, (*c*) storage and incidentals, (*d*) taxes and (*e*) insurance. Depreciation was included as a fixed charge regardless of hours of operation per year, as improved designs make obsolescence a deciding factor. The operating charges are (*a*) shop repairs and overhaul; (*b*) field repairs; (*c*) operating cost while employed; and (*d*) cost of maintenance, which consists of overtime work of adjusting, oiling, greasing and the like. The cost of time lost while equipment is on a project but not in condition to operate is obviously to be distributed among the other components, but a record is kept of all time lost and all nonproductive labor so as to complete the performance record of both the machine and the operator. Continuous records have been kept of the various types of equipment in our service, and the rental basis has been adjusted each year to meet the fluctuating costs of labor and supplies. Such records prove to be educational, promote economy in purchase, operation, maintenance and repair, and are of value to the company from which equipment is purchased.

Without going into details, illustrations of three typical accounts will be of interest. Actual costs as percentages of the original investment and based on 1600 operating hr. per year were as shown in Tables 2 and 3

TABLE 2—COMPARATIVE RECORDS OF EQUIPMENT OPERATION

Equipment	Truck		Tractor		Steam Shovel	
Items	Percentage of Initial Cost	Actual Annual Cost	Percentage of Initial Cost	Actual Annual Cost	Percentage of Initial Cost	Actual Annual Cost
Average Annual Depreciation..	19	$1,083	15	$900	11	$880
Equivalent Annual Interest..	4	228	4	240	4	320
Storage and Incidentals.....	3	171	3	180	3	240
Insurance......	1	57	1	60	1	80
Taxes.........	2	114	1	60	1	80
Total Fixed Charges......	29	$1,653	24	$1,440	20	$1,600
Shop Repairs...	8	$470	12	$720	11	$904
Field Repairs..	7	400	10	600	6	467
Operating Cost.	35	2,010	50	2,960	100	8,028
Maintenance...	2	114	4	264	4	320
Total Operating Charges......	52	$2,994	76	$4,544	121	$9,719
Total Annual Cost....... ..	81	$4,647	100	$5,984	141	$11,319

TABLE 3—COST PER MILE, PER HOUR OPERATED AND PER CUBIC YARD EXCAVATED

Items	Truck	Tractor	Steam Shovel
Fixed Cost:			
Per Mile Traveled[1]	$0.18	$0.80	$0.021
Per Hour Operated	1.03	0.90	1.000
Operating Cost:			
Per Mile Traveled[1]	0.33	2.43	0.126
Per Hour Operated	1.87	2.84	6.060
Total Cost:			
Per Mile Traveled[1]	0.51	3.23	0.147
Per Hour Operated	2.90	3.74	7.060

[1]For the steam shovel, this is per cubic yard excavated.

for a 5-ton dump-truck, a caterpillar tractor and a steam shovel. Being actual accounts of individual units these records are not intended as typical of the several classes of equipment, but are given only to illustrate the practical solution of the economic principles involved. The 5-ton dump-truck was purchased in 1920 and cost $5,700. Its estimated useful life is 4 years, and its total operation was about 3200 hr. when the truck was completely overhauled and retired, ready for the 1922 season. The caterpillar tractor was purchased in August 1919, and cost $6,000. Its estimated useful life is 5 years. It was used largely with elevating graders and grub plows. Its total travel was 4000 miles and the total time of operation 3400 hr. The tractor was put in complete repair for the 1922 season. The ¾-cu. yd. steam-shovel was purchased in August 1919 and cost $8,000. Its estimated useful life is 7 years. The total length of operation was 3921 hr. and the total amount of material handled was 190,700 cu. yd. This equipment was put in complete repair for the 1922 season.

The records of these particular units are given for three reasons: (*a*) the equipment built prior to 1919 was not up to the standard of the present day and comparisons would not be equitable; (*b*) each of these units has been operated continuously for 2 years, somewhat more than the estimated 1600 hr. per season, and has been completely overhauled and reequipped ready for the coming season's work so that all charges are included; (*c*) the truck represents the lowest ratio of operating cost to fixed charge, and the steam-shovel the greatest.

OPERATING-COST-FIXED-CHARGE RATIO

I do not know how many are accustomed to view the cost of mechanical service from this angle but I wish to call attention to what seems to me the outstanding feature of these records, namely, the ratio of operating costs to fixed charges; 2 to 1 for the truck, 3 to 1 for the tractor and 6 to 1 for the shovel. The ratio of practically all other equipment comes between these extremes. Initial cost of equipment enters into the cost of service only in the fixed charges and, of these, depreciation is in every case more than 50 per cent. The percentage of depreciation varies inversely as the quality of the machine and as the initial cost. Machines of good design and high standard depreciate slowly. The higher the grade of the equipment is, the lower the fixed charge will be, as the decrease in depreciation will more than offset the small increase in the other components. Good design and construction decrease every item of operating cost; reduce shop and field repairs, lessen the amount of maintenance, tend to a more economical use of fuel and oil and result in better morale among the operating force. Good design and construction reduce lost time, increase output and decrease unit costs of work. I know of no better argument for the elimination of cheap or even doubtful materials and designs, regardless of the increase in the initial

cost, than the fact that, based on the annual cost of productive performance, a high initial cost increases only from 7 to 10 per cent of the total and reduces all other factors.

Records such as these carried through the life of a machine, with all operating costs itemized in detail, should and will become standard practice among all operators who expect to continue in and profit by the business. They will be good economy, for they will prove to the owner that

(1) Equipment is only as good as its operator, and cheap manpower is a fallacy
(2) Proper maintenance after operating hours decreases all other operating expenses, increases performance and justifies as much overtime as may be necessary
(3) With funds to cover the cost set aside from earnings, shop repairs in winter are well done without stinting and lost time becomes negligible
(4) First cost is no criterion; as fixed charges, aside from depreciation, which will reduce as standardization is perfected, are not excessive
(5) The guide to purchase is operating cost and performance made possible by continuous efficient service

Such records will show the designer and builder the success or failure of the product, what improvements are essential or desirable, the changes that will adapt the equipment to new conditions that may be encountered, the necessity of more educational propaganda, the desirability of a better inspection of materials and especially of the assembly, and the economic necessity of building the best that their ingenuity can devise for the operator who is in business to stay and buys and handles his equipment accordingly. Records that will permit the analysis of the cost of operation, maintenance and repair, as related to the fixed charges, are the means by which those who operate can assist the automotive engineer and the builder in the final standardization and perfection of types that will bring success in construction work.

THE DISCUSSION

A. H. BATES:—Are the labor rates you have used in computing the shop repairs higher than those that would be paid by a commercial house?

A. C. GODWARD:—I believe they are. We operate practically at the union scale. We pay time and one-half for overtime and the union scale in Minneapolis is higher than that paid by the ordinary tractor or commercial house. Operators received from 70 to 90 cents per hr. during 1920 and 1921. The operator of a steam-shovel gets about $1.25 per hr.

MR. BATES:—I understand that some operators were allowed 25 per cent of their actual time for repairing and keeping the machines in condition.

MR. GODWARD:—That is not so in our department. We pay our men by the hour and expect them to operate 8 hr. per day, the same as ordinary construction forces. After the day is over we expect every operator to see that his machine has the proper maintenance. It takes from 30 min. to 1 hr. He is paid for the time he puts in. On Sunday we insist that he drain the crankcase and clean the machine. For that time he is paid or will be paid straight time; in 1920 he was paid straight time and in 1921 time and one-half.

This maintenance charge, when many records of automotive equipment are analyzed, amounts to a considerable sum in dollars and cents, but the cost per mile or per cubic yard or per ton hauled is very small. Proper maintenance eliminates lost time almost entirely. Lost time is intangible; the lost time of a truck or shovel or piece of equipment is small and does not mean much. But if it disorganizes a crew of 100 men and 50 teams for 1 hr., it means money. It is something you cannot put down in black and white; it is nothing to which one can take oath; but by the time a job is finished the lost time is appreciable.

COKER F. CLARKSON:—How much of value has Mr. Godward been able to get from the recommendations of various organizations, including motor-truck companies, as to methods of maintaining cost-systems of motor trucks? Also, to what extent does he feel that the system that he has had in effect could be a basis for a generally acceptable system of keeping costs? Men who are engaged in research work in which the keeping of such costs is involved are very anxious to get on some sort of standard basis on which they can figure and which will, if possible, be generally accepted. It has been suggested that some such system be devised and promulgated, but this does not seem to be an easy thing to do.

MR. GODWARD:—We do not claim originality in our methods of keeping the cost of equipment. We consult with the distributors, get letters from the builders, and in many instances are personally acquainted with the designers. We find that their recommendations are not always the same; but we try to use the best of their several recommendations for our particular use. Being a comparatively small organization I doubt if our cost-system would be applicable to a fleet of from 30 to 50 trucks or tractors. We have a number of different types with diversified equipment and adapt our accounting system to particular cases.

MR. SCARRATT:—The hours of service that Mr. Godward figures these different pieces of equipment can stand is rather interesting. I believe he said they expect to use the caterpillar 1600 hr. in a year and estimate its useful life at 5 years. That is a total of 8000 hr. of service and usually it is pretty hard service. If this is compared with average automobile usage, taking even 25 m.p.h. as the normal driving speed, which is rather easy service, it is equivalent to 200,000 miles of use. You cannot get that much out of many cars.

MR. GODWARD:—The reason that we get 8000 hr. of work out of a caterpillar, or expect to, is that it is given proper maintenance. The man who drives that caterpillar does not abuse it, he sticks to it, he thinks something of it. Those who drive automobiles and many who drive trucks do not own the vehicles and do not care about costs. They get out at 20 deg. below zero fahr. and try to start with the starter when all that would be necessary is a little cranking. They do not watch the road. One could name many instances where money is spent that could be saved. In handling a caterpillar, wheeled machine or steam-shovel, each operator knows that the length of his job depends upon the cost of operation, and governs himself accordingly. One cannot emphasize too much the statement that maintenance means life, lower depreciation and service. The reason we get these things from high-priced machines is that we get high-priced men to run them, men who care for them.

CHAIRMAN W. G. CLARK:—An automobile driven at 25 m.p.h. for 200,000 miles is not comparable in the amount of work that the engine is called upon to do with a tractor working for 8000 hr. The average American automobile is overpowered. Four, 6, 8 and 12-cylinder engines, developing from 100 to 120 hp., are used in cars up to 4000 and 5000 lb., when all that is needed is 60 hp. This is a thing that many of us are prone to forget.

Those who are used to driving a car that loafs practically all the time do not realize this as strongly as they would were they to get into work where they would have to handle heavy-duty engines that work hard all the time until something happens. It takes more time for maintenance on a heavy-duty machine than on one that is loafing most of the time.

MR. GODWARD:—It is unfortunate that the man who governs is supposed to furnish the brains and set the example. The passenger car has lost its usefulness after 3 years at the most. Depreciation, instead of being 15 per cent, is 25 per cent. Fixed charges and operating costs run about 50-50. The same is true of trucks. The depreciation is higher than it should be. We hope with the assistance of the builder and the designer, and with a little cooperation from the operator, that the depreciation of trucks will not be 25 per cent.

L. C. HILL:—The public must take better care of its automobiles if it wants to get the results that are shown in Mr. Godward's figures. There is no reason we cannot eventually bring the passenger car to the point of efficiency that it should attain to be an economic factor and a really reliable and inexpensive means of transportation. The only reason it is an expensive means of transportation at times is that the owner insists on making it so.

MR. GODWARD:—In regard to the 8000-hr. life and some of the other lives that might seem too long, the machines are rented at a fixed rate per hour. The revenue from this rental is sufficient to give proper field-repairs in addition to maintenance and proper shop-repairs in winter. We do not consider that the machine is junk from the fact that it is in poor condition; we figure that as operating expense and have money for replacement. By proper replacement and repairs the life of the machine is lengthened, but this cannot be financed by hauling material at 10 cents per ton-mile in a truck. There must be the proper revenue to give the machine proper care.

W. J. MCVICKER:—What kind of fuel is used in the equipment?

MR. GODWARD:—Kerosene and distillates are used in some rollers, but in the higher-priced equipment, the equipment from which the best service is obtained, gasoline and the best grade of oil on the market are practically always used. In winter, low-grade gasoline is never used. High-grade gasoline cuts down condensation and assures proper lubrication. "Economy" in the use of fuel or oil is never tried. This shows immediately in the unit cost of work done.

O. W. YOUNG:—Of what do the operations consist in the case of caterpillar tractors? Was the work all a maximum drawbar load?

MR. GODWARD:—No. In the first 2000 hr. it was practically all elevating-grader work, largely old stumpage, about as much "grief" as can be given to a machine; some sand and some clay work. The winter work with snow-plows is small in hours and light in duty, ordinarily. There has been very little haulage of materials with this particular machine. It has been used in pulling stumps and trees; very heavy work. The elevating-grader work consisted of about 90,000 cu. yd. of material loaded by elevator, and varied from 400 to a maximum of 1000 cu. yd. per day. The heaviest duty on the tractor is in elevating-grader work where a turn, usually a one-way turn, has to be made every 600 ft.

MR. SCARRATT:—Does Mr. Godward mean to indicate that track maintenance is the chief item of expense of the track-laying tractor?

MR. GODWARD:—So far as I recall, the tracks and rollers are the principal items of depreciation.

MR. YOUNG:—How long do you operate the track-laying machines? When did you get the first machine?

MR. GODWARD:—I did not expect that caterpillar to be taken as representative of the type. I used it as an illustration of what our cost record brought out in the relation of the operating costs to the fixed charges. This particular caterpillar may be a good one or it may be a bad one. It was built in 1919, was operated continuously in 1920 and 1921 and is still going. It was cited to emphasize the small importance of the initial cost relatively to the actual cost.

SULPHUR IN BENZOL

PREJUDICE against the use of benzol as a motor fuel is being aroused by complaints that certain benzol-gasoline blends are proving very destructive to automobile engines. This is peculiarly unfortunate because benzol is the one fuel that has been found to be in all ways suitable for supplementing the gasoline supply.

The trouble is caused by sulphur introduced as an impurity, presumably in the benzol used for blending. An average crude benzol will contain as much as 1 per cent each of carbon bisulphide and thiophene, both of which form highly corrosive sulphurous acid when burned in a combustible hydrocarbon-air mixture. During the process of refining crude benzol, a large proportion of the carbon-bisulphide content is removed by rejecting the first runnings in the distillation process, and all but a trace of the thiophene is taken out by a sulphuric-acid wash. Refining practice has developed these two sulphur-removing operations to such a degree that the occurrence on the market of a motor-fuel benzol with a sulphur-content above the permissible limit is not the fault of the present state of the art of refining. The difficulty must be regarded rather as a preventable accident. Either the original supplier of the high-sulphur benzol was unskilled in modern refinery methods, or substitution, conscious or unconscious, has taken place. The low-boiling volatile benzol fraction that accumulates in the distillation process and is cheap because it contains the larger part of the undesirable carbon bisulphide, may have been substituted for refined benzol.

The dealer can easily safeguard himself by adopting the precaution of testing the benzol he puts into his blends. The simple copper-strip test, as used for sulphur in gasoline, is directly applicable to the detection of carbon bisulphide in benzol, and a benzol blend giving a positive indication by this test should be rejected as unfit for use in a blended motor-fuel. The extremely sensitive color-tests shown by thiophene could undoubtedly be adopted for detecting the presence of any undesirable amounts of thiophene sulphur in such motor fuel as may be offered for sale.

A situation analogous to the present instance existed in the early days of gasoline refining, and only a few years ago, when casinghead gasoline was being introduced, gasoline containing sulphur was on the market. Under present conditions, however, the consumer is well protected against the presence of sulphur in the gasoline he buys. There is no technical reason why the same reputation for reliability should not apply to the benzol supply.

The Comparative Merits of Benzol and Gasoline as Engine Fuels

By W. O. HINCKLEY[1]

MID-WEST SECTION PAPER *Illustrated with* CHART

THE process of manufacturing benzol is described briefly and a specification for motor benzol is stated. Comparison is made with a specification for gasoline and distillation curves are shown. Comparisons are made between motor benzol and gasoline in regard to end-point, heat value and vapor-tension, together with miscellaneous data on pure benzol, motor benzol and gasoline.

A description of comparative engine tests made with motor benzol and with gasoline as fuel is included and the observed results are enumerated. The author states that the data relating to the fleet of vehicles he mentions show about 10 per cent in favor of motor benzol over the results obtained when operating on gasoline.

THE respective merits of gasoline and benzol as internal-combustion-engine fuels have caused many a discussion in the automotive world and, for this reason, I will give a short history of benzol that is taken from actual occurrences in Chicago. When benzol in its crude state was first introduced there, the large oil companies tried hard to annihilate it, because it is the only fuel that has the market name of being a competitor of gasoline. Due to this fact the oil companies came out with various derogatory phrases that, at the time, were justified in some degree and in some instances are logical to-day. A few years ago several companies were making benzol for the internal-combustion engine and were successful until the cold weather set in; then the benzol froze in the gasoline tanks on the street. Many incidents of this kind and troubles with sulphur and acids left a doubt in the public mind about the value of benzol as an engine fuel. Experiments continued and, due to research, benzol was introduced as a stable fuel on the market. Were it not for the limited supply, benzol would, in my estimation, replace gasoline as a motor fuel within a short time.

The question is often asked: What is benzol and where does it come from? It is derived from coal carbonized in a by-product or recovery oven. There are four principal resultant products; ammonia, gas, tar and coke. While small quantities of benzol are recoverable from the light oil of coal tar, almost all benzol recovered is from an oil obtained by scrubbing coke-oven gas.

Briefly, this coke-oven light-oil is obtained by passing oven-gas through "scrubbers" against a counter current of heavy petroleum. The light oil removed in this way is separated from the scrubbing oil by distillation, and the quantity of oil obtained varies with the nature of the coal coked, ranging from 1½ to 4 gal. per ton of coal. The light oil is then refined by distillation in fractionating column stills, sulphuric-acid washing and subsequent neutralization with caustic soda, thus producing certain water-white distillates which, when properly refined, respectively create pure benzol, toluol and xylol.

[1] Superintendent of transportation, Peoples Gas Light & Coke Co., Chicago.

TABLE 1—SPECIFICATION FOR WATER-WHITE BENZOL

	Temperature, Deg. Fahr.	Temperature, Deg. Cent.
Distillation:		
First Drop	175	79.4
60 Per Cent	212	100.0
90 Per Cent	250	121.0
Dry	275	135.0

[2] Residue, none. It must be neutral, containing no solids and practically no foreign bodies except slight quantities of sulphur present as CS_2, or thiophenes to total approximately 0.25 per cent. Specific gravity at 15.5 deg. cent. (60 deg. fahr.) 0.878 or 29 deg. Baumé.

Motor benzol therefore is a distillate conforming approximately to the specification given in Table 1.

Motor benzol will pass the doctor test and the corrosion and gumming test for gasoline. The doctor test is supposed to show sulphur, and the corrosion and gumming test to show residues. Comparative distillation curves are shown in Fig. 1.

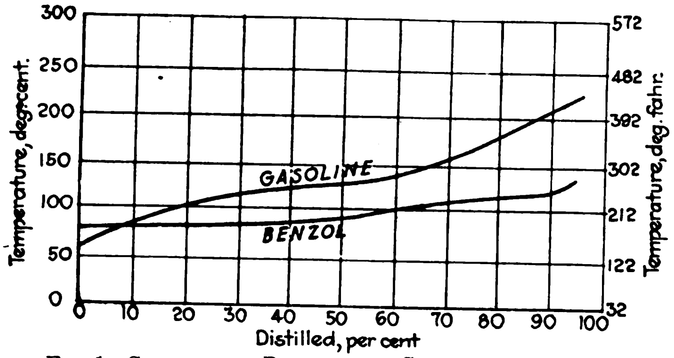

FIG. 1—COMPARATIVE DISTILLATION CURVES OF BENZOL AND GASOLINE

COMPARISON WITH GASOLINE SPECIFICATIONS

A comparison of motor benzol that meets the specification in Table 1 with commercial gasolines produced to-day is interesting. The products of refiners vary over a wide range. A few years ago the standard end-point for gasoline, the temperature at which it is completely vaporized, was from 330 to 350 deg. fahr.; to-day it varies from 400 to 500 deg. fahr. and in some instances it is as high as 600 deg. fahr. A distillation test of a high-grade sample of motor gasoline is given in Table 2, this

TABLE 2 — HIGH-GRADE MOTOR-GASOLINE DISTILLATION-TEST[3]

	Temperature, Deg. Fahr.	Temperature, Deg. Cent.
Distillation:		
First Drop	135	57
20 Per Cent	212	100
60 Per Cent	288	142
90 Per Cent	387	197
93½ Per Cent Dry	417	214

[3] Residue 0.2 per cent. Specific gravity at 15.5 deg. cent. (60 deg. fahr.) 0.732 or 62 deg. Baumé.

test having been made recently. However, we are not getting very much of this grade of fuel in Chicago.

While the initial point for the gasoline sample in Table 2 was 135 deg. fahr. as compared with 175 deg. fahr. for the motor benzol in Table 1, the gasoline shows only 20 per cent distilled off at 212 deg. fahr., the boiling point of water; whereas the motor benzol distills about 60 per cent at 212 deg. fahr. The end-point of the sample of gasoline was 417 deg. fahr. as compared with 275 deg. fahr. for the motor benzol. There was a residue of 0.2 per cent with gasoline, but the motor benzol shows no residue on distillation.

What the End-Point Indicates

The so-called end-point referred to is a very important indication of the efficiency of motor fuel. A high end-point indicates the presence of non-volatile constituents such as kerosene, which, if present in too great an amount, would reduce the efficiency of the motor fuel. Various methods are used in demonstrating this. A simple one is to compare the time of evaporation or volatilization of a fuel having a high end-point with one of a low end-point. This was done with the sample of gasoline referred to in comparison with a typical motor-benzol sample. This test was made by an arbitrary method; an equal amount, 5 cc., of each material was permitted to evaporate from an alberene stone dish, approximately of 4-in. internal diameter, under similar conditions. The evaporation time of benzol was about 26 min. and of the typical motor gasoline 145 min. High-test gasoline, such as is used in aviation, evaporates in about 11 min.

The heat value of motor benzol is about 18,700 B.t.u. per lb.; whereas gasoline has about 17,460 B.t.u. per lb. Benzol is considerably heavier than gasoline, weighing about 7.3 lb. per gal. as against 6.5 lb. per gal. for gasoline. The heat value for motor benzol is therefore about 137,000 B.t.u. per gal. compared with 113,500 B.t.u. per gal. for gasoline.

The vapor-tension of motor benzol is high compared with that of gasoline and, therefore, benzol is quick-firing in spite of its greater weight. The boiling-point of a liquid is an indication of its vapor-tension; for, as a general rule, as the boiling-point goes up, the vapor-tension comes down. Motor benzol has an average boiling-point of about 190 to 194 deg. fahr.; gasoline will run about 256 deg. fahr. or better.

Table 3 gives comparative data submitted by C. S. Heath, a chemist who has grown up with the development of benzol in this section of the Country, and is largely responsible for having solved the problem of preventing benzol from freezing.

During a recent test on a four-cylinder engine rated at 35 hp., we were successful in developing 36 hp. with motor benzol against 32¼ hp. with gasoline at 1900 r.p.m. A run of 6 hr. at approximately 1064 r.p.m. was made with both fuels. The following results were determined with the throttle wide-open. Before starting, the engine was dismantled and various parts were cleaned; new piston-rings were installed and the valves were reground.

During the run with straight gasoline the engine developed 24 b.hp. and consumed 19.1 gal. of fuel; or 2.94 gal. per hr. This is at the rate of 0.122 gal. per hp-hr. After this run was completed the engine was dismantled and what carbon had adhered to the top of the pistons was scraped off and weighed. On account of the shortness of the run there was only about 1.8 grams of carbon, some of which was due to the pressure of oil in the cylinder, caused, no doubt, by the installation of new piston-rings. The porcelain of the spark-plugs was slightly carbonized. The valves and valve-caps had a light carbon soot that could be scraped off readily.

For the run with motor benzol, we adopted the same procedure as when gasoline was used, reassembled the engine and put fresh oil into the crankcase. Then we adjusted the carbureter to a leaner mixture and retarded the ignition slightly. On this test the engine developed 26 b.hp. and consumed 13 gal. of fuel. The increase in power over that of the gasoline run was noticeable. During this operation the consumption was 2.89 gal. per hr. or 0.109 gal. per b.hp-hr. This shows about 10.7-per cent greater efficiency than the gasoline test. The engine ran more evenly throughout this portion of the test than in the previous run on gasoline, and did not appear to labor as much. An examination of the engine showed less carbon on the pistons; in fact, except for the small amount of carbon around the edges of two of the cylinders, there was not enough carbon to weigh. The centers of the pistons had practically no carbon, and the valves, valve-caps and spark-plugs were practically free of carbon. What little carbon we succeeded in locating was easily removed, due to the fact that it was not of the hard texture found when using gasoline as fuel. We also noted that the average temperature of the cooling water was not so high as when gasoline was used as fuel. With benzol we obtained a better running engine than when using gasoline.

In conclusion, I wish to state that the real test of any fuel is the result obtained in actual service. However, we have data showing that we are about 10 per cent better off in operating our fleet on motor benzol than we were when gasoline was used.

TABLE 3—COMPARATIVE DATA ON BENZOL AND GASOLINE

	Pure Benzol	Motor Benzol	Gasoline
Approximate Flame-Temperature, deg. fahr.			
With Air	3,650	....	3,500
With Oxygen	7,180	....	6,750
Heating Value, B.t.u. per gal.	137,000		113,500
Air Required for Combustion of 1 cu. ft. of Vapor, cu. ft.	36	39	45
Air Vaporized per Gallon, cu. ft.	39	35	25

THE DISCUSSION

H. W. Haines:—Is there any corrosion after some months due to the small sulphur-content? That is the principal point in the whole benzol situation.

W. O. Hinckley:—It is found with some grades of benzol but, when benzol is properly distilled, I think that the greater part of that difficulty is obviated. Other companies are not getting any of our benzol to-day because it is a question of our getting enough of it for our own use, due to the fact that we obtain only 4 gal. of benzol per 1000 cu. ft. of gas. One company in Chicago built up a reputation through the use of this fuel, because it was as nearly perfect a product as anything they found in research. One or two of the companies in Chicago are having a little trouble with corrosion.

Mr. Haines:—Do you contend that if benzol is properly refined the sulphur-content can be neglected?

Mr. Hinckley:—Absolutely. I will not say that I have no trouble. It does not happen often, but I have to caution our powerplant occasionally.

C. B. Page:—Mr. Hinckley has indicated that not very much benzol is available. Does this mean that benzol

will not cut much figure in reducing the price of gasoline?

MR. HINCKLEY:—Yes. Benzol sells to-day for 1½ to 2 cents more per gal. than gasoline, and it will continue to do so.

MR. PAGE:—Is there any prospect that the supply of benzol will become so great as to add materially to the supply of volatile fuels and thus assist in keeping down the price of gasoline?

MR. HINCKLEY:—No. The oil companies thought it would at one time but it will never come to that point unless gas is used universally as a fuel. That possibly would enable the gas companies to get enough production, but it would mean that residences would be heated by gas, steam boilers and office buildings would use it and, instead of getting fuel from various points, it would be taken from one center, practically speaking. The gas company hopes to be in a position some day to get the price of gas down where you will be heating your homes and business buildings with gas.

MR. PAGE:—My recollection is that, for general heating purposes, it was not contemplated to enrich that gas, but simply to pipe a gas having no illuminating qualities and a low heat-value through the streets.

MR. HINCKLEY:—The public-utility commission specifies a certain number of heat units for gas. I think the commission will never allow gas quality to get down to as low a point as 125 B.t.u. per cu. ft. of gas.

G. W. CRAVENS:—What is the relationship between the benzol Mr. Hinckley referred to and benzoline, which has been marketed as a commercial fuel for some time?

MR. HINCKLEY:—Benzoline is a motor benzol. Just what the mixture is, I am not in a position to say, but it is pure benzol derived from the steel coke-ovens, or the recovery by-product ovens of coke companies.

MR. CRAVENS:—In the winter of 1920-1921, when the question of corrosion was raised, I tried to find out about it. We ran about 3000 miles on benzoline and found less carbon in the engine and no corrosion whatever. We had three Ford cars running as light trucks and service cars. We had been told that the use of benzoline would ruin the engine.

Mr. Hinckley said that when operating on benzol the engine ran better than on gasoline. Did he mean by that less detonation?

MR. HINCKLEY: — I meant less vibration; in other words, the exertion did not seem to be there. We got better explosions on benzol.

MR. CRAVENS:—Do you mean you got less detonation and really better combustion?

MR. HINCKLEY: — Yes, better combustion all around. We are continually dismantling engines and they show very little carbon deposit; even on oil pumpers, the residue is nothing like it is when operating on gasoline. I have seen the carbon deposit on gasoline-operated engines loosened with a chisel but I never see one of my men do anything more than to use a screw-driver and push the carbon deposit off of the cylindler. If the engine is not an oil pumper, a rag will get all the carbon out. The only carbon that can be seen has the color of lampblack, with a little reddish hue to it.

CHAIRMAN B. S. PFEIFFER:—You spoke of a 50-50 benzol-gasoline mixture.

MR. HINCKLEY: — That is approximate and varies somewhat according to the weather. In cold weather very often 60 per cent gasoline and 40 per cent benzol is used; in hot weather, the proportions may be reversed, but those two points will cover any mixture that is used in the average motor-benzol. In hot weather we use as much benzol as we dare until we hit what we term the heating-point; that is, the danger point of heating up the engine; and we mix the fuel accordingly. With pure benzol, I can tie up almost any engine because of expansion, right on the street, if the weather is at all warm and the engine uses an ordinary water-cooling radiator.

CHAIRMAN PFEIFFER:—Have you ever made any experiments on increasing the amount of radiation?

MR. HINCKLEY:—No; but, judging from what we have done on Ford and Pierce-Arrow cars, nearly three times as much radiation is needed as the cars have at present, and very nearly that much extra water to keep the engine cool.

JOHN W. STACK:—With regard to detonation, there are some peculiar and I might say not thoroughly understood properties about those hydrocarbons that are different from gasoline. Possibly they operate differently in an engine. I have never tried benzol.

CHAIRMAN PFEIFFER:—I think that Mr. Stack's point about the differences between these hydrocarbons and gasoline is the whole thing.

MR. HINCKLEY:—I have known men who have changed from gasoline to benzol when they knew they were having trouble with carbon. When they put benzol in their engines they got just enough heat to cause that carbon to preignite the mixture and they had difficulty in reaching the nearest service-station. It is not unusual to hear that benzol will cause knocking and that gasoline will stop the knock, and you will hear the same thing about gasoline. The average benzol will eat the carbon out of any cylinder in 90 days or more. It will eat out all of that hard carbon unless the engine is an oil pumper, in which case the oil is causing the carbon.

A MEMBER:—In regard to whether benzol is a fast or a slow-burning fuel, it seems to me that it must be slow-burning because it heats the radiator more and carries the flame longer, more heat going into the water-jackets.

MR. HINCKLEY:—There is more heat to it, but it has a very short flame.

A MEMBER:—I cannot understand how it heats the radiator more.

MR. HINCKLEY:—I can explain that for pure benzol. If it has a short flame it has a terrific heat at the explosion point when properly mixed with air; whereas gasoline, which is less volatile, does not give so much heat.

A MEMBER:—But when you make a mixture that burns slowly and carries the flame down for a long time it always heats the radiator. That is the same condition.

MR. HINCKLEY:—If you carry that flame down in the open it might, but when confined in a cylinder it is so instantaneous that you do not get much effect from that heat in the lower part of the cylinder. If you did, you would have to give more clearance on the piston, right straight down. In other words, you would be getting the same temperature at the bottom of the cylinder because the flame is spent by the time the piston gets down there, and it is coming back so quickly that that space is not open to the firing point the same as the top of the cylinder.

M. J. HENDRY:—Thomas Midgley, Jr. deduced the fact that benzol is a slower-burning fuel, although it is volatile. Argument that the hydrocarbons in gasoline burn faster, offsets the argument about the carbon. If the engine of a car is loaded with carbon and benzol is put in there, that knock will be eliminated altogether, even though the carbon is in there.

MR. HINCKLEY:—That occurs sometimes.

MR. HENDRY:—In a large percentage of cases benzol will gradually take the carbon out of the cylinders, and

that offsets the other argument about carbon causing the knock. The carbon does not cause the knock; it is caused by the fuel. The hydrocarbons burn more slowly and that eliminates the knock, but they burn faster in gasoline and cause the same kind of detonation as with dynamite.

MR. HINCKLEY:—I disagree on that point. In a long flame the heat temperatures remain longer in the cylinder due to the fact that the cylinder has no chance to clean out and cool. In a short flame, the flame does not have the force behind it and the piston has a chance to clear the cylinder. The more quickly the flame clears from the cylinder, the less heat it will maintain. I defy anybody to show that gasoline has a shorter flame than benzol, irrespective of what make it is. Benzol is a quick-firing fuel.

A MEMBER:—The indicator-card should show that.

MR. HINCKLEY:—It does. Benzol shows a very short flash, also motor benzol mixed with gasoline.

L. E. GOIT:—An intensely luminous flame can be produced under some conditions that carries a very great percentage of heat which can be lost instantly in the jacket without delivering any sensible heat to the air or the other gases in the mixture; whereas a blue flame, or something approaching that, although there is seldom a real blue flame in the cylinder, has a tendency to turn the heat into sensible heat, and only what can be lost by contact with the cylinder is rejected to the jacket. I do not know how benzol acts in a cylinder or whether it has a blue flame. but there would be a chance for an explanation in that.

MR. HINCKLEY:—The benzol flame is almost blue.

MR. GOIT:—That would be against my explanation of the enormous heat-loss that pure benzol or perhaps a mixture of benzol and gasoline gives, because of having a luminous flame. C. A. French advanced a theory about some of the causes of detonation,[4] based on the relative sizes of molecules, the possibility of burning a molecule of the fuel at one flash or having it so big that the air could not possibly get into it to burn it at one stage, but there may be various waves of pressure and perhaps several detonations in the burning of a fuel that is as heavy as kerosene, for instance. Benzol is a comparatively light fuel. According to that theory, it would be possible to burn it at one flash without having the oxygen delayed in getting into a lower layer of the atoms in the molecule.

MR. HINCKLEY:—That is my argument; the longer the flame continues, the more heat you will get out of it. On the other hand, with almost any engine, retarding the ignition a little and making the fuel mixture leaner, will give from 18 to 24 per cent more brake horsepower than you can get out of the engine when using gasoline, so long as the engine lasts. With a proper amount of cooling water, that can be done continuously.

MR. HAINES:—Has the matter of corrosion due to benzol been investigated at average combustion-temperatures?

MR. HINCKLEY:—No. We did not have the necessary instruments. However, several benzol users who have experimented in that regard say that motor benzol does not create much more temperature at the explosion point than is the case with gasoline.

CHAIRMAN PFEIFFER:—It is difficult to put forward any theory as to why so-called motor-benzol acts as it does, because you have the effect of the benzol on the other half of the mixture.

MR. HINCKLEY:—The condition is not due to benzol alone; you are putting toluol in there and that helps a whole lot.

CHAIRMAN PFEIFFER:—It is as bad as trying to prove a theory in connection with so-called gasoline. You have a number of different fuels and each one acts on the other; it is difficult to determine the resultant action.

MR. HINCKLEY:—It is difficult to buy gasoline from any two stations in Chicago that will show the same analysis. There are various causes for discussion about benzol as a fuel. Few people are familiar enough with it even to-day to say whether benzol or gasoline is the better fuel. It is only a personal opinion as a general rule.

HENRY FARRINGTON:—Have you any comparative data on the mileage per gallon with Ford cars, using gasoline, this benzol mixture and the pure benzol?

MR. HINCKLEY:—We cannot give comparative costs on the same make of car. The cost on an 800-lb. Ford truck is about 10.5 cents per mile, including depreciation, insurance, overhead, tire mileage and everything else. The Ford 1-ton truck operates on an average of about 12.5 cents per mile.

MR. CRAVENS:—If you consider that commercial Benzoline is similar to your motor-benzol, I will give a comparison on two Ford cars each having a standard passenger-car chassis. One had a special body that made it weigh about the same as a Ford sedan and was used as a passenger car. The other had a light body and was used as a truck. They averaged about 12 miles per gal., in winter, on gasoline such as we bought in Indiana in 1921. This was on short hauls and through more or less snow. On Benzoline that cost 2 cents more per gal., we averaged between 18 and 19 miles per gal.

Referring to the injection of water, or to using steam in the intake-manifold, we made some tests on a Dodge touring-car with what was known as a steam carbureter. This is a device that takes water from the radiator and pumps it through a valve into the intake-manifold. When running along at 1200 r.p.m. without the steam injection, we found that by merely turning a valve and injecting a little vapor the engine speed immediately increased to 1800 r.p.m.

MR. HINCKLEY:—You would find the same effect if air were injected into the manifold. You can take air as it comes out from under the hood, cut an opening in the manifold and place a spring so that the vacuum will not open the spring until you reach a speed of 15 m.p.h. When the spring opens, the speed will jump to 20 m.p.h. without moving either the spark or the throttle. My objection is that the faster those engines are made to rotate, the faster the truck drivers will drive them, and vibration has a serious effect, especially on a 1-ton Ford truck. However, this does not increase the mileage, and we had no success with it.

MR. CRAVENS:—One injector device consists of a plunger with a dash-pot control to prevent it from operating too quickly. It leads into the intake-manifold for the purpose of letting air in, but it is controlled entirely by the throttle; that is, it will not come into action until the throttle has been opened to a certain point. We have found on testing Ford, Dodge, Nash and several other makes of car, that we increase the mileage from 30 to 70 per cent by the use of the valve, depending upon the make of the car.

[4] See THE JOURNAL, September, 1921, p. 182.

Some Aspects of Air-Cooled Cylinder Design and Development

By S. D. HERON[1] DAYTON SECTION PAPER

ON account of the exceptionally interesting character of this paper a departure was made from the usual practice of printing Section papers and the discussion following their presentation in the same issue of THE JOURNAL. To enable the members who were unable to be present at the March meeting of the Dayton Section to have the benefit of Mr. Heron's experience the paper was published in the April issue of THE JOURNAL. In accordance with the customary procedure the stenographic report of the discussion was sent to the various speakers for any corrections that they wished to make before the report was published and also to the author for him to reply. While Mr. Heron has replied to the discussion at the meeting, it is his expectation to incorporate the results of some later work in the discussion when it is printed in the TRANSACTIONS for the first half of the present year.

For the convenience of the members an abstract of the paper precedes the discussion.

ABSTRACT

THE paper reviews some of the salient points arising in the design and development of the modern high-output air-cooled cylinder. It is based to a very large extent upon the work of Dr. A. H. Gibson at the Royal Aircraft Establishment, which in turn was principally a development of the pioneer efforts of Renault, supplemented by some post-war work of the author for British companies and tests made by the engineering division of the Air Service. While the paper may, therefore, lack somewhat in originality, many of the results presented, it is stated, have not been published previously. The problems of an aircraft cylinder of approximately 40 b. hp. are dealt with primarily, but some aspects of automobile-engine cylinder design are considered.

The first point treated is the heat to be dissipated, this being followed by a consideration of how to secure an even temperature-distribution in the various parts of the cylinder. Cooling by a direct air-blast and by conduction is discussed, the importance of removing the heat from the cylinder at the point where it is given to the head, the ports and the barrel being particularly emphasized. The effects of mixture-strength and cooling air-supply upon the cylinder temperature are commented on, the text being supplemented by a number of tables. Methods of finning different forms of cylinder, the cooling surface required; the effect of the compression-ratio on the output, fuel-consumption and wall-temperature; cylinder materials; types of cylinder, with a summary of the advantages and disadvantages of the different forms of construction, valve-seat inserts in aluminum cylinder-heads; exhaust-valve cooling; and valve-gears, all receive attention.

The conclusions reached are that (*a*) successful air-cooling is not limited to 50 b. hp. per cylinder, (*b*) fragility of the fins is a disadvantage of air-cooling and (*c*) the compromises necessary in the design of air-cooled cylinders have been made at the expense of the cooling efficiency.

The effect of the position of the spark-plugs on the power output and the fuel-consumption is discussed in an appendix and the use of two spark-plugs located on a common horizontal axis that passes through the vertical axis of the combustion-chamber in such a position that neither plug can project a flame-wave against the exhaust-valves is commended. The influence of gas velocity through the valves on the performance of an air-cooled engine is the subject of a second appendix. In this, as throughout the paper proper, numerous illustrations and tabulations of test results supplement the text.

THE DISCUSSION

CHARLES L. LAWRANCE:—Having had some experience in preparing a paper of this sort, and seeing the tremendous detail and difficulties involved, I am awed at Mr. Heron's work on this paper. As it follows closely along the line of my own work, I cannot think of any criticism to make although in one or two instances I have had experience that differs somewhat from his.

One of the few things that I might mention is the question of a cylinder constructed with a shrunk-in liner, which is the type we now use. With this type of cylinder I find, contrary to European experience, that the performance after a number of hours of running, which is supposed to produce a high fuel-consumption, was, if anything, better after 70 hr. of running than at the start; probably due to decreased friction.

In some ways the cylinder with a shrunk-in liner has some advantages; it is easier to produce. In sizes such as we have here, it is very probable that it would be unsatisfactory, although I have had no experience with them and speak only from experience with small cylinders up to 30 hp.

J. H. HUNT:—One thing that impresses me on reading this paper is that it would have been a mighty good thing for the development of the automobile industry if every engineer had been compelled to work first with an air-cooled engine. There is considerable important information implied in the curves of Fig. 5, showing the variation of the head temperature with the quality of the mixture. The facts implied are just as true of the water-cooled as of the air-cooled engine, but the average engineer working on the former type has failed to get the point. So often he has set up his engine with his cooling system connected to the city water-supply and turned on enough water to maintain the temperature at the point that he finds gives the best performance without making sufficiently careful measurements of the distribution of the rejected heat, that he has failed to learn many things that he would have been compelled to learn if he had worked with air-cooled engines. I hope that those having the information as to engine conditions indicated in the curves of Fig. 5 will soon be in a position to give us their analysis. Mr. Heron states in his paper that in his opinion cast iron is the best all-around material for air-

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cooled cylinders for other than aviation purposes. I have not read Mr. Heron's paper as carefully as it deserves, but have not seen many data in support of this particular statement. If Mr. Heron can discuss this point more fully it would be greatly appreciated, as more data on this question are very desirable.

H. M. CRANE:—I feel that this paper is a challenge to every man who is designing a water-cooled engine; it shows him that he is many laps behind. The designer of an air-cooled engine has had to make an effort to make it run. This paper is of value to the designer of water-cooled cylinders for the reason that it contains a very thorough study of where the heat appears in the cylinder and where to get rid of it, which is as essential in water-cooling as it is in air-cooling. The cylinder developed in the past has been designed to act as we wanted it to; then we put a water-cooled jacket around it and hoped that the water would find the weak spot. I have had cylinders in the past that would pump 2 gal. of water out of the radiator when the engine was practically stone cold. It was just a case of the water refusing to do what I expected of it, which was natural because it was not treated right.

I am not ready to believe that the water-cooled engine is dead for aircraft work. One of the advantages of water-cooled engines is that we can give more consideration to the shape and arrangement of the cylinders and the general layout of the engine than is possible with the air-cooled types. Fundamentally the condition is the same in both. Heat is delivered to the cylinder and must be disposed of. In the case of water-cooled engines, we lead it away and then it is dissipated in the air; in the case of the air-cooled engine, it is dissipated to the air as close to the place where it originates as possible.

Another feature of air-cooled engines in aviation that is far from settled as yet is whether cooling directly in the slipstream is justifiable. I feel that it will not be on high-speed airplanes and, after looking over Mr. Heron's work, I am fairly well satisfied that it is not necessary to do it in the slipstream. In types of engine other than the radial, air-cooling may prove to be even more satisfactory. I certainly hope so because I think that the radial engine, due to its form and larger size, is a very unfortunate form for airplane use, as it is hard to push through the air and cuts off the pilot's view in a very important direction.

CAPT. LORENZO L. SNOW:—The air-cooled engine is somewhat out of my field, but I am in accord with Mr. Crane's statement that in the present status of development the water-cooled engine is much easier for us to handle and we gravitate to it on that account. The development of air-cooled engines is before us, particularly where cylinders of the larger sizes are involved, and this should be a challenge for us to keep at it. I suggest that a little more emphasis be placed on the actual ground covered and the success achieved to date with large air-cooled cylinders. We know that with small cylinders we are getting along very well; with larger ones the results are not so generally understood. The latter are being developed and I suggest that Mr. Heron tell us more of how they are actually performing in the air, and what his predictions are as to their filling military requirements.

V. E. CLARK:—The matter of air-resistance of exposed air-cooled cylinders as adversely affecting speed and general performance is important, and designers of radial air-cooled engines must go far to reduce this before the same speed is possible with air-cooled as with water-cooled engines. Suppose for example we have an airplane with which we are trying to get a speed of 150 m.p.h. at a great altitude. The specifications include the requirement that the engine cool while running on the ground without excessive fuel-consumption. It is assumed that the engine is mounted in the nose of the airplane in such a manner that the form of the nose of the fuselage is not susceptible of change. Now the air-speed past the cylinders at 150 m.p.h. is more than three times that of the propeller slipstream while standing on the ground. Therefore, the resistance at high speed is more than nine times as great. Hence, the horsepower actually consumed in cooling the engine at high speed is more than nine times that actually necessary. As against this, refinements in radiator design and installation for water-cooled engines, such as retractable radiators, go far to obviate this condition. The comparison is even worse at great altitude, because then the air is much colder and it may be that, in such conditions, as much as 15 times the horsepower is used in cooling radial air-cooled cylinders as is actually necessary. In an airplane otherwise well cleaned-up and streamlined, the percentage of available horsepower that is lost in this manner may become very great.

E. H. DIX, JR.:—Mr. Heron's paper gives us a clue to the reason for his success with air-cooled engines; namely, his wide and very practical acquaintance with all phases of his subject. Of course I appreciate his consideration of the foundry more than anything else. Most of the early failures of air-cooled cylinders, either by blowing off the head or the opening up of cracks, could be traced back to stresses set up during casting. The foundry was called upon to carry too much of the designer's burden.

Our first experience with air-cooled cylinder-heads came in connection with a cast-on head with cast-in valve-seats and spark-plug bushings. It was a four-valve flat-head design and, to make matters worse, we had an all-wood pattern that had to be sent back to the pattern-shop to have the fins glued up every time we put it in the sand. So when Mr. Heron brought us the first of his patterns, the three-port Type I design with very slick aluminum fins, we had reason to know that he had not been as mean as he might have been. Accordingly we told him that there was nothing to it and got busy. We found we had to experiment with sand mixtures to obtain a good slick surface on the fins without getting too dense a mold and so cause blows. We lost two castings before we determined the lowest pouring-temperature at which we could completely run the fins and a third when the mold washed where it was not properly secured, but then the next two were satisfactory and we have every reason to believe that we could make a very fair production record on this particular cylinder.

The next cylinder that we worked on was the two-port Type J design, which caused some trouble in the molding due to one place that was hard to lift, but the first two castings that we poured were satisfactory. The four-port Type H design caused us more trouble in the molding than either of the other two, but again we saved the first two cylinders that we poured.

I have mentioned our experience with the casting of these cylinders because undoubtedly the foundry problem is one of the most difficult to be met by the designer of the air-cooled cylinder. It is chiefly a molding difficulty; for, once the mechanical difficulties of preparing the mold are overcome, the metal problems are slight. We have had little occasion to worry over shrinks, draws, porous metal or cracks in the three designs mentioned.

The contrast between molding with the first all-wood

pattern and the later metal-fin patterns gives an indication of the improvement in ease of molding that may be made possible by refinements in patterns and perhaps the use of jolt ramming-machines. The foundry must be educated to take its proper share of the burden in the development of air-cooled cylinders. It is not sufficient to design a cylinder amenable to good molding, but we must go farther and develop the method of molding so that we *know* it is a good production job.

C. F. TAYLOR:—The water-cooled engine designers might have learned much if they had started in on air-cooling. This is particularly well illustrated by some of the exhaust-valve temperatures observed in the laboratory of the Engineering Division. On one of Mr. Heron's cylinders running at present, the exhaust-valve appears to run much cooler than any exhaust-valve in a water-cooled cylinder of a similar size. There are many theories as to the cause of this, but none has been proved definitely. Given a valve of the same size and cylinders of approximately the same size, the air-cooled exhaust-valve will run very much darker in color than the water-cooled exhaust-valve.

With regard to the necessity of air-cooling under certain conditions, we look forward to aviation that will encircle the globe, and we think possibly that one of the great fields for aviation will be over territory where there are no other quick forms of transportation. I refer to such regions as Alaska and others having very cold climates where transportation facilities are needed but at present are not available, and where the airplane may become one of the most important methods of travel. Temperatures in Alaska run as low as 40 deg. below zero fahr. in winter. Automobile drivers can understand what that means for the water-cooled engine. I do not believe that there is any satisfactory anti-freezing solution that could be used at 40 deg. below zero. This means that if aviation spreads out into such territory as that, we must use air-cooling.

The foregoing remarks apply also to very hot climates that make water-cooling very difficult, because there is a high limit of the temperature at which the water-cooled engine can be run. It is practicable to run air-cooled engines at cylinder temperatures considerably higher than is possible with water-cooled engines. In some of the Mesopotamian campaigns it was said that the desert was strewn with the remains of water-cooled engines in which the radiation surface had not proved large enough to withstand the heat. I believe I am right in saying that the British used air-cooled engines almost exclusively in this region during the war.

LIEUT. ERIK NELSON:—Shortly after my trip to Alaska, I visited my own country, Sweden, where they were doing considerable flying. Only air-cooled engines were used there. They were carrying mail every day 50 or 60 miles farther north than we were. In Alaska we got as far as Nome, 60 miles south of the Arctic Circle. They were flying that winter in Sweden 30 miles north of the Arctic Circle, and only air-cooled engines could be used. They were of British design. The temperatures run from 35 to 50 deg. below zero cent. (31 to 58 deg. below zero fahr.). The people are anxious to get mail service in Alaska and, as the weather there in winter is very cold, satisfactory air-cooled engines are absolutely necessary. Heated hangars and numerous other auxiliaries are necessary to handle the engines satisfactorily.

GLENN D. ANGLE:—Will Mr. Heron explain why aluminum construction is preferable to the use of metals showing high conductivity?

CHAIRMAN G. E. A. HALLETT:—While flying at Petrograd, Russia, in the fall of 1912, the cold weather did not bother the engine but it did cripple our airplane. We used a Curtiss pusher-type hydroplane and, in running on the water preparing to take-off, the spray froze on the tail and jammed our controls and we found ourselves on the water with no means of controlling the airplane. The weather was not cold enough to give us trouble with the engine.

LIEUT-COM. KARL F. SMITH:—A rather paradoxical fact has been developed in connection with the cooling of aeronautic engines which is that by coating the cylinders with certain enamels that are excellent heat-insulators, the effect is to increase rather than diminish the radiation. The reason for this is not fully understood, but much research on this particular phase has been done by Doctor Gibson and his co-workers in England. The failure of certain Hispano-Suiza engines in use was attributed by the manufacturers to the fact that the enameling was imperfect or broken off. This enameling seems to work equally well with water-cooled cylinders. The enamels employed are usually those having a Kauri Copal base, the mastics being less efficient. Black seems to be the best color to use, irrespective of theoretical considerations. The increased efficiency of heat dissipation in using cast-iron cylinders is about 2 per cent but varies with the cylinder material and increases to 15 per cent with aluminum. This means that the enameling of cylinders will reduce the temperature about 30 deg. fahr.

It strikes me that for purely military reasons air-cooled engines must be used in preference to water-cooled engines under certain conditions. In my estimation, the air-cooled aeronautic and automotive engine must be used for cross-country work over arid territory, in extremely cold climates and at high altitudes. According to a recent report of the Spanish fighting in Africa they are transporting the wounded over a stretch of 1800 km. (1118 miles) of hostile desert country, and in this work air-cooled engines are used almost exclusively due to the difficulty in obtaining water. I spent part of two winters in Siberia where, at times, the cold is intense. I found it necessary to make long trips by automobile. The low temperature around seaports did not bother us, as we had heated garages, but in the interior we had our troubles. After the introduction of air-cooled engines this difficulty was over. Even with air-cooled cars, trouble was experienced in starting in the morning, but starting on ether was usually successful.

For combat airplanes of the future and altitude work in general, I believe the air-cooled engine to be absolutely essential. The temperatures at an altitude of 35,000 to 40,000 ft. are extremely low. In descending from a great altitude the engine is ordinarily cut-out for a great portion of the time and the descent made by a series of glides. If anything goes wrong with the engine, unless all of the water in the cooling system is got rid of immediately, the cylinders will be cracked and the radiator injured before reaching ground.

CHAIRMAN HALLETT:—On all our supercharged airplanes we have what we call a dump-valve that makes it possible to release all of the water from the cooling system quickly. In case of engine trouble the pilot can dump the water out of the cooling system to prevent destruction by freezing.

S. D. HERON:—In reply to Mr. Lawrance it is admitted that the shrunk-in liner construction is much more satisfactory in sizes below 100-cu. in. capacity than in larger sizes. The British had, however, considerable trouble with this construction even in 67-cu. in. capacity cylin-

ders, difficulties with the liners burning away at the top, leakage of oil between the liner and the jacket and fractures of the jacket due to localized stresses at the holding-down-bolt bosses were experienced. Such failures were almost entirely a dynamometer condition produced by extended periods of full-throttle running at full speed, a test erring perhaps on the side of seventy, but which was considered necessary by the Royal Aircraft Establishment for the production of a reliable air-cooled cylinder.

Possibly Mr. Lawrance would find that some of the British experiences would be duplicated with his construction if submitted to 100 hr. of full throttle on a single-cylinder engine. The percentage of cylinders developing liner trouble was small, this practically never developing in flight. The lack of uniformity was, however, considered sufficient to condemn the construction.

I know of one R.A.E. 4E-cylinder that successfully withstood 350 hr. of full-throttle running. This goes to confirm the excellent results obtained by Mr. Lawrance with his shrunk-in liner construction. In spite of this, however, my experience even with cylinders of below 100-cu. in. capacity is that the thermal contact and thus the performance of the shrunk-in liner construction is a much more variable quantity than that of the screwed-in type.

Answering Mr. Hunt, it was not intended in the paper to indicate that cast iron is the best material for automobile-engine cylinder-construction and any lack of clarity causing the paper to be so read is regretted. However, in my opinion, cast iron is the most commercial material for car engine-cylinders, although I would not use it if cost was no object and cooling efficiency and high output were the main considerations.

The conclusions regarding the cooling efficiency of cast-iron cylinders are based upon the cast-iron R. A. E. 4D-cylinder and the B. S. A. car engine. The difference between the aluminum and the cast-iron R. A. E. 4D-cylinders is not excessive, although this cylinder size is beyond that for which cast iron is recommended. The figures given for the B. S. A. car engine are, I think, the best evidence in favor of cast iron for the air-cooled automobile-engine cylinder. That this car is able to cool efficiently without a fan, in spite of a fairly high weight-power ratio, speaks volumes for the possibilities of such cylinders. Mr. Hunt can readily confirm my remarks on the cooling efficiency of this engine by reference to reports of its performance in recent British competitions.

It is hoped that it will be possible shortly to furnish domestic data on the relative performance of cast-iron and aluminum air-cooled cylinders. The Air Service has under construction a 4½ x 5½-in. design on which comparative tests will be run with cylinders having cast-iron and aluminum heads, the construction being otherwise identical.

Replying to Mr. Crane, I would say that those who claim that the water-cooled engine will be superseded for aircraft use by the air-cooled type are rather wild optimists. Doubtless improvement in either type will but spur the other type to further improvement, paralleling the ceaseless competition that has gone on for years between the naval gun and armor plate. In my opinion, both types have advantages for particular applications. The air-cooled type has the advantage of a greater effective working temperature-range. That is of importance in extremes of climate. Furthermore, the air-cooled engine is not so vulnerable to gun-fire, since leaks in the ducts carrying the cooling medium do not cause almost immediate failure as in the case of a modern water-cooled engine. Water-cooling does not restrict the engine layout; whereas, air-cooling in the present state of knowledge does so very seriously. It is possible at present to air-cool successfully only a few types of layout; namely, radial, Vee, X and possibly line types.

Experience with large air-cooled radial engines in flight is as yet very limited, and it appears to be somewhat premature to draw conclusions regarding the relative head-resistance of this type with water-cooled engines of more conventional layout. Recent certified tests of European pursuit-planes, in which speed has by no means been the main consideration, have tended to shake the belief which was previously held in this Country to the effect that the large air-cooled radial engine causes excessive head-resistance.

In answer to Captain Snow, practically no experience in flight with the larger sizes of air-cooled cylinder has yet been obtained in this Country. The satisfactory experience of the Engineering Division with large cylinders on the dynamometer is somewhat limited and almost entirely confined to the type-J cylinder. In this latter design, 200 hr. of full-throttle testing has been obtained and considerable trouble with exhaust-valve breakage has been experienced, this being due in the main to the shock resulting from the somewhat crude exposed valve-operating gear, rather than from excessive temperature. A 50-hr. full-throttle run at 1650 r.p.m. has, however, been obtained on one exhaust-valve. During this run the only enforced stoppages were due to a broken push-rod and fouled spark-plugs. At the conclusion of this test the exhaust-valve, which was untouched throughout, showed some scaling on the stem and the neck and slight wear on the tip, the seat not having scaled in the least and being only very slightly pitted.

Preliminary tests on the type-I cylinder have demonstrated its inferiority in comparison with that typified by the J cylinder. The former type clearly demonstrates the disadvantages of the flat-head construction for air-cooled cylinders. This cylinder has shown markedly lower output, higher wall-temperatures and fuel-consumption than the type-J cylinder.

Flight tests of large American air cooled radial-engines should provide data on the relative head-resistance of this type in the near future. Should experience show that the radial engine is an undesirable type, satisfactory air-cooled V-type engines can certainly be produced. Allied experience on the Western and other fronts demonstrated that the air-cooled V-type engine is a satisfactory and reliable type, in spite of the fact that types used were fitted with what is now known to be very poor cylinder construction.

Mr. Clark also raises the question of the relative head-resistance of the air-cooled radial-engine and considers this to be excessive. In view of the paucity of evidence on this subject, I should be interested to have details of any comparative tests of air-cooled radial engines with conventional water-cooled types known to Mr. Clark. It is rather unfair to condemn the air-cooled radial engine on the score of head-resistance when so little real experience exists with this type and airplane designers have lacked an opportunity to develop efficient methods of fairing such engines.

The question of cooling on the ground with an air-cooled engine is really comparable to the similar problem of the water-cooled aircraft-engine; few of the latter will stand wide-open throttle on the ground for more than a few minutes without boiling. Past experience with air-cooled engines has shown that, with a cooling system designed to cool efficiently in the air, it is possible to run the engine full-open on the ground for about 2 min. and

then take-off without overheating. No useful purpose is served by running an engine full-open on the ground for a long period; opening full out for a minute or so to determine whether the engine will run up to speed is really all that is required.

The successful production of the later designs of the Engineering Division's air-cooled cylinder has been very largely due to the able assistance of Mr. Dix and his foundry staff. I wish to second Mr. Dix's remarks regarding the importance of the foundry in connection with air-cooled cylinders. If commercial success is to be obtained, ease of foundry production is essential.

In reply to Mr. Angle, I would say cast aluminum is at present the most practical material for aircraft cylinder-head construction as, for equal weight, greater total heat-conduction capacity in the walls and fins can be obtained than with any other present practical material. If materials other than aluminum be used for the cylinder-heads, the resulting design, if of equal weight, will be seriously lacking in stiffness to resist distortion due to the explosion pressure and temperature variations. The cast-aluminum head allows the production of a light engine, in which the port and combustion-chamber design is not cramped by considerations of ability to machine such shapes. Such considerations have to be given close thought in the built-up welded-steel construction used for water-cooled aircraft-engine cylinders, often to the detriment of the design. It is noticeable that water-cooled aircraft-cylinder design in this Country is steadily gravitating toward the aluminum head on account of the ease of production, apart from thermal considerations and questions of reliability.

For equal strength, some of the high-tensile aluminum casting-alloys are on a weight basis only inferior to forged alloy-steel, the disadvantages of which material for air-cooled cylinders are discussed in the paper. Preliminary investigations by the Engineering Division of the Y alloy have shown a tensile-strength at 600 deg. fahr. of 29,000 lb. per sq. in. in heat-treated sand-castings. The suitability of the alloy for cylinder construction has yet to be tested in actual service, but it has proved easy to cast into cylinder-heads.

In conclusion, I desire to amplify parts of the paper slightly, to avoid the possibility of misleading other workers on the subject. From the later experience of the Engineering Division it appears that the suitability of the copper-silicon alloys for cylinder-heads is open to question. One of the cylinders of this alloy recently tested developed stretching at the point of juncture with the steel barrel, apparently pointing to a very low elastic-limit at high temperature.

In the use of shrunk-in aluminum-bronze valve-seat inserts caution should be exercised in the amount of shrink used. As high shrinking-allowances set up excessive stresses in the cylinder-head, it seems that a suitable shrinking-allowance is 0.002 in. per in. of seat-insert outside-diameter.

Mention of the length of thread to be used in severed-in constructions was omitted in the paper. A thread length of not over one-quarter of the bore has proved satisfactory. Increasing the length of thread beyond this is liable to be productive of trouble, as the greater the length of the thread is, the greater will be the disparity of pitch between the steel barrel and the aluminum when heated-up, thus resulting in a poor thermal contact. Aluminum R. A. E. 4D cylinders with the liners threaded over the whole range of contact with the barrel gave exceedingly poor results, due to an almost entire lack of effective contact with the jacket and excessive oil-leakage and carbonizing.

It can be proved readily that, if the length of the United States Standard form thread be made 0.577 of the effective diameter, the head cannot be shrunk onto the liner or the barrel, because the effect of any increase in the pitch diameter due to heating of the head is counteracted by axial-pitch disparity.

MEMBERSHIP

To the Members of the Society of Automotive Engineers:

Each and every one of us appreciates the unusual conditions that have prevailed for some time, especially in the automotive industry, which affected the employment situation and naturally the opportunities of the Society of Automotive Engineers for getting new members, as well as to hold those members already within the fold.

You may be interested to know that in spite of many resignations, in spite of general conditions working against us for some time, the total membership in 1922 is somewhat in excess of the enrollment at the end of 1921.

A campaign to interest those who should in our judgment become affiliated is now in progress, with the idea of getting the names of good prospects among the production men, draftsmen, body engineers, etc.

The Membership Committee appreciates fully the hearty cooperation extended in our various efforts to interest additional worthy timber, but we would like at this time to emphasize the importance of continuing this good work so that we can make an exceptionally good showing for 1922.

The qualifications for full membership in the Society are specifically outlined in the application forms, and the Grading Committee and the Council which takes the final action desire as well as attempt to classify properly the applications received. We believe that those directly in charge of this work are taking a broader view regarding the qualifications for and the grades of membership. In addition, re-classification has had its effect, and with the untold advantages that accrue from an affiliation, combined with the rather nominal expense of being a member, it seems extremely possible that we, before the end of this year, should considerably increase our membership in all of the different grades.

It is only natural that applicants should desire full membership grade if at all obtainable, even though as associates they would secure all privileges except those of holding office and voting. Our desire is to classify the applicants so as to satisfy them, at the same time safeguarding the interest of what, in reality, is a strictly engineering society.

So, in soliciting applications we want you to feel, as we actually feel, that honest consideration is given each applicant's qualifications, as outlined in the applications. The references cited must of necessity be heard from before action can be taken. The recommendations of those given as references have due weight with the Grading Committee as well as the Council.

Let us all get intensely busy on this membership campaign and see if what we know to be the benefits of belonging cannot be passed on to many hundreds who we think should and could belong, putting up a figure covering a total membership of worthwhile chaps for the succeeding Membership Committee to shoot at for some time.

MEMBERSHIP COMMITTEE,
LON R. SMITH, Chairman.

New Principles in Rotative Balance

By Amos F. Moyer[1]

Minneapolis Section Paper *Illustrated with* Diagrams and Photographs

With the advent of high-speed machinery, the ordinary methods of rotative balancing were found insufficient. Rotative bodies that had been given the most careful static or standing balance would vibrate seriously when running, because of an unbalanced couple consisting of two quantities of equal magnitude on opposite sides of, but displaced from each other along, the axis. Thus, a second consideration in rotative balance, called dynamic or running balance, was introduced.

Until the invention of the balancing machine that the author describes, there were no means for measuring directly the resultants for the two separate ends of a rotative body, but now these quantities can be measured directly and located without the necessity of separating the standing from the running balance. The new principles are demonstrated mathematically with the aid of diagrams, the balancing machine is illustrated and described, and comments are made to afford a correct conception of the subject of balance.

The need of balancing rotative parts probably has been recognized to some extent since the invention of the first crude machines in which man caused a wheel to revolve on a shaft. If the wheel was not symmetrical in all respects and one side was heavier than the other, the labor required to revolve the wheel was increased and the first lesson in rotative balancing was forced upon primitive engineers. To overcome this difficulty, the equalization of weight as evidenced by the action of gravity upon the wheel in various angular positions was found to suffice and, with man's increase in mechanical skill, refinements of this process were developed and employed with a fair degree of success so long as the rotative body was not too large in dimensions or revolved at too high a speed.

With the advent of high-speed machinery, such as electric machines, centrifugal pumps, cream separators, steam turbines and automotive engines, these methods were found insufficient. Repeatedly, rotative bodies that had been given the most careful static or standing balance would vibrate badly when running. The only possible condition capable of producing vibration of a revolving body that is in static or standing balance is the state of an unbalanced couple, consisting of two quantities of equal magnitude on opposite sides of, but displaced from each other along, the axis. A familiar example of this is a two-throw crankshaft with the crankpins 180 deg. apart. Thus, a second consideration in rotative balance sometimes narrowly called dynamic or running balance was introduced.

It has been the almost universal practice in producing rotative or complete dynamic including static balance to obtain first a state of static or standing balance and then to determine the remaining couple of so-called dynamic or running-balance correction. This couple must be applied as two equal quantities in two artbitrarily selected planes of revolution; usually consisting of the two ends of the body. Obviously, the initial or static correction for standing balance could be applied in either or both of the same planes of revolution. In either one of these planes, there would then exist two corrections. A single correction in each plane of the proper resultant magnitude and position would necessarily produce the same result.

New Principles

Until the invention of Dr. B. L. Newkirk, of the research department of the General Electric Co., there was no means available for measuring directly the resultants referred to for the two separate ends of a rotative body. Now, these quantities can be measured directly and located, without the necessity of separating the standing from the running balance. That one and only one correction is required in each of two arbitrarily selected planes of revolution, dependent only upon the state of unbalance of the body and the position of the selected planes, is proved by the following analysis:

Since elements of unbalance are of the dimension weight times the radius, which is proportionate to centrifugal force at a given speed of rotation, we can, for purposes of analysis, speak of any such element as unit weight existing at a geometric radius of the proper magnitude.

In Fig. 1, let the plane Xx_0Y represent any longitudinal plane of the body containing the axis of rotation XX. Any point such as A may represent the position of unit unbalanced weight at the radial distance x_1A from the axis of rotation. Assume two longitudinal positions such as YY and y_1y_1 for the determination of two quantities x_0D and x_2C, resulting from x_1A. By moments, we can write the equations

$$x_2C = (x_1A \cdot YA) \div Yy_1 \quad (1)$$
$$x_0D = (x_1A \cdot y_1A) \div Yy_1 \quad (2)$$
$$x_2C + x_0D = x_1A \quad (3)$$

Equations (1) to (3) show that by the application of x_0D_1 and x_2C_1, the equals and opposites of x_0D and x_2C, the system will be in complete equilibrium both statically and centrifugally.

If the position YY is taken as a fixed center or axis of moments and the position y_1y_1 considered variable, the resulting quantity on y_1y_1 will be the intercept on the hyperbolic curve AC which has the equation $x\ y = k$; only one such curve containing the point A. Likewise, when y_1y_1 is fixed, there can be but one similar curve DA which determines the position of D on YY, so that the corrections required, x_0D_1 and x_2C_1, depend only on the magnitude of x_1A, the existing unbalance, and on the location of YY and y_1y_1, the assumed positions of correction. By summing algebraically similar values for any number of elements of unbalance existing in the plane

[1] M.S.A.E.—Chief engineer, Precision Balancing Machine Co. Minneapolis, and consulting research engineer.

XY, it will be seen that two corrections of the proper magnitude at assumed positions will always suffice.

It can be proved that after the application of one of the required corrections such as x_2C_1, which balances x_1A about *YY*, a second center of moments can be chosen at any point not contained in *YY*, and the remaining correction required on *YY* will still be the equal and opposite of x_0D as evaluated previously. Also, it can be proved that this value and location of correction is the only means of balancing about the second center of moments which will establish complete equilibrium.

If, after the application of the quantity x_2C_1, which balances x_1A about *YY*, a center of moments such as y_1y_2 is chosen not containing the point *C*, the resultant R_y on *YY* will be:

$$R_y = [x_1A \cdot (y_1A + y_1y_2)] - [x_2C \cdot y_1y_2]_1 \div (YA + y_2A + y_1y_2) \qquad (4)$$

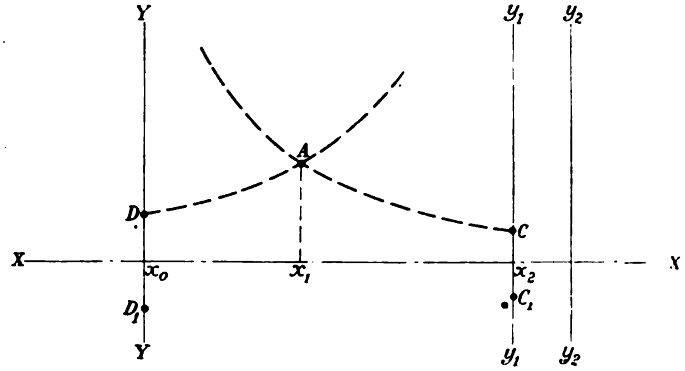


FIG. 1—DIAGRAM SHOWING HOW THE APPLICATION OF PAIRS OF EQUAL AND OPPOSITE QUANTITIES WILL CAUSE A SYSTEM TO BE IN COMPLETE EQUILIBRIUM, BOTH STATICALLY AND CENTRIFUGALLY

Where

$$x_1A = [x_0D \cdot (YA + y_1A)] \div y_1A$$
$$= [x_2C \cdot (YA + y_1A)] \div YA$$

Whence

$$x_2C = (x_0D \cdot YA) \div y_1A$$

Substituting in equation (4) the values of x_1A and x_2C in terms of x_0D, then expanding and collecting terms, we have

$$R_y = x_0D \qquad (5)$$

Equation (5) comes under the general theorem in mechanics that, after a solution by moments for one resultant force of a system, there will remain but one other force to establish equilibrium, and that force acts through the center of moments.

This shows that the resultant on *YY* after the application of x_2C is the same, irrespective of the location of the second center of moments. Here again the question holds only when the second correction is applied on the line of the first center of moments.

Fig. 2 is a three-dimensional representation of a body to be balanced, *XX* representing the axis of rotation. Let a center of moments be taken in any plane of revolution *oo*. In the plane of revolution *aa*, *xA* represents an element of unbalance and, likewise, in the plane of revolution *bb*, x_1B is an element of unbalance which may be of a different magnitude and in a different angular position about *XX*. The curves *AC* and *BD* are hyperbolas of the variety shown on Fig. 1, lying in the respective planes *XXA* and *XXB* and commonly asymtotic to the axis *XX* and the plane of revolution *oo* containing the center of moments.

In any desired plane of revolution *cc*, not coinciding with *oo*, the quantity x_2C has a moment about *oo* equivalent to *xA*. Likewise, x_2D is an equivalent of x_1B. The

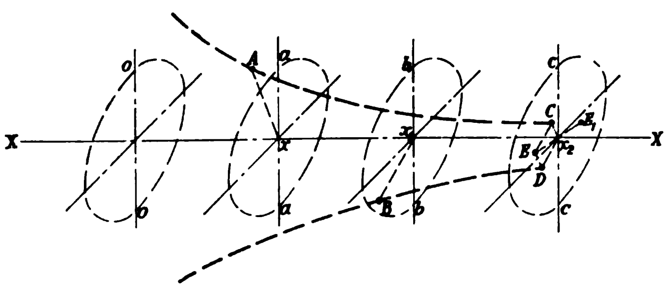


FIG. 2—THREE-DIMENSION DIAGRAMMATIC REPRESENTATION OF A BODY TO BE BALANCED

resultant in *cc* of x_2C and x_2D will be x_2E and both *xA* and x_1B can be balanced about *oo* by applying x_2E_1 equal and opposite to x_2E.

By a similar construction, taking a second center of moments in the plane *cc*, there can be evaluated the resultant in *oo*, which will be of a different magnitude and angular position from x_2E. It is also true that, after the application of x_2E_1, there will be only one resultant of unbalance lying in the plane *oo*, irrespective of the location of the second center of moments, and the application of the proper correction in *oo* will be the only means of obtaining complete equilibrium.

When *oo* is fixed with reference to *aa* and *bb*, changing the position of *cc* effects a variation in the magnitude of x_2E according to the same hyperbolic equation, but does not alter the angular position in the plane *cc*. The magnitude of the correction required in *oo*, and usually also its angular position, will be dependent upon the position chosen for *cc*. Likewise, with *cc* fixed, a change in the position of *oo* will affect the magnitude, and usually the angular position of x_2E_1 also.

As definitely as a multiplicity of quantities may exist in the plane Xx_0Y of Fig. 1, and give but two resultants at the respective centers of moments, each being the algebraic sum for all quantities, the three-dimensional body of Fig. 2 requires but two corrections, each balancing the geometric resultant of all quantities referred to common centers of moments.

Actually, the centers of moments may take the form of pivoted supports, and the measurement and location of the resultants about these supports will complete the determination of corrections required to place the body in absolute rotative or complete dynamic balance, which is

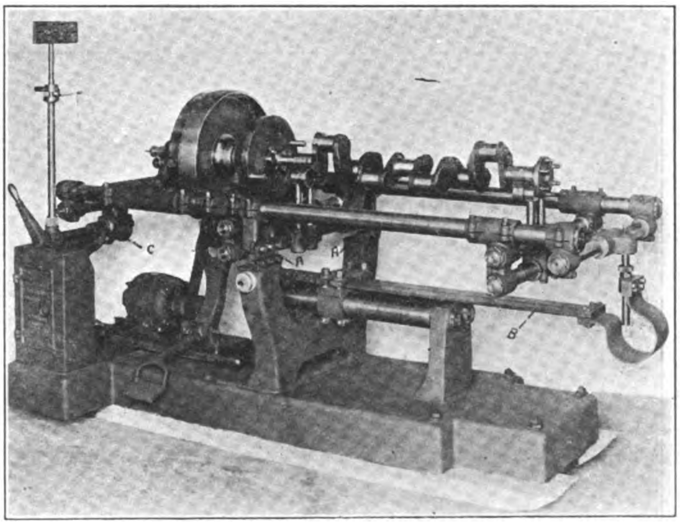


FIG. 3—A RECENTLY DEVELOPED MACHINE FOR DETERMINING THE BALANCE OF A ROTATING BODY

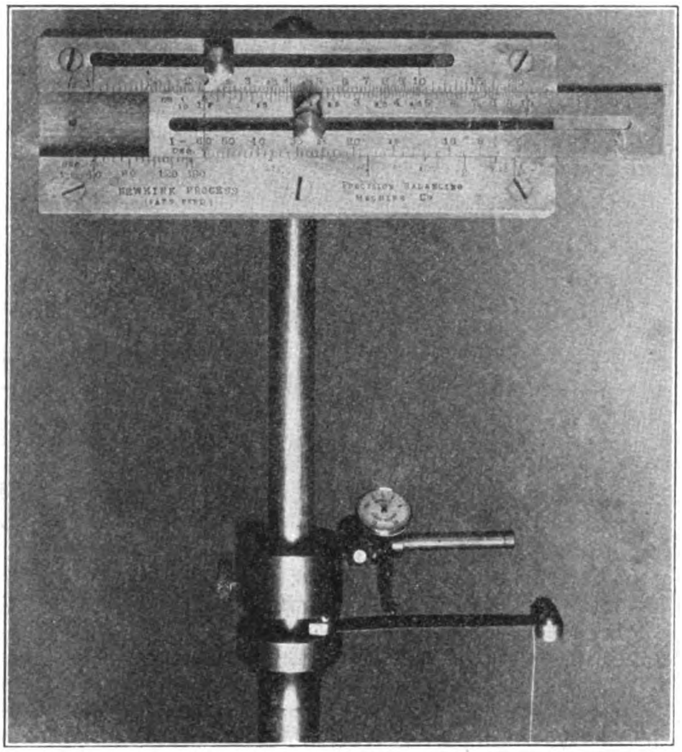


FIG. 4—APPARATUS FOR DETERMINING THE AMPLITUDE OF OSCILLATION

the state of complete equilibrium with respect to centrifugal force.

THE BALANCING MACHINE

Fig. 3 illustrates the precision balancing machine, the joint design of F. McDonough and myself, constructed under the Newkirk inventions. A frame is incorporated pivoted at *AA*, which supports the work in the desired positions to bring the selected planes in coincidence with the pivots or centers of moments. In this position the correction required in the overhanging end of the work is determined while revolving, and the ends of the work are then interchanged for determining the correction required in the plane initially over the pivots. These two corrections, at their respective angular positions, will place the body in complete dynamic or rotative balance without separately determining static balance.

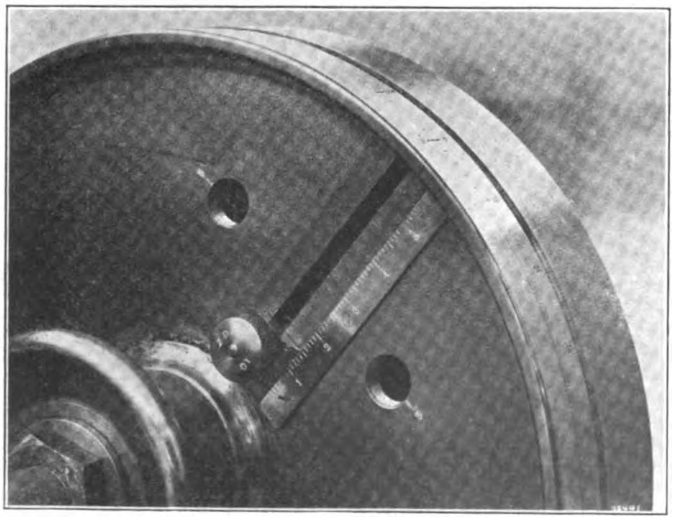


FIG. 5—STANDARDIZED WEIGHT IN THE HEADSTOCK OF THE MACHINE

The method of measuring the magnitude of a correction and the essential features of the machine for so doing are as follows: All revolving parts are ball-bearing mounted, including the rollers that carry the work. The frame is controlled in its position about the pivots *AA* by the main cantilever spring *B* so that, when the lock *C* is released, a free oscillation can occur in a vertical plane about the pivots *AA*. This suspension is so deli-

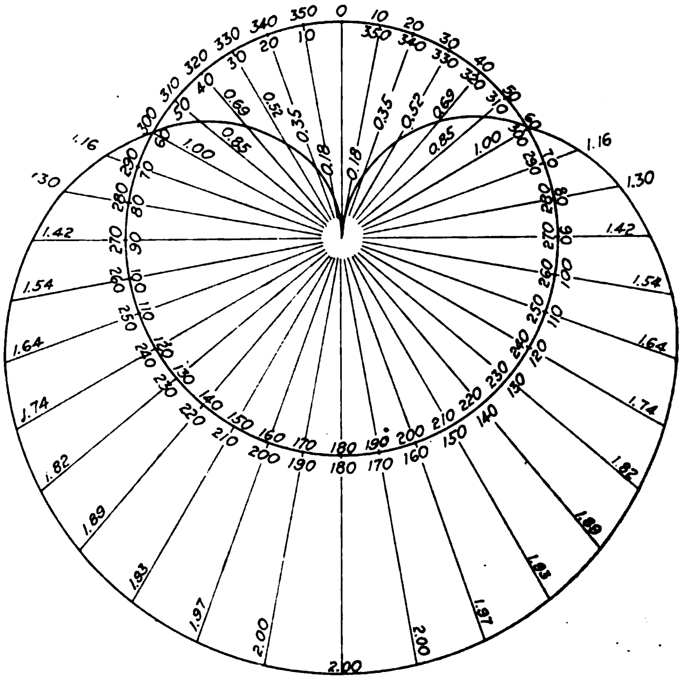


FIG. 6—POLAR DIAGRAM SHOWING THE ANGLE AT WHICH THE CORRECTION WEIGHT MUST BE APPLIED TO SECURE BALANCE

1—Observe the Greatest Amplitude of Vibration Obtained from the Freely Rotating Work
2—Move the Correction Weight from Zero the Proper Amount, Depending on Calibration, for This Amplitude and Observe the Second Amplitude
3—Divide the Second Amplitude by the First Amplitude, Expressing the Ratio as a Decimal
4—On the Chart of Angles Find Where Radius Equals the Ratio Determined and Turn the Correction Disk to This Angle

cate that, although the combined weight of the parts carried on the frame is more than 500 lb., perceptible oscillations can be produced by blowing upon the apparatus with one's breath. The amplitude of oscillation can be observed on the dial indicator at the left of the operator, which is illustrated in Fig. 4.

The revolving parts of the headstock are coupled to the work, and they are perfectly balanced. To take a reading, the system is set into revolution at a speed slightly above the natural period or critical speed of the frame, 100 to 110 r.p.m., and allowed to slow-down only by the slight and constant friction of the ball bearings. As the critical speed of the frame is passed through, a maximum oscillation will occur which is directly proportionate to the resultant of unbalance in the overhanging end of the work. By a separately determined calibration factor, the correction in ounce-inches is known for the plane of a standardized weight provided on the headstock, as shown in Fig. 5. This is revised subsequently according to the ratio of the longitudinal distances from the pivots of the standardized weight and the plane of actual correction on the work.

To obtain the angular position at which this correction is required, the quantity is applied arbitrarily on the headstock at an assumed angle, and the system is speeded-up again and allowed to slow-down through the

critical speed as before. The second amplitude will differ from the first amplitude by a ratio dependent upon the angle between the point of arbitrary application and the point required. The trigonometric value of this ratio is $R = 2 \cos \frac{1}{2}$ (180 deg. — A), where A is the required angle. This function, plotted on a polar diagram, is shown in Fig. 6, which can be used for ascertaining the angle after obtaining the ratio of the second amplitude to the first. The equivalent of Fig. 6 also is built into a special slide-rule which can be seen mounted on the instrument column in Figs. 3, 4 and 7. Scales also are provided here for employing the calibration factor and taking ratios; so, the operator is relieved of all computation. Lastly, the angle ascertained is set off on the headstock and the result is checked by another run.

Production Procedure

The production procedure of these steps is shown by these illustrations as follows:

Fig. 4 shows by the dial reading the amplitude of a first vibration due to unbalance of the work only, and the corresponding setting of the slide-rule; block F on the upper scale remaining at the calibration factor, and block 1 on the sliding scale being set to the dial reading. Setting the slider for multiplication, the number of ounce-

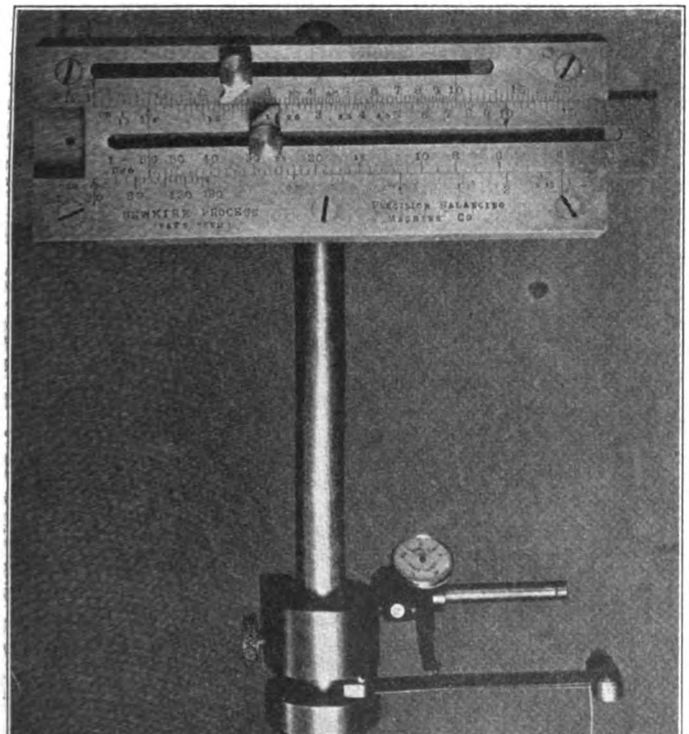


Fig. 7—Special Slide-Rule by Which the Angle at Which the Correction Weight Must Be Applied Can Be Calculated

inches of correction on the headstock is read on the upper scale above block 1. Fig. 5 shows the corresponding adjustment of the correction weight on the headstock with the disc remaining at zero-angle division. Fig. 7 shows the second amplitude and the corresponding setting of the slide-rule, with the sliding scale moved over until block 1, still set at the first amplitude, indicates the second amplitude on the upper scale, which is the position for division. The ratio of the second amplitude to the first amplitude might then be read at the left-hand

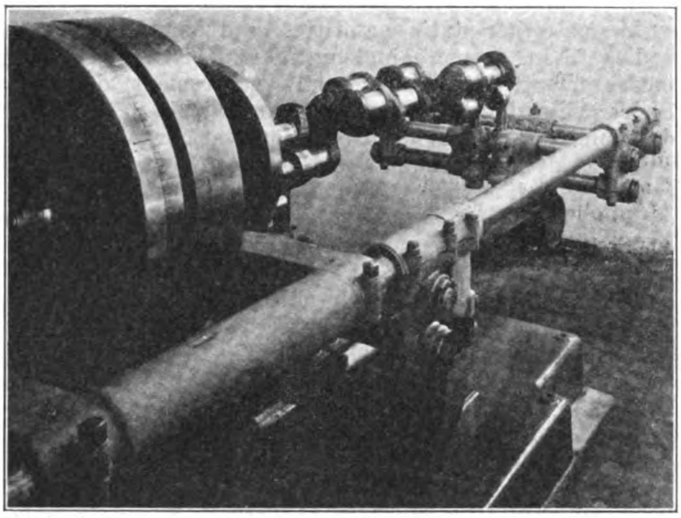


Fig. 8—View of Headstock Showing the Disc Turned to the Angle Indicated by the Slide-Rule as Being the Proper One To Secure Balance

index but, instead, the angle is read directly from the bottom scale. Fig. 8 shows the disc turned to the angle determined, and also gives visual evidence of a check result, as this series of photographs was made with an actually balanced crankshaft in the machine, that had been thrown out of balance by the application of a weight on the flange. Fig. 8 shows the self-determined alignment of the correction on the headstock with the weight on the flange; both weights lining with each other since they are on opposite longitudinal sides of the pivots.

Fig. 9 represents a cut-meter, a production instrument used with this machine on certain classes of work such as conventional six-throw crankshafts. By setting the blades to the quantity and angle determined as balancing-machine readings, the relative depths of cut required to remove from adjacent corners of the crank forging are read directly from the blade intersections with reference to the parallel rulings on the dial.

Accuracy and Rapidity of Measurement

The accuracy of measurement made with this machine can be guaranteed when appearing perfect within 0.2

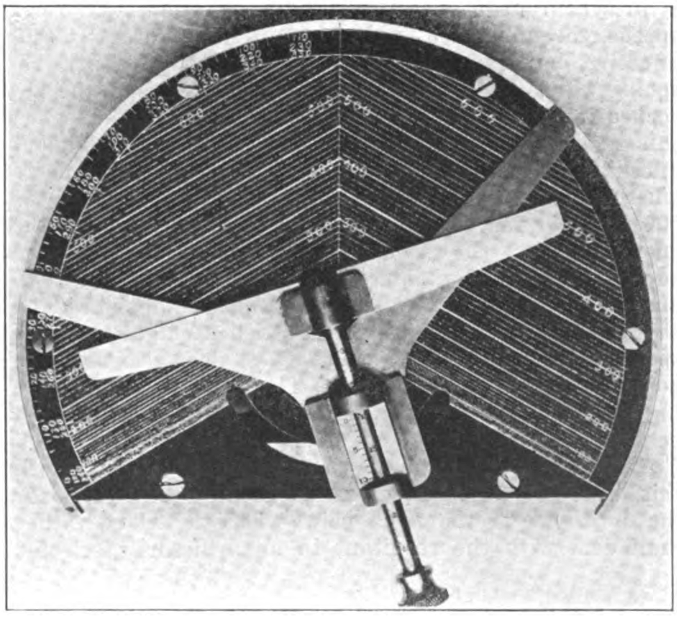


Fig. 9—Cut Meter That Indicates the Amount of Material That Must Be Removed from a Crankshaft To Secure Balance

oz-in. for either static quantities or dynamic couples on bodies 18 in. or more in length. This is a much finer measurement than commercial requirement demands. Short bodies, such as flywheels, are within 1 oz-in. for dynamic couples and 0.2 oz-in. for static values. Inasmuch as dynamic couples are of the dimension weight times the radius times the axial distance between planes of application, the above accuracy for flywheels is in every respect as fine as for longer bodies, such as multiple-throw crankshafts.

The machine is universally adjustable for any size of work up to a 24-in. swing and 32 in. between bearings. The accuracy above specified would suffice in most cases for rotors as light as 15 lb.; and work up to 750 lb. can be handled in the same machine, thus affording a combination of exceptional range in balancing facility.

The rapidity with which measurements can be made is evidenced by the fact that the process is definite. It leads by successive steps to the result, which is independent of any personal skill or knack on the part of the operator, and eliminates experiment. Determinations are made at the rate of 10 to 15 per hr., depending upon the amount of error existing in the work and upon the accuracy desired. Marks placed on the work direct the operation of applying corrections, which can be done at any time later, and does not require any rechecking of the result. This rate of production is worthy of considerable note when it is remembered that the accuracy is within a small fraction of work regularly passed and at a slower rate under present factory conditions. In fact, it can be said truthfully that many balancing operations, as now employed in factories, are but very little better than none.

Misconceptions of Balance

In conclusion, reference should be made to several misconceptions that are prevalent and are related to the general subject of balance. First is the idea that a body may be in balance at one speed and out of balance at another speed. This conception has arisen from the evidence that objectionable vibration may occur at one speed, but not at some other speed. In such instances the speed of objectionable vibration is only a "critical speed," or one where the elastic period of the supporting means is in synchronism with the speed of rotation. At other speeds, the unbalance producing vibratory forces is still effective, but does not meet with a periodic elastic response. In other instances, vibratory forces may be absorbed by the structure up to a certain limit and, when this is exceeded due to an increase in the speed, violent knocking or rattling may result. To the question as to whether a body balanced at a low critical speed such as 100 to 110 r.p.m. will be balanced at higher speeds, the answer is that the critical speed is a function of the support and a body balanced at its critical speed is balanced at all speeds. Trials of crankshafts balanced on this machine and run on a test-stand at higher critical speed have proved this conclusively, as does also the foregoing mathematical deduction. The use of a low balancing-speed has the added advantage that the body will not be distorted by centrifugal force.

Another question sometimes raised in connection with this process is whether we cannot locate and correct for the unbalance at the exact point where it exists, rather than deal with the resultant in an arbitrarily selected plane. It is neither necessary nor desirable to do this, as evidenced by the following: The engineering materials employed for all rotors are sufficiently rigid so that small corrections at desirable points may balance as resultants any errors existing elsewhere. Consider for a moment the metal composing a single crank-throw. If we were to measure the unbalance existing at this point and correct it by removing metal, the amount required to be removed would be the entire mass composing the member. Therefore, it is necessary that we correct for the resultant of the entire shaft, including all throws, and not for the errors existing at individual points in the piece.

The use of this machine does, however, open a wide field of research on the design of crankshafts, making possible the determination of centrifugal bending-stresses and the bearing pressures that result from them, and this would be of great assistance in problems of counterbalance and the like.

THE DISCUSSION

Chairman W. G. Clark:—After seeing the balancing machine in operation, I wish to state that the two things which impressed me as being new and absolutely reasonable are first, that this machine will balance a rotating part that is put in the rotating balance so that it absolutely eliminates all previous stages of balancing; and second that, contrary to our previous beliefs on the subject, it is not necessary to balance various parts of a crankshaft, because this machine eliminates all counterbalancing except two balancing operations in two separate and distinct points that comprise the entire balancing operation. This is radically different, according to my knowledge, from previous balancing operations.

A. F. Moyer:—With reference to the counterbalancing of the shaft, the point is simply that we can overcome any small variations that are due to the process of manufacture. A counterbalanced shaft may obviate a tendency to run in a bent condition, but that has nothing to do with balancing a shaft as a rotating body.

Chairman Clark:—I did not mean to eliminate counterbalance. I meant external counterbalance after the shaft is forged.

Mr. Moyer:—Only two points are necessary.

Chairman Clark:—That is my meaning.

A. W. Scarratt:—I attended the demonstration of the balancing machine and certainly was very much impressed with it. To my mind, it is the best balancing machine that has ever been developed. The engineers of the Society, especially, will be very much interested in getting first-hand information regarding it.

J. L. Mowry:—Will Mr. Moyer restate the application of the hyperbolic law, which he began with?

Mr. Moyer:—This is a simple matter of computation by moments. In other words, a quantity existing at one distance from the center of moments will have the same effect as a smaller quantity farther from the center, in inverse proportion, and the same effect as a larger quantity nearer the center. The algebraic expression of this gives the equation $xy = k$, which represents an hyperbola according to analytical geometry.

E. R. Greer:—Is there only one hyperbola that can pass through a given point?

Mr. Moyer:—Yes, when referred to a given set of axes, only one hyperbola can pass through a given point.

Coming Meetings of the Society

THE DETROIT PRODUCTION MEETING

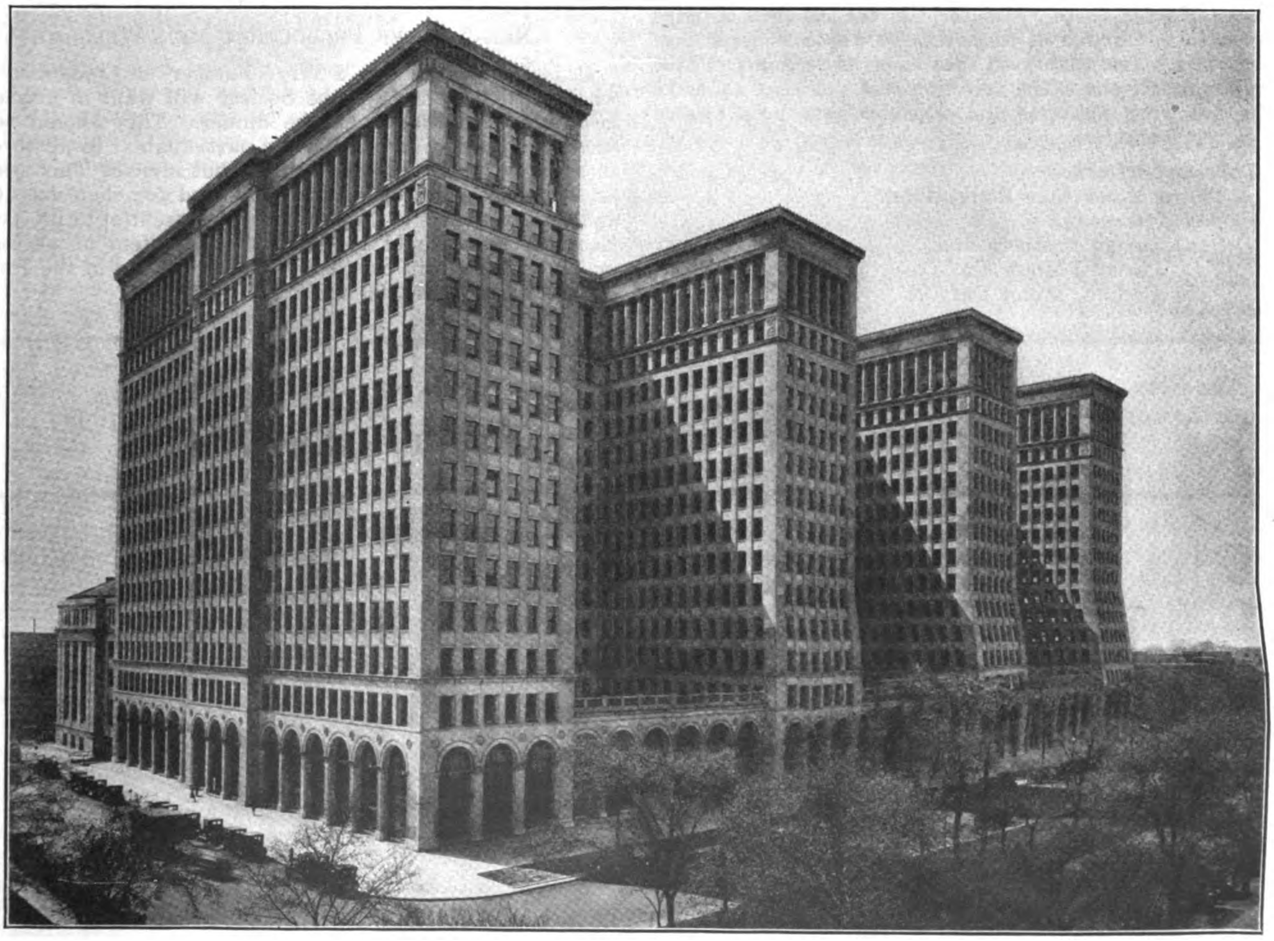
THE production executives and factory men of the automotive industry will hold their first meeting of national scope in Detroit, Oct. 26 and 27. This meeting has been arranged by the Society to bring the advantages of the business or professional meeting to the producing arm of the industry. Automotive sales, service, engineering and administrative conventions have been conducted in an organized way for many years. The production department alone has been unable to enjoy the benefits of cooperative reasoning and a closer social unity. It seemed logical that the automotive production men could be served by the Society of Automotive Engineers just as the designers and engineers have been. Thus we find the establishment of the annual Production Meeting and its location for 1922 appropriately set in the city producing the largest volume of automotive vehicles.

The meeting is expected to attract production and machine-tool men from all of the important centers of the industry. It will continue for 2 days and will include professional meetings, factory inspection visits and the Production Dinner. The final and definite details of the program will reach the members in a special issue of the *Meetings Bulletin* about Oct. 15. At least four factories will be visited during the two afternoons of the meeting. The two morning meetings will include papers and discussion covering a diversified group of production subjects, all of which will be found of value in practical production work. Little of a theoretical nature will be presented, it being considered far more important from the shop man's viewpoint to deal with actual factory experience. The following paragraphs deal with the individual features of the meeting.

The Professional Meetings

The professional meetings will be held in the General Motors Building on the Grand Boulevard near Woodward Avenue. The building may be reached from the center of the city by either the Woodward Avenue surface car or the Second Avenue motorbus line. The meetings will start promptly at 10 o'clock on each of the two mornings. There will be about five papers presented in each meeting with periods after each paper for questions and discussion. All of the papers have been written by members of the production staffs of automobile building plants. The program was intentionally limited to this group of men by the committee in charge of papers. Although the names of all of the authors cannot be announced at this time, papers are assured from the Studebaker, Ford, Packard, General Motors, Maxwell, Wills and Willys-Overland organizations. There is a possibility that other names will be added to this representative list by the time the final program is published in the *Meetings Bulletin* previously mentioned.

The Meetings and Papers Committee composed of K. L. Herrmann, chairman, T. J. Litle, Jr., C. Harold Wills and F. A. Whitten has worked diligently to secure material for



Office Building of the General Motors Corporation, Detroit, in Which the Production Meeting of the Society Will Be Held, Oct. 26 and 27

presentation that will reward those in attendance with a fund of practical and valuable information. The subjects vary from a treatment of miscellaneous shop practices to the problem of labor control. The group-bonus plan of labor control promoted by the General Motors Corporation will be described and some of its accomplishments presented. Unusual experiences in the study of gear noise will be recorded by the Packard production men. The chairman, K. L. Herrmann, will read a very comprehensive study of gear tooth variations and their relation to noise. His experience represents that of the Studebaker Corporation of America. The highly competitive automobile market demands a continued reduction of production costs. This condition directs attention to the savings resulting from salvage work and a member of the Overland organization will talk on this subject.

Inspection methods, machining of splined shafts and tool costs are other matters on which papers are expected. The Maxwell production department will be represented with a paper on simplified tools for parts production, comparing machining, labor and overhead costs with those on similar operations where expensive equipment was used.

Arrangements have been made to serve lunch in a room adjacent to the meeting hall immediately following the close of each professional session. This will not only prove to be an accommodation to the members but the lunch period will serve as an appropriate time for personal chats and afford an opportunity for better acquaintance among the production executives. The charge for the lunch will be a nominal one.

The Factory Visits

The factory inspection visits will start from the General Motors Building at 2 o'clock each afternoon soon after the lunches are finished. The committee in charge of this phase of the meeting is composed of K. K. Hoagg, chairman, T. J. Litle, Jr., Howard A. Coffin, E. F. Roberts and George E. Goddard. It was realized by the committee that the limited time available for these visits forbade the conduct of trips through a large number of factories. It was considered best to select only a few plants so that each inspection could be made a thorough one. The selection was not easy to make but the following factories and schedule have been agreed upon as representative.

Thursday Afternoon
Ford Motor Co.—Rouge Plant
Friday Afternoon
(a) Dodge Brothers and
Packard Motor Car Co.
(b) Cadillac Motor Car Co.

The Ford-Rouge plant is probably the most comprehensive of its kind in the world, embracing the production of raw metals, finished parts of all kinds and the construction of bodies. It will be necessary for the members to select which of the optional trips they wish to take on Friday afternoon. The Dodge and Packard factories are well known and both will be visited in one trip. The Cadillac factory represents the most recent practice in factory buildings, machine tools and equipment for it has been completed and occupied since the war. Members will be driven to the factories in cars provided by the companies visited. They will be conducted through the shops by members of each company's production staff who will be familiar with the various operations that are performed in the different departments of the plant and the machines that are employed for the purpose.

The Production Dinner

The Production Dinner will be held at the Statler Hotel, Thursday evening, Oct. 26, at 7 o'clock. Tickets for the Dinner may be purchased from the Detroit Section office of the Society in the Book Building or ordered by mail from the New York City office. All seats will be reserved, giving whatever preference there may be in accordance with the order in which the applications are received. All requests for tickets must be accompanied by a check to cover the cost of tickets ordered at $3.50 each. The Production Dinner will be strictly informal both in dress and atmosphere. It is intended as an occasion for the promotion of intimate friendships among the men who are responsible for the construction of motor vehicles and their parts. Only one dinner speech will be made other than the toastmaster's introduction. This speech will be inspirational and the committee may be relied upon to select an able talker with a message. Look for the *Meetings Bulletin* around Oct. 15 for the dinner details; meantime send along a remittance for your ticket so that you will be sure of a seat.

Non-Member Production Men Welcome

It is recognized that a large number of production men who are not members of the Society will want to attend the meetings, inspections and the dinner. They should be assured by the members that their participation in all three of these activities is not only welcome but invited. This meeting is a producers' meeting. It is intended for shop executives, and their presence in large numbers is essential to its success. Each member should cooperate to the extent of personally inviting the men in his factory to participate in the Production Meeting on Oct. 26 and 27.

COMING SECTIONS MEETINGS

Will Be Found Described in This Issue of THE JOURNAL on p. 380

Rear-Axle Gear Calculations by the Compressive-Stress Method

By JOSEPH JANDASEK[1]

Illustrated with DRAWING

THE practice of checking the design of gears by the bending-stress method, now in common use, is shown to be merely guesswork, when the question of durability is involved. It is sufficient when the bending strength only is sought, but as the wear on the face of the tooth often is a source of more trouble than is breakage, the ability to resist crushing is the best measure of the tooth's capacity. As the wearing quality depends upon the maximum surface-pressure, the size of gears must be based on the compressive stresses to which they are to be subjected. When the unit pressure for a given material is raised above a certain value, the wear of the tooth will increase rapidly. Consequently, it is important to know the limiting value of the unit pressure under which a gear will stand up.

Beginning with the theory of pressure between elastic bodies with curved surfaces, the author develops formulas for calculating the greatest allowable load on the pitch-line for spur and bevel gears. Incidentally attention is called to the variation of the capacity of pinions, when meshing with a large and with a small gear and, similarly, of ball bearings, when the diameter of the balls and the size and shape of the races are varied.

The capacity and strength of gears with straight and helical teeth are investigated. Formulas for checking the pitch and for computing the number of teeth when the compressive and bending stresses are equal, the face of gears in contact and the pitch-diameter of differential gears and pinions are worked out. Examples are given showing the application of the formulas to the design of helical bevel and differential-gears.

IN present-day practice most gears with straight teeth are calculated, or rather checked, by the bending-stress method by which the maximum permissible load that will assure freedom from breakage of the teeth at the roots can be determined. As a rule, this is the only checking that the average gear-designer does. Such calculation, however, is merely guesswork, as will be proved in this article, mainly because it is the wear on the face of the tooth and not the breakage at the root that causes the most trouble. It is a well-known fact that gears designed to give at least a reasonable amount of service do not break but wear out.

So far as tooth breakage is concerned the strength of gears calculated by the bending-stress method is sufficient for passenger cars as well as for trucks and tractors, for the rear-axle as well as for the change-speed gears. We cannot, however, say the same regarding the wear on the face of the tooth and it is only natural that the designer employing this method should not obtain the desired results on the first attempt. Then, of course, the process of redesigning, building and trying out the gears anew follows. This is not only expensive but requires time and some driving before the wearing quality of the gear can be ascertained; that is, whether the pressure on the face of the teeth is excessive so that backlash and noise will develop and result finally in the complete destruction of the mechanism, if replacements are not made in time.

[1] Engineer, Olds Motor Works, Lansing, Mich.

The breaking load is merely a measure of the strength of the gear. The only true and rational measure of the capacity of gear teeth is the ability of the face of the tooth to resist crushing, or the resistance of a tooth to abrasion or wear at certain maximum permissible pressures or compressive stresses. Gear sizes must be determined on the basis of the stresses on the tooth face, that is, the working, wearing and noise-causing surface. The wearing quality of gear-tooth faces depends upon the maximum surface-pressure, provided the quality of the material, toughness, coefficient of friction and tooth action are constant. The value of this maximum permissible unit compression depends upon the ultimate strength, which, again, is closely proportional to the hardness. This is the reason the designer always specifies the hardness number to obtain the desired resistance to wear. The other properties of the material, such as toughness and the coefficient of friction, must be taken care of in the way that different unit surface-pressures, or compressive stresses, are computed in different gear materials, even when their hardness numbers are equal.

The relation of face hardness to wearing quality explains the great success of case-hardened gears. They possess a hard and strong case of extremely high ultimate-strength but have a soft and comparatively weaker core. The bending-stress method takes into consideration only the low-carbon core; but the tooth faces that are under very high compressive-stresses are also much stronger, being case-hardened. Hence, the design can be successful.

Another example proving this theory is tempered gears. They are never used for rear-axle gearing, because they wear out; that is, the tooth faces cannot stand the pressure. Yet we know that tempered steel possesses a considerably higher ultimate-strength than the soft core of the low-carbon case-hardened tooth

It is apparent that it is the compressive-stress on the tooth face that determines the capacity of gears and not the bending-stress at the root of the tooth. The compressive-stresses in gear teeth are determined by the area of the contact surface, which is dependent upon the curvatures of the gear and pinion teeth and the face over which the load, or pressure normal to the tooth, is distributed. This contact surface carries the load. The pitch should be sufficient only to resist fracture. A larger pitch is uneconomical, as there is no increase of power-transmitting capacity. As the number of teeth decreases, the uniformity of motion decreases and the consequent noise increases.

It is known that there is a certain value of the unit pressure for a given material of a definite hardness, above which the wear will increase rapidly and the life of the gear will be shortened greatly. On the other hand, when the unit pressure is kept sufficiently below this critical value, the gears stand up well and have a long life. Consequently, it is not the amount of wear in a definite time that we want to know but the limiting value of the unit pressure on the tooth face under which the gear will stand up.

The wearing quality of the final-drive gears can be slightly improved by allowing the teeth to find their proper bearing surfaces by starting under a partial load only. Mounting the gears on rigid shafts in rigid housings is also necessary. The driving pinion should be provided with a bearing on each side to ascertain the correct and efficient mesh along the whole length of the tooth face. An overhanging pinion is usually out of alignment, especially under heavy pulls, because of the large bending-moment on the shaft, and is bound to wear-out soon on account of the increased compressive-stresses. Strong housing must be provided, with easy and accessible gear and pinion adjustments. Bearings ought to be rather oversized to assure good and lasting contact.

The propeller-shaft brake or the transmission brake ought to serve only as an emergency brake because the action of changing the function of a pinion from that of a driving member to a driven one is very severe and hard on the gears and bearings. Using the engine as a brake is also an expensive practice for the same reason.

Spur and Bevel Gears

In the foregoing I have shown that the wear of gears depends upon the unit compressive-stresses on the contact surface of the tooth. To determine this contact area, or rather the maximum compressive-stress, we must begin with the theory of the pressure between elastic bodies with curved surfaces. In the following I have derived formulas for the greatest allowable load on the pitch-line.

The capacity in pounds of spur gears with straight teeth is

$$W_c = [10^{-8}C^2pf]\ [n/(1+a)] \tag{1}$$

The capacity of bevel gears with straight teeth is

$$W_c = [10^{-8}C^2pf]\ [n/\sqrt{(1+a^2)}]B \tag{2}$$

Further, it is known that the strength of spur gears with straight teeth is

$$W_b = Spfy \tag{3}$$

The strength of bevel gears with straight teeth is

$$W_b = Spfy_1B \tag{4}$$

where B is the reducing factor and equals

$$B = 1/3 + l/3L + l^2/3L^2 \tag{5}$$

$a = d/D =$ value of the gear-ratio, pinion to gear, which must be added for external gears and subtracted for internal ones
$C =$ the maximum allowable compressive-stress on the tooth face at the pitch-line in pounds per square inch for a given speed, V
$f =$ the face of gears in contact in square inches
$L =$ outer pitch cone radius for bevel gears
$l =$ inner pitch cone radius for bevel gears
$n =$ number of teeth on the pinion
$p =$ circular pitch in inches
$S =$ the maximum allowable bending-stress in pounds per square inch
$W_c =$ the maximum tangential force in pounds at the pitch-diameter with regard to the compressive-stress at the pitch-line, assuming that only one pair of teeth is in mesh at a time
y and $y_1 =$ factors depending upon the number of teeth

From the equations (1) and (2) it can readily be recognized that the capacity of a pinion according to my compressive-stress method depends not only upon its own dimensions but also upon those of its mating gear.

The same pinion meshing with a large gear possesses a greater capacity than when meshing with a small one, this being the influence of ratio a. Placing the capacity W_c of a spur gear, meshing with a gear of the same size, equal to unity, I have found that the same gear possesses a capacity 2.7 times greater when in mesh with an internal gear having the ratio of 1 to 4. The total difference in capacity, therefore, can be 270 per cent, if the influence of the mating gear is neglected. Hence, it is evident that when bending stresses alone are used for determining the capacity of gearing the process is merely guesswork. It might or might not give the results desired depending upon the factors used. These include gear-ratio, number of teeth, quality of the material, workmanship and exactness of tooth profile.

We find the case to be exactly the same with ball bearings where the capacity depends not only upon the diameter of the balls but upon the size and shape of the races. But the strength of a pinion calculated solely upon the basis of bending stresses depends upon its own dimensions only and remains constant whether the pinion meshes with an external gear, rack or an internal gear of any size.

According to the same theory of maximum surface pressures I have found the formula for helical spur-gear capactiy to be

$$W_c = [10^{-8}C^2pf]\ (n/[(1+a)\cos^2 H]) \tag{6}$$

where p is not the normal but the circumferential circular pitch and H is the angle of helix.

As it is not possible to calculate the strength of a helical tooth by bending stress owing to the complicated shape of the tooth, the shearing stress K of a tooth at a pitch-line is taken as a measure of the gear strength. Hence,

$$W_s = 0.4pfK \tag{7}$$

Before we step to the calculation of helical bevel-gear capacity let us recall a few facts about this type of drive.

There is practically only one type of final-drive in use for passenger-car axles and this is the helical bevel-gear, shown in Fig. 1. This drive is being used on almost all car models now on the market. The main advantage of this gear is in noiseless operation at all speeds. It permits the use of a smaller number of teeth than can be used with straight bevel-gearing, where the number of teeth must not drop below a certain minimum, usually 12 because, as the number of teeth decreases, the non-uniformity of motion and, consequently, the noise, increase rapidly. Helical gears also wear evenly along the tooth flank so that such gears, even if partly worn, transmit the motion uniformly and do not get noisy as soon as straight bevel-gears. For the same reason the increment of load due to the irregularity of impulse cannot become dangerous even at the highest speeds. Thus, the capacity of the gear is increased and a finer pitch can be used.

The angle of helix should be such as to allow sufficient angular advance, corresponding to the length of the gear face, and should be from 10 to 20 per cent greater than the circular-pitch.

The high-speed engines used on automobiles require comparatively high gear-reduction, that is, from 3½ to 1 to 4¾ to 1. Helical bevel-gears have made it possible to obtain this higher reduction in a single step without the danger of non-uniform and noisy operation. In reference to these large gear-ratios it should be noted that the accelerating ability is increased at the expense of the fuel economy. Economy demands the conservation of fuel. Consequently, a smaller top gear-ratio should be used. This small gear-ratio could be easily combined with either a geared-up fourth speed in the transmission or still better by employing a two-speed rear axle. It is interesting that on one side there is so much talk about increasing engine economy by 0.1 per cent while on the other side such well-known principles as those mentioned remain practically overlooked.

Helical Bevel-Gears

Helical bevel-gear drives with pinions having as few as six teeth have been produced successfully in extreme cases. With a small number of teeth the pinion must be made integral with the shaft, which is, however, the best practice anyway, as it does away with the troublesome key in the shaft, especially when the transmission brake is used.

As to the strength against fracture, it is usually assumed that a helical bevel-pinion of a definite diameter and a given pitch is capable of transmitting the same power at a certain speed as a straight bevel-pinion of the same diameter, pitch and face, although it is plain that the tooth action is different because the teeth of the helical bevel-pinion are curved and at an angle to the gear axis. Mathematically, such an assumption never has been proved. In fact, the total normal load on teeth and the end-thrust are considerably greater with the helical than with the straight pinion. Finally, the factor depending upon the number of teeth is absolutely unknown. On the other side, there is always a pitch-line contact somewhere and more teeth are in contact at one time. If the helix advance,

$$h = ftgH$$

is greater than the circular-pitch, by which is meant not the normal but the circumferential pitch, then the arc of action always includes at least one more tooth per gear than in a similar straight bevel-set.

In the foregoing I have demonstrated that the formulas (2) and (4) determine the capacity and strength of bevel gears with straight teeth with respect to their compressive and bending stresses. However, in the case of helical bevel-gears of comparatively small diameter, low peripheral velocity and small number of teeth, a further simplification can be made.

On the basis of the compressive-stress principle, I have derived the formulas for helical bevel-gear capacity, which are exact in all factors simply because the curvature of any tooth profile always is known with mathematical exactness. We have for the capacity of helical bevel-gears

$$W_c = [10^{-8}C^2pf](n/[\sqrt{(1+a^2)}\cos^3 H])B \qquad (8)$$

The factor $\sqrt{(1+a^2)}$ can be left out, because its value for large reductions in the case of rear-axle gearing is equal to almost unity. However, for racing cars with small ratios this factor cannot be omitted. The formula for helical bevel-gears with large ratios then will read

$$W_c = (10^{-8}C^2pf)(n/\cos^3 H)B \qquad (9)$$

As the helix angle is always around 30 deg., the maximum permissible compressive-stress, $C = 110{,}000$ lb. per sq. in., and the reducing factor, $B = 0.75$, or close to it, we can obtain a very simple formula for quick, approximate calculations.

$$W_c = 125pfn \qquad (10)$$

The above formulas constitute the only dependable way known for calculating the dimensions of helical bevel-gears.

For checking the pitch of helical bevel-gears I use the formula

$$W_s = 0.4\cos H(KpfB) \qquad (11)$$

Here it is assumed that the shearing stress is a measure of the gear-tooth breaking-strength, as it is known in the case of herring-bone gears. The value 0.4p represents the thickness of a ring-gear tooth on the pitch-line and p is not the normal but the circumferential pitch, $\pi d = np$.

For quick, preliminary calculation this formula can be simplified to

$$W_s = 0.26Kpf \qquad (12)$$

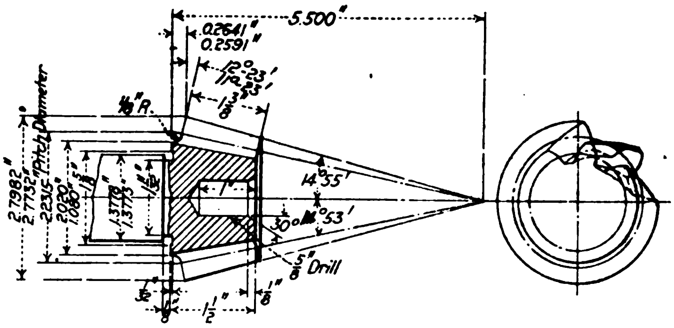


Fig. 1—Helical Bevel Gear Used To Drive Passenger Cars

Placing, in the equation, $W_b = W_c$

$$0.4\cos^3 HK = 10^{-8}C^2n$$

we get

$$n = 0.4\,(10^8\cos^3 H\,[K/C^2]) \qquad (13)$$

Thus we recognize that by keeping the stresses K and C constant, we can, within given limits, decrease the number of teeth n by increasing the helix angle H. The smaller the number of teeth, the larger will be the angle H for the same stresses.

Another way to obtain a smaller number of teeth n is to keep the stress C constant and decrease K. In practice the helix angle ranges from 28 to 32 deg., with an average of $H = 30$ deg.; the ratio of face to pitch, $b = 1.5$ to 1.8; the number of teeth, $n = 7$ to 12; the ratio $f/L = 0.25$. Hence, $B = 0.77$.

The pitch-diameters of bevel gears are limited, because of the required ground-clearance, to about 10 in. for the lightest cars, of some 2500-lb. total weight, complete, ready for the road; 10½ to 11 in. for medium-sized cars; and 11 to 12 in. for large cars.

The maximum allowable compressive-stress is $C = 105{,}000$ to 110,000 lb. per sq. in., according to the workmanship, at a speed of about 750 ft. per min., calculated on the basis of a piston speed of 1000 ft. per min. and a direct drive; and the maximum allowable shearing-stress, $K = 4000$ to 5800 lb. per sq. in., both stresses being figured for a 3½-per cent nickel-steel case-hardened pinion. The mating gear can be made from 1¼-per cent nickel or chrome-nickel steel, case-hardened. Fig. 1 is a sample drawing of a helical bevel-pinion with the necessary dimensions and specifications.

Application of Formulas

Example 1.—To determine the size of helical bevel-gears for a passenger-car rear-axle under the following conditions:

Engine, 3¼ x 5 in., six-cylinder
Gear-ratio, 1 to 4.5
Engine speed, N, at a piston speed of 1000 ft. per min. = 1200 r.p.m.
The horsepower on the basis of the National Automobile Chamber of Commerce formula = 25.35
The corresponding torque, T = 1330 in-lb.

Solution.—First, we have to decide the maximum diameter of the ring gear. Because of the road-clearance we select $D = 10¾$ in. when used for medium-sized cars. The pinion diameter will be $d = 10.75/4.25 = 2.39$ in. According to the formula (10) we have

$$W_c = 125\,pfn$$

However

$$np = \pi d$$

and

$$W_c = 2\ (T/d)$$
$$W_c = 125\pi df \qquad (14)$$

Hence, we get

$$2(T/d) = 125\pi df$$

and for f we have

$$f = (T/d^2) \times (2/125\pi)$$

and, finally,

$$f = 0.0051\ T/d^2 \text{ in., for } C = 110{,}000 \quad (15)$$

and

$$f = 0.0056\ T/d^2 \text{ in., for } C = 105{,}000 \quad (16)$$

Let us specify that the maximum compressive-stresses must not exceed $C = 105{,}000$. The width of face according to formula (16) will then be

$$f = 1\ 5/16 \text{ in.}$$

It is of interest to note that we have determined the dimensions of gears; that is, of the diameter and the face, without knowing the number of teeth or the pitch. This is the best way to calculate gears, as it is the diameter and face combined that carry the load.

Now we can calculate the number of teeth of the pinion, using formula (13)

$$\begin{aligned} n &= 0.4\ [10^{-8} \cos^3 H (K/C^2)] \\ &= 0.4 \times 10^{-8} \times 0.65 \times 4000/105{,}000^2 \\ &= 9.44 \end{aligned}$$

Thus, for the pinion we have $n = 10$ teeth, and for the gear $n_g = 45$ teeth. The pitch, not normal, will be

$$P = (\pi d) \div n = 0.75 \text{ in.}$$

Instead of using equation (13) to calculate the number of teeth, it would be better to use the formula (11) and determine the necessary pitch directly.

$$\begin{aligned} &W_s = (0.4\ K \cos H)\ (pfB) \\ &1110 = 0.4 \times 4000 \times 0.866 \times p \times 1.3125 \times 0.78 \\ &p = 0.786 \end{aligned}$$

We find that the nearest number of teeth is $n = 10$, and that the shearing-stress will be slightly increased in proportion to the reduced pitch, $K = 4190$ lb. per sq. in., which is perfectly safe.

Now we must check the helix angle, H, for the necessary overlap. Since the circular pitch $p = 0.75$, the advance of helix

$$\begin{aligned} h &= f\ tg\ H \\ &= 1.3125 \times 0.5774 = 0.758 \text{ in.} \end{aligned}$$

This leaves a comparatively small overlap, but nevertheless it is sufficient. A slight increase in the angle to say 32 deg. or, still better, an increase in the shearing stress K and the number of teeth on the pinion to say 11, would give a better overlap. It is preferable to have an overlap of from 10 to 20 per cent. Shearing-stresses up to $K = 5800$ lb. per sq. in. are safe if the workmanship and the adjustment of the gears are correct. By using a higher shearing-stress, up to certain limits, we obtain a finer pitch, more teeth, a greater overlap, a more uniform motion and noiseless operation.

We have now to check in our example the values B, K, and C, from the original formula (9), especially when the rounding of the face width and the original value of the coefficient B were much different. In our example the difference amounts to very little.

Differential Gears for Passenger Cars

In calculating the dimensions of differential gears with respect to the breaking stress, we should bear in mind that the gear dimensions must depend upon the maximum torque at the rear axle at low gear. However, to make certain that the differential gears will not wear-out too soon and to keep on the safe side we should check them for compressive-stresses to ascertain how much service can be expected.

The following data are for a good, compact design of differential gears:

Number of teeth of pinion, $n = 10$ to 11
Number of teeth of side gear, $n_g = 18$ to 20
Ratio of face to cone radius, $f/L = 1/3$ to $3/8$
Ratio, $l/L = 0.67$ to 0.625
Coefficient, $B = 0.71$, when $f/L = 1/3$
Coefficient, $B = 0.67$, when $f/L = 3/8$
Compressive-stress on the direct drive at a torque calculated on the basis of National Automobile Chamber of Commerce horsepower formula, $C = 120{,}000$ to 130,000 lb. per sq. in.
Bending stress on direct drive, $S_d = 15{,}000$ to 16,500
Bending stress on low speed at ratio of 3.3 to 1 in the gearbox, $S_l = 50{,}000$ to 55,000

In the foregoing example we start with the formula (2), which gives the capacity of straight bevel-gears.

$$W_c = [10^{-8} C^2 pf]\ [n/\sqrt{(1 + a^2)}]B \quad (2)$$

Further,

$$W_c = T/Dm$$

where

D = pitch-diameter of side gear
m = number of pinions, usually four

Substituting ⅜ L for f, 0.67 for B and aD for d, we obtain

$$\begin{aligned} L &= \sqrt{[(D/2)^2 + (aD/2)^2]} \\ &= D/2\sqrt{(1 + a^2)} \end{aligned}$$

and

$$f = 0.1875 D\sqrt{(1 + a^2)}$$

Further, $n\ p = \pi\ d = \pi\ a\ D$ and for $C = 120{,}000$ we will get

$$D = 0.20\sqrt[3]{(T/ma)} \quad (17)$$

For our case, $a = 10/18 = 0.55$, selecting the smallest possible number of teeth, and $m = 4$.

Then from formula (17) the pitch-diameter, $D = 3.62$ in.
We will take $D = 3.6$ in. for the gear and $d = 2$ in. for the pinion. Knowing the diameters we find the width of face from equation (2) to be $f = 9/16$ in., and the diametral pitch, from equation (4), to be 5 — 7, for stub teeth.

Maag Gears

An interesting example demonstrating the possibilities of improvement along the line of gear design is found in Maag's gears, which proves that research work in this field, when properly conducted by able engineers, is bringing forth remarkable results. Maag's gears represent the most scientific gear system of the present time.

Endeavoring to obtain gears of great durability and noiseless operation even at large reductions, the Swiss engineer Maag based his system on the well-known involute curve. The tooth action of two involute gears remains mathematically correct at different center-distances and is independent of the pitch-circle. By increasing the center-distance the tooth action remains correct, the angle of the action only being changed. The backlash, of course, then increases, but this can be corrected easily by increasing the thickness of the pinion teeth, a feature to be desired. Different involute angles, ranging from 15 deg. at small reductions to 25 deg. at the largest ones, are used for different gear-ratios. In this way the tooth action and the proportion of roll to slide for each particular ratio is much more favorable than when the same angle is used for all ratios. At present we use a 20-deg. pressure-angle for spur as well as for bevel gears of any reduction for the sake of interchangeability. However, for final-drive and for large production this interchangeability is of no importance and special gears can be used.

Maag's pinion teeth are strong and noiseless and the gears have a greater capacity owing to the increased involute angles and the proportion of roll to slide. Hence the allowable compressive-pressures can be increased and

extremely high reductions are obtainable. All these features are very desirable in automotive gears and therefore the application of this or a similar system to automotive practice would result in increased efficiency, noiseless action, durability and greater power-transmitting capacity of gears.

In the foregoing I have demonstrated the application of my compressive-stress method, which is the only method of calculating the capacity of gears in connection with durability or wear. In the case of helical gears it is the only rational way of determining them at all. This method is of still greater value in making possible a ready investigation of a tooth of any form and affording a rational comparison of the merits of different tooth-forms from the standpoints of capacity and wear, especially where weight and size are of primary importance.

Compressive stresses on tooth faces and the ratio of roll-to-slide action are the principles that require study and must be taken well into consideration if we wish to obtain a really efficient and durable gear.

LETTER BALLOT ON ADOPTION OF STANDARDS

THE proposed standards and recommended practices printed on p. 475 of the June 1922 issue of THE JOURNAL and approved by the Standards Committee at the meeting at White Sulphur Springs were approved by the Society members by the letter-ballot which closed Aug. 19.

The recommendation on Flywheel Housings, however, which involved increasing the minimum diameter and depth of the clearance space for crankshaft flywheel bolts, received 10 negative votes and as a result of the reasons submitted in support of these votes the Council voted to withhold it from publication in the S. A. E. HANDBOOK and to refer the recommendation back to the Engine Division for joint consideration with the Transmission Division.

The report of the Screw-Threads Division covering allowances and tolerances for screws and nuts taken from the report of the working committee of the Sectional Committee on Screw-Threads of the American Engineering Standards Committee was also withheld from publication in the S. A. E. HANDBOOK by vote of the Council. This action was taken because the Society had been informed by the American Society of Mechanical Engineers that at a meeeting on Sept. 15 the Sectional Committee proposed certain changes which affected in a general way the entire report of the Screw-Threads Division on Screw-Threads.

The complete vote on all of the recommendations considered at White Sulphur Springs is tabulated herewith. The first column gives the number of affirmative votes; the second column the negative votes; and the third column the number of members who did not vote either way.

	Yes	*No*	*Not Voting*
Tractor Drawbar Adjustments	91	1	210
Motor-Truck Front-Axle Hubs	137	2	163
Breaker-Contacts	157	0	145
Ignition-Distributor Mountings	157	0	145
Magneto Mountings	165	1	135
Starting-Motor Flange Mountings	166	0	136
Flywheel Housings	168	10	124
Leaf-Spring Steel	152	2	146
Steel Spring-Wire	158	0	142
Electric Incandescent Lamps	145	0	155
Electric Incandescent Lamp Voltage	150	0	150
Head-Lamp Illumination	142	0	158
Head-Lamps	143	0	157
Motorboat Lighting Voltages	113	0	187
Aluminum Alloys	120	1	179
White Bearing Metals	117	0	183
Wrought Non-Ferrous Alloys	113	0	187
Ball Studs	163	1	136
Lock-Washers	187	1	112
Passenger-Car Front Bumpers	147	1	152
Plain Steel Washers	185	0	115
Serrated Shaft Fittings	168	2	130
Rod-Ends	173	1	126
Tank and Radiator Caps	166	1	133
Top-Irons	106	0	194
Screw-Threads	192	0	108
Screws, Bolts and Nuts	194	0	106
Frame Brackets for Springs	146	2	152
Spring-Eye Bushings	154	2	144
Flywheel Pulley Lugs	99	2	199

The August, 1922, issue of data sheets for the S. A. E. HANDBOOK will be sent to the members early in October. It is important for members using the S. A. E. HANDBOOK in their work to check their books so that they may be sure that all, as well as the most recent, sheets issued to date have been inserted properly. A list of all the S. A. E. HANDBOOK pages and their latest publication dates is given on the first sheet of the August issue of data sheets. These dates should be checked to the dates under the emblems appearing at the bottom of each right-hand data sheet to verify the completeness and uptodateness of the S. A. E. HANDBOOK.

It should be borne in mind that the Contents is arranged so as to aid members in limiting their S. A. E. HANDBOOK to only those pages which are of interest, the standards being classified according to the automotive industries to which they apply by a system of key letters. Members interested, for instance, in only tractor engineering may want to retain in their books only those standards which are followed by the letter *T* in the Contents. The standards in the S. A. E. HANDBOOK may therefore be considered as being classified in the Contents in two ways; by parts and materials, such as powerplant and electrical equipment, the classifications being denoted by letters prefixed to the page numbers; and by automotive industries, the industries being denoted by letters appearing in the right-hand column in the Contents.

Activities of the Sections

THERE is every reason to believe that the Society is embarking upon one of its most successful Section years. It is evident from the meeting programs announced to date that the Section officers are selecting speakers of the first order and confining discussion to topics of specific interest. October opens the Section year in earnest. The schedule of meetings announced herewith includes every Section. The subjects are sufficiently diversified to reach all corners of the industry and in each case the papers are to be given by men of prominence in the respective fields.

Photographs of some of the Section Chairmen and Vice-Chairmen are reproduced on the facing page. The following list gives the name of the office and the Section for the various officers:

Name	*Office*	*Section*
A. H. Bates	Chairman	Minneapolis
O. C. Berry	Chairman	Indiana
T. F. Cullen	Chairman	Pennsylvania
K. K. Hoagg	Vice-Chairman	Detroit
B. S. Pfeiffer	Vice-Chairman	Mid-West
C. H. Warrington	Vice-Chairman	Washington

Buffalo Section

The Buffalo Section of the Society will open its season with a meeting on Oct. 18 at the Engineering Society Club Rooms in the Iroquois Hotel. L. H. Pomeroy, consulting engineer, will read a paper on Light Reciprocating Parts and Their Effect on Car Design. Mr. Pomeroy's contributions to the Society are always received with interest. It will be remembered that his experience was largely acquired in Great Britain where at one time he was chief engineer of Vauxhall Motors, Ltd.

Cleveland Section

The Cleveland Section will start its season with a meeting on Oct. 20. George M. Graham, vice-president of the Chandler Motor Car Co., will address the meeting on The Sales Manager and the Engineer. Mr. Graham is well known in the industry as an able speaker and a student of the automobile owner's tastes and demands. He is qualified to predict those changes in design that increase the sales appeal of the motor car and these are certain to impress the automotive engineer. There will be an organized discussion of this paper which is expected to contribute greatly to the value of the meeting. The Cleveland Section holds its meetings in the rooms of the Cleveland Engineering Societies, at the Hotel Winton, and they begin promptly at 8 o'clock.

Dayton Section

The members of the Dayton Section will gather at the Dayton Engineers Club on Oct. 3 to hear C. F. Kettering, president of the General Motors Research Corporation, speak on Some Problems Confronting the Automotive Engineer. Mr. Kettering needs no introduction to the members of the Society. His broad experience and keen vision guarantee the presentation of much food for engineering thought. Few men are able to present scientific discussion in a manner that is both serious and entertaining. This C. F. Kettering can do and he is always assured an attentive and appreciative audience. This meeting will start at 8 o'clock and members from all Sections of the Society are welcome.

Detroit Section

The Detroit Section has not planned any special meeting for October because of the Society's Production Meeting that will be held in the General Motors Building on Oct. 26 and 27. The Section has taken an active part in the planning and arrangement of the Production Meeting. Its officers and members have been working diligently to make the program both interesting and instructive. It was largely due to the success of the Detroit Section's production meetings last spring that the Council decided to schedule this national meeting and to designate Detroit as its location. The Detroit Section welcomes the opportunity of entertaining the members of the Society from other parts of the Country and hopes that they will come in large numbers to the Production Meeting.

The September meeting of the Detroit Section was addressed by A. A. Bull, chief engineer of the Northway Motor & Mfg. Co. Mr. Bull abstracted the paper on Oil-Pumping that he read at the Summer Meeting of the Society last June. He also extended his discussion of the subject, adding experience gained subsequent to that meeting. The Section had arranged to have the paper discussed by a number of the leading automotive engineers of Detroit whose study of this problem qualified them to present information and data supplementing that given by Mr. Bull. The material presented in the discussion period added greatly to the interest and value of the meeting. Piston and ring design, lubrication systems, oil-pressures and crankcase-oil dilution all received consideration. It was generally agreed that the thorough treatment of aggravating troubles, such as oil-pumping, at Section meetings, led to a better general understanding of the controlling factors and enabled engineers to combat them to the benefit of the entire automotive industry.

Metropolitan Section

W. B. Stout, president of the Stout Engineering Laboratories, Inc., Detroit, will present a paper on the Modern Airplane and All-Metal Construction before a meeting of the Metropolitan Section on Oct. 19. Mr. Stout has been engaged in the design and construction of all-metal monoplanes for the Navy during the past year. In this work he has developed structural shapes and methods for working duralumin that are a distinct advance in this field. Mr. Stout has long been an advocate of the internally trussed type of metal monoplane for commercial flying because of its decreased parasite resistance and increased durability. There

Schedule of Sections Meetings

OCTOBER

3—Dayton Section—Some Problems Confronting the Automotive Engineer—C. F. Kettering

4—Minneapolis Section — Manufacture of Petroleum Products for Automotive Uses —N. S. Kingsley

5—Washington Section—A Road Test Method for Comparing Fuels—R. E. Carlson

11—Buffalo Section — Light Reciprocating Parts and Their Effect on Car Design— L. H. Pomeroy

12—Indiana Section—Trend of Research—C. F. Kettering

19—Metropolitan Section—The Modern Airplane and All-Metal Construction—William B. Stout

20—Cleveland Section—Sales Manager and the Engineer—George M. Graham

20—Mid-West Section—C. F. Kettering

26—Pennsylvania Section — Manufacture of Automobile Bodies

27—New England — Wire Wheels and Disc Wheels—O. J. Rohde

O. C. BERRY

BENJAMIN S. PFEIFFER

K. K. HOAGG

CHESTER H. WARRINGTON

A. H. BATES

T. F. CULLEN

seems to be a general tendency to adopt this type of airplaine, particularly abroad, and Mr. Stout's paper is expected to be of special interest for this reason. An informal dinner at 6:30 p. m. will precede the meeting which will be held at the Automobile Club of America, 247 West 54th Street, New York City, at 8 o'clock.

The Metropolitan Section Meeting of Sept. 21 resulted in a very lively discussion of lubrication systems. G. A. Round, who gave a paper on Oil-Pumping, contended that the problem was most serious in engines employing force-feed systems. The adherents of the force-feed scheme of lubrication naturally defended this design and considered it superior to the splash system. Finley R. Porter's paper outlined the advantages of the various systems of lubrication and this fitted in well with the discussion of Mr. Round's paper.

The members of the Metropolitan Section who attended the annual outing to West Point on Sept. 16 acclaimed in accord that they had an ideal trip. Everyone was curious to know how Mr. Bergmann bribed the weatherman for such perfect weather. Colonel Mettler supervised the inspection of the various buildings of the academy, after which dinner was served at the West Point Hotel. After dinner Mr. Bergmann introduced the new chairman of the Section, W. E. Kemp, who in turn introduced the honor guest of the day, President B. B. Bachman. The last speaker, Colonel Mettler, gave a very interesting address, outlining the work of West Point and stressing the democratic spirit that prevails there.

Indiana Section

The first meeting of the Indiana Section for the season will be held on Oct. 12. C. F. Kettering of the General Motors Research Corporation, Dayton, will present his predictions on the Trend of Research. The organization headed by Mr. Kettering is the largest of its kind devoted exclusively to commercial and scientific research along automotive lines. It is only natural that this association, coupled with Mr. Kettering's unusual ability to sense the developments of the future, should guarantee the presentation of thought deserving the attention of every member of the Society. There are few men more appreciative of the commercial value of research than Mr. Kettering. He has been a strong advocate of research ever since his first venture in the automotive industry. Several prominent engineers have been invited to discuss Mr. Kettering's predictions and an interesting meeting seems assured.

Minneapolis Section

The Minneapolis Section will open the season with a technical meeting on Oct. 4 at the Manufacturers Club. N. S. Kingsley will present a paper on the Manufacture of Petroleum Products for Automotive Uses. As the subject is very timely, a number of prominent engineers are expected to be present to discuss the important points of this paper.

New England Section

O. J. Rohde, vice-president of the Wire Wheel Corporation of America, Buffalo, will present a paper on Wire Wheels and Disc Wheels at a meeting of the New England Section on Oct. 27. Mr. Rohde has been associated with the wire wheel building industry for many years. He has made many exacting tests of wood, wire and steel disc wheels with a view to comparing their relative strength and effect upon tire wear. His talk will be illustrated with lantern slides and motion pictures. Opinion as to the most desirable type of wheel for automotive vehicles is at wide variance. Much of the discussion has been controversial but it is hoped that the quantitative data to be given by Mr. Rohde will clarify matters to a considerable degree.

L. B. Ehrlich gave an extremely interesting paper on automotive starting and lighting systems before the New England Section on Sept. 22. The meeting was attended by many of the electrical service-men in Boston and those in attendance were rewarded with an interesting outline of starting motor and generator characteristics. Mr. Ehrlich believed that engine designers should pay greater attention to the proper location of these electical units so that they would be more accessible for service purposes.

Pennsylvania Section

The first fall meeting of the Pennsylvania Section will be held in the rooms of the Engineers Club on Thursday evening, Oct. 26. The topic of the evening will be the manufacture of automobile bodies, particularly those of steel construction. It is expected that the address of the evening will be illustrated with slides showing bodies in the course of construction.

Washington Section

The Washington Section will start the Section year with a meeting on Oct. 5 at the Cosmos Club. R. E. Carlson of the Bureau of Standards staff, will present a paper on A Road Test Method for Comparing Motor-Fuels. Mr. Carlson is engaged in the conduct of the road tests now being run at the Bureau of Standards in connection with the Society's Fuel Volatility Research project. He will describe the methods and apparatus being employed in this work and give some idea of the conclusions that may be deduced from the data recorded. Mr. Carlson is associated with W. S. James, whose paper on road performance-testing aroused much favorable comment at the Summer Meeting. Members of the Society from other districts will be welcomed at this meeting, for it is recognized that the interest in the work at the Bureau is very widespread. The meeting will start at 8 o'clock.

Mid-West Section

The Mid-West Section meets for the first time this season on Oct. 20, in the rooms of the Western Society of Engineers in the Monadnock Building, Chicago. The meeeting will start at 7:30 o'clock. Here again we find our good friend, C. F. Kettering, announced as the speaker of the evening. Needless to say, Mr. Kettering's unusual presentation of any subject results in an incessant demand upon his speaking time. The members in the Mid-West district have not heard from him in some time and they prevailed upon him to open their series of Section meetings. The mechanical, civil and electrical engineers have been invited by the Section to participate in this meeting which assumes the character of a joint assembly.

TAP DRILL SIZES

A MEETING of the National Screw Thread Commission was held at the Bureau of Standards, at which consideration was given to the question of tap drill sizes to be recommended for producing holes in conformity with standards adopted by the Commission. It was voted to recommend drills of a diameter midway between the maximum and minimum minor diameters of the tapped holes already established. These limits result in a thread in the nut from 75 to 83⅓ per cent of the full depth, with a mean value of 79⅙ per cent of the full-thread depth. The recommended sizes are to be considered further with reference to a series of decimal wire gage sizes that is also under consideration.

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Aug. 10 and Sept. 9, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ALVUT, EDMOND JONES (E S) student, Rensselaer Polytechnic Institute, Troy, N. Y., (mail) 1116 Elm Street, *Utica, N. Y.*

ANDERSON, KENNETH B. (E S) student, California Institute of Technology, *Pasadena, Cal.*, (mail) 59 South Wilson Avenue.

BARNES, ORRIN HAYWARD (E S) student, California Institute of Technology, Pasadena, Cal., (mail) 910 East Harvard Street, *Glendale, Cal.*

BARTHOLOMEW, J. R. (J) assistant engineer, Westinghouse Pacific Coast Brake Co., Emeryville, Cal., (mail) 2612 Durant Avenue, *Berkeley, Cal.*

BLAKELEY, LOREN E. (E S) student, California Institute of Technology, Pasadena, Cal., (mail) 2013 East Florence Avenue, *Los Angeles, Cal.*

BORDEN, EDWARD ROY (M) mechanical engineer, Mudge & Co., 4425 West 16th Street, *Chicago.*

BREAKEY, EDWIN T. (J) factory superintendent, Columbia Carbureter Co., *Chicago*, (mail) 7024 Wentworth Avenue.

BREWER, HENRY (A) district sales manager, Leeds & Northrup Co., Philadelphia, (mail) 53 West Jackson Boulevard, *Chicago.*

BUELL, ROY D. (A) president, Buell Mfg. Co., *Chicago*, (mail) 2975 Cottage Grove Avenue.

CATE, GARTH W. (A) general manager, Flexo-Motive Corporation, *Chicago*, (mail) 551 McCormick Building.

DAVIS, FRED (A) engineering tests, General Motors Research Corporation, *Dayton, Ohio.*

DEMELTO, VINCENT MARION (E S) student, Ohio State University, Columbus, Ohio, (mail) 1614 Lake Front Avenue, *East Cleveland, Ohio.*

DOTY, E. M. (J) mechanical engineer, Denby Motor Truck Co., *Detroit*, (mail) 404 Lenox Avenue.

FAIRCHILD, SHERMAN M. (J) president, Fairchild Aerial Camera Corporation, New York City, (mail) 318 Main Street, *Oneonta, N. Y.*

GABBER, JACOB E. (E S) student, Ohio State University, Columbus, Ohio, (mail) 22 South Horton Street, *Dayton, Ohio.*

HANNAH, WALTER JOHN (F M) consulting engineer, Hannah, King & Co., *Glasgow, Scotland*, (mail) 209 Hope Street.

HAWLEY, B. [illegible]. (M) mechanical engineer, Holt Power Light Co., 4196 Bellevue Avenue, *Detroit.*

JACKSON, H. GARDNER (A) assistant general manager and sales manager, Wire Wheel Corporation of America, 1700 Elmwood Avenue, *Buffalo.*

JOHNSON, WILLIAM G. (J) 708 Sterling Avenue, *Joliet, Ill.*

KING, TOWER W. (J) airplane materials testing engineer, engineering division, Air Service, McCook Field, Dayton, Ohio, (mail) care L-W-F Engineering Co., *College Point, N. Y.*

KNERR, HORACE C. (S M) metallurgist, Navy Yard, *Philadelphia*, (mail) Naval Aircraft Factory.

KNIGHT, RALPH B. (A) superintendent of inspection, North East Electric Co., Whitney Street, *Rochester, N. Y.*

KOLB, GEORGE F. (A) manager of motorcycle division, Bullard Machine Co., Bridgeport, Conn., (mail) *Fairfield, Conn.*

LEWIS, MAJOR BURTON O. (S M) Office Chief of Ordnance, Ordnance Department, Room 3702, Munitions Building, *City of Washington.*

LUTZENBERGER, LOUIS D. (E S) 139½ South William Street, *Dayton, Ohio.*

MARSH, HALLAN N. (E S) student, California Institute of Technology, Pasadena, Cal., (mail) 1225 West Brookes Avenue, *San Diego, Cal.*

MILLER, H. S. (A) general manager, Plowman Tractor Co., *Waterloo, Iowa.*

PALMER, H. A. (M) chief engineer, Reynolds Spring Co., *Jackson, Mich.*, (mail) 1702 Francis Street.

PARROTT, R. B. (A) secretary and treasurer, Service Products Corporation, 201 South Rural Street, *Indianapolis.*

RICARDO, HARRY R. (F M) consulting engineer, Ricardo & Co., Ltd., Bridge Works, Shoreham, Sussex, England, (mail) *Penstone, Lancing, Sussex, England.*

ROOT, RALPH C. (M) engineer and sales manager, Service Products Corporation, 201 South Rural Street, *Indianapolis.*

ROUSE, GEORGE ALAN (J) assistant supervisor of motor vehicles, Standard Oil Co. of New Jersey, *Baltimore, Md.*, (mail) Y. M. C. A.

SAVANT, A. K. (J) student, University of Michigan, Ann Arbor, Mich., (mail) 2739 Second Boulevard, *Detroit.*

VAUGHAN, SAMUEL E. (E S) student, Leland Stanford, Jr., University, *Stanford University, Cal.*

WHITEHEAD, C. I. (A) P. O. Box 7187, *Johannesburg, South Africa.*

WITMER, GEORGE (A) branch manager, Sterling Motor Truck Co., Brooklyn, N. Y., (mail) 489 West 130th Street, *New York City.*

Applicants *for* Membership

The applications for membership received between Aug. 15 and Sept. 15, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

AIKEN, FRANK, ignition engineer, Atwater Kent Mfg. Co., *Philadelphia.*

BELL, JAMES L., general superintendent, Fulton Motors Corporation, *Farmingdale, N. Y.*

BOCA, JOSEPH, body engineer, Pierce-Arrow Motor Car Co., *Buffalo.*

BRIDWELL, J. W., chief engineer, Ruggles Motor Truck Co., *Saginaw, Mich.*

CAMPBELL, WILLIAM B., sales engineer, Nordyke & Marmon Co., *Indianapolis.*

CARLSON, RAYMOND A., draftsman, Rockford Drilling Machine Co., *Rockford, Ill.*

CARSON, RAY EDGAR, draftsman, Robert H. Hassler, Inc., *Indianapolis.*

DEN TEX, NICHOLAAS J., designer, Gallaudet Aircraft Corporation, *East Greenwich, R. I.*

EVENBURGH, R. L., designer, Advance-Rumely Co., *La Porte, Ind.*

FLOGAUS, HOWARD A., power transmission engineer, Willys-Overland Co., *Toledo.*

FOISY, GEORGE A., designing engineer, U. S. Cartridge Co., *Lowell, Mass.*

GAINES, WALTER S., assistant engineer, Rubay Co., *Cleveland.*

GERDES, HARRY, instructor in automobile mechanics, El Paso Public Schools, *El Paso, Tex.*

GILBERT, H. HOYT, research engineer, Cadillac Motor Car Co., *Detroit.*

HALL, G. P., Central district manager of manufacturers' sales, Westinghouse Union Battery Co., *Swissvale, Pa.*

HANNON, BOYD E., automotive engineer, Texas Co., *Chicago.*

HAYLETT, R. E., assistant manager, manufacturing department, Union Oil Co. of California, *Los Angeles, Cal.*

HECKMAN, J. A., treasurer, Heckman Signal Co., *Denver, Col.*

IRELAND, MAJOR MARK L., Quartermasters Corps, Massachusetts Institute of Technology, *Cambridge, Mass.*

KENNEDY, CAPT. GRAFTON S., Ordnance Department, *Aberdeen Proving Ground, Md.*

KIZER, H. W., automotive engineer, Texas Co., *Chicago.*

KREBS, HENRY, engineer of tests, Willys-Overland Co., *Toledo.*

LEA, R. W., executive vice-president and general manager, Stevens Motor Car Co., *Moline, Ill.*

MCQUAID, HARRY W., vice-president and chief engineer, National Pressed Gear Co., *Canton, Ohio.*

MARTY, MATHEW J., engineer, Chicago Mfg. Co., *Chicago.*

MILWARD, W. F., chief engineer, W. S. Laycock, Ltd., Millhouses, *Sheffield, Eng.*

OTT, T. F., superintendent lubricating division, manufacturing department, Union Oil Co. of California, *Los Angeles, Cal.*

PEASE, FRANK D., laboratory engineer, Studebaker Corporation of America, *Detroit.*

PORTER, ROLLAND COURT, designer, Prudden plant, Motor Wheel Corporation, *Lansing, Mich.*

PRICE, CHARLES S., chief engineer, Bethlehem Spark Plug Co., *Bethlehem, Pa.*

RAGSDALE, EDWARD T., draftsman, Pierce-Arrow Motor Car Co., *Buffalo.*

REDMANN, W. F. O., supervisor of motor vehicles, Municipal Garage, *Dayton, Ohio.*

SANFORD, ROY S., Sanford Fore-Wheel Brake Co., *Pasadena, Cal.*

WALLACE, DONALD E., experimental engineer, Shakespeare Co., *Kalamazoo, Mich.*

WATTS, WILLIAM SEWELL, student, Coyne Trade and Engineering School, *Chicago.*

WRIGHT, W. C., manager main line branch, John Warren Watson Co., *Philadelphia.*

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XI November, 1922 No. 5

Chronicle and Comment

National Screw Thread Commission

GEORGE S. CASE has been appointed by Secretary of Commerce Hoover a member of the National Screw Thread Commission, in place of Edwin H. Ehrman whose resignation was accepted with regret. The Act of Congress that established the Commission provided that the Society should nominate two members thereof. Earle Buckingham, appointed by the Secretary of Commerce under nomination by the Society, is also serving as a member of the Commission.

Part 1, 1921 Transactions

COPIES of Part 1 of Volume XVI of the TRANSACTIONS are being distributed by parcel post this month to the members who have ordered them. The issue contains about 48 papers and reports, together with the discussion thereon, presented at the 1921 Annual Meeting of the Society and at various meetings of the Sections. The volume is constituted of over 800 printed pages. Less than 1500 members ordered copies of this Part of the TRANSACTIONS. There was no charge to the members for this volume in addition to payment of annual dues. This will likewise be the case with regard to the three next forthcoming Parts, namely those for the last half year of 1921 and for the two periods of 1922.

Gear Noise

GEAR noise persists as one of the industry's annoying problems in spite of continued study of gear mechanism over a long period of years. Basic tooth-form, pitch, pressure angle and addendum have been varied without material progress; non-metallic materials have been used with some success, but there is still room for improvement from the standpoint of decreased wear and noise. Two papers on gearing were presented at the Production Meeting of the Society in Detroit and both are printed in this issue of THE JOURNAL. K. L. Herrmann's paper directs attention to appreciable errors in form and spacing that are common in gear teeth cut by our present-day machine tools and methods. The observations given represent the findings of a searching investigation into the causes of gear noise by the Studebaker production organization. The paper by Messrs. Crain and Brodie records some rather unusual discoveries made by Packard production engineers who conclude that the mounting of gears is an important factor in the reduction of noise. Both papers deserve most careful study by automotive designing and production engineers.

The Production Meeting

PRIOR to the Production Meeting of the Society that was held in the General Motors Building at Detroit last month, one of the men who has been very prominent in the automotive industry for a number of years said

> I am quite willing to concede that I know something about production, and believe that I can originate ways and means to produce an article at as low cost as anyone else, yet I do not possess the ability to transmit that knowledge to an organization such as the Society of Automotive Engineers. I would be delighted if I could sit in as an interested listener at such a meeting as you are going to hold.
>
> I would also like awfully well to be able to make the trip through the various Detroit plants enumerated. It would be an education in itself.

Though unduly modest as to his ability to impart knowledge to others, this man foretold in a strikingly accurate manner what the Production Meeting proved to be. The taking up of production matters at a national meeting of the Society was a distinct success. It was a thing for which there was much need.

New Data Sheets

DATA sheets incorporating the Standards and Recommended Practices last approved by the Society, following the White Sulphur Springs meeting held in June, have been mailed to the members. These 44 sheets, dated August, 1922, present new or superseding matter. They are *for use* in connection with current and new designs. They are the result of able conscientious work by many members. They are not only worthwhile but an incomparable source of essential information for designers and their associates in the automotive field.

Instructions as to assembling the new sheets in the S.A.E. HANDBOOK accompany this last shipment of Standards information to the members. An important feature of the service is the readiness of the Society to furnish

without charge to members copies of sheets that are missing from but should be contained in their S.A.E. HANDBOOKS. A check-list of all live data sheets has been mailed them in this connection. Keep your S.A.E. HANDBOOK uptodate. Are you getting the benefit of the use of the S.A.E. HANDBOOK in your work? Are your employers, and in turn the users of your product getting the benefit of it?

Fuel Research Tests

AT a meeting of the steering committee held in the City of Washington at the Bureau of Standards on Sept. 20 and attended by representatives of the automotive and the petroleum industries and of the Bureau, the results already obtained from the cooperative fuel-research program were reviewed and examined critically. A vote taken on the advisability of continuing the tests during the winter was passed in the affirmative.

It is gratifying to note that only about one-half of the sum allotted by the industries for this work has been expended uptodate, and that the program originally provided for in the estimates is almost complete.

Automotive Engineer and Highway Engineer

THE automotive engineer needs data of rolling and air resistance rather than data of gasoline consumption from road tests, since the latter depend upon the individual characteristics of the vehicle used, the carbureter and the habits of the driver. Having the power factors, the automotive engineer will design the plant to furnish the power required.

From the standpoint of the highway engineer, a problem like the following is set:

A reduction in grade of 4 ft. will reduce the amount of energy required 8000 ft-lb. What will this be worth in the saving of gasoline and other operating expense for the traffic considered? Will the solution be for cars as built at present, or for cars using the roads 15 years from now? The mutual relation of the highway and the vehicle is expressed in the effect of low grades on changes in the engine design and the power requirements of the car. Inasmuch as the large expense of highway transportation is in the expense of operation of rolling stock rather than in fixed charges and maintenance of the road, the investigation of the costs of operation of traffic is particularly needed, it was felt at a Conference on Tractive Resistance of Roads held under the auspices of the Advisory Board on Highway Research of the National Research Council.

It is considered that a 5-per cent grade on a hard-surfaced road is not excessive for passenger cars and 2 to 3 per cent for large-capacity trucks. The fuel cost and the time required, as affected by the type and design of the surfaces and grades, are measurable by experimental methods, and the depreciation of the vehicle and other maintenance costs should be evaluated from a study of the statistics of motor vehicles obtained in actual operation.

Recording Car Road-Performance

ONE of the most important accomplishments in research is the perfection of instruments and appliances that makes it possible to observe and record conveniently and accurately just exactly what we want to know. Sometimes it is even harder to find out exactly what to record than to devise a means for recording it.

One of the best recent examples of a success in this line is furnished by the very ingenious combination instrument for recording almost all one needs to know concerning the performance of passenger cars on the road that was shown by W. S. James at the Summer Meeting. A complete description of this outfit will be published in THE JOURNAL in the near future.

Briefly, this recorder places on a road vehicle practically all the facilities of the usual dynamometer laboratory and several others besides. Moreover, the records are entirely automatic and yet in most of the measurements they have all the precision commonly obtained on the test stand. It should go far to settle the ancient quarrel between the exponents of laboratory tests and of road tests by putting the laboratory on the road.

The apparatus consists of a photographic recorder, which records the rate of fuel consumption, and another instrument with recording pens that gives on a single strip of paper a record of the following quantities:

(1) Engine speed
(2) Manifold depression
(3) Temperature of inlet water
(4) Temperature of outlet water
(5) Temperature of oil
(6) Temperature of the differential
(7) Temperature of air at the entrance to the carbureter
(8) Temperature of the atmospheric air
(9) Weight of air being used by the engine
(10) Acceleration of the car
(11) Air speed
(12) Apparent direction of the wind

Grade Transfer of Members

ON p. 127 of THE JOURNAL for August a comprehensive statement was given of the policy of the Society in the grading of applicants for membership. The Council may transfer a member from one grade to another, upon written request from him and the presentation of evidence that he is qualified for transfer to the other grade. The most usual cases are of course the applications for transfer from Junior or Associate to Member grade. There are no limitations of age for Associate grade. The Junior grade is open to those who are under 30 years of age and engaged in technical work. A Junior may on reaching the age of 26 be transferred to Member or Associate grade; upon reaching the age of 30 he may not retain Junior membership, but must be transferred to one of the grades mentioned. Upon transfer, he is required to pay the difference in initiation fee of the grades.

An Associateship is not as such *inferior* to a Membership, although the latter only carries voting and office-holding power in the Society. The fundamental difference is that of line of work or qualification. Associate grade is composed of those who are engaged in the automotive or related industries in responsible commercial, financial or manufacturing capacity. A principal feature is qualification to cooperate with automotive engineers.

Associates or Juniors who feel that they are entitled to Member grade should transmit written application therefor to the office of the Society, stating what work they have done since the time of their election that would in their opinion justify the transfer, and giving the names of three members as references who know of this work. Such applications are given careful attention by a committee appointed by the Council, the latter in turn giving due consideration to the written records and the committee recommendations.

The Production Meeting

THE unusual degree of success attained by the Production Meeting in Detroit, Oct. 26 and 27, is attested best by the size and character of the attendance. Close to 600 persons registered at the meetings, factory visits and the Dinner. They came from all corners of the Country, some from Minnesota and Wisconsin, others from Kentucky, Iowa and Massachusetts. Production managers, factory superintendents, tool supervisors, time-study men, chief inspectors and others of similar high rank constituted the major portion of the attendance. It is difficult to imagine how a meeting could have been more representative of a single branch of an industry.

The Production Dinner attracted close to 400 members and guests. The two professional sessions included audiences that averaged 350 persons and at times passed the 400 mark. This is a record for professional meetings of the Society, either sectional or national. As tangible approval of the Society's purpose to enter the production field actively, these figures and the attentiveness and enthusiasm of those present could hardly be surpassed. The Production Meeting is established as a Society institution without question.

Morning Sessions

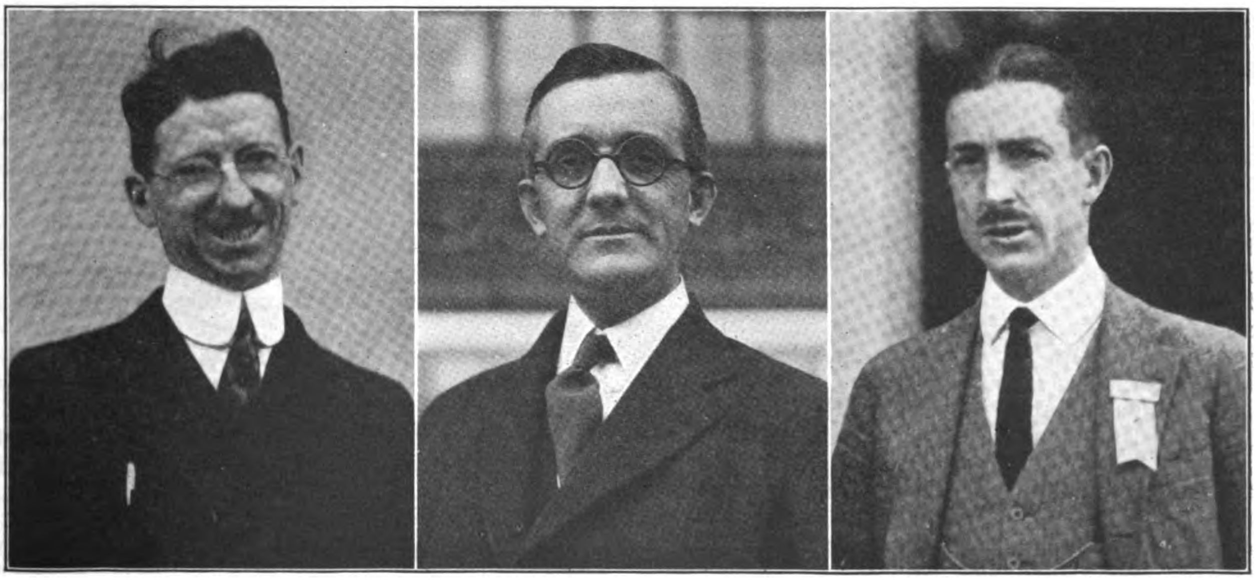
The Thursday morning session was convened at 9:43. H. W. Alden, as presiding officer, observed that the builders of the Society Constitution had proceeded wisely in providing for production a place in our activities equal to that of design. The day of experiment and design has of course not passed; these are more important than ever before. But production problems must have more and more attention. The engineer must know about production. The production man must know something of engineering.

E. Karl Wennerlund, who presented the first paper of this session, prefaced his able oral outline of it by giving a sketch of the stages of industrial development which preceded the present status of wage incentives; beginning at the time when foremen set piece prices based on their judgment. There was little method of checking quantities of articles produced until a few years ago. Even to-day probably 50 per cent of industry does not have this quantity check. The coupon system of checking production became impractical in factories in which thousands of operations were going on. To-day the system of numbering lots serially is generally used. An effort is being made to depart from this practice which is too unwieldy in large production in which several thousand men are employed; it being necessary commonly to check 15,000 tickets daily.

Mr. Wennerlund believes that the group-bonus wage-incentive plan, which is described in his paper printed in this issue of THE JOURNAL, is the best system for application in the automotive industry; having in mind the use of the progressive system of manufacture, in which, generally speaking, the work in process does not touch the floor. He is of the opinion that it combines the merits of the piece rate so far as incentive is concerned and of the day wage in respect to flexibility.

The paper by F. A. Mance, on the Control of Operating Tool and Supply Cost, which is printed in this issue of THE JOURNAL, dealt briefly with means taken to regulate the number of tools and amounts of supplies used in car-building factories. Reference was had to a system that has been in effect in the Studebaker plant for 4 years, with a resultant decreased cost of renewals of perishable tools. In connection with studies of the life of perishable tools, some data with regard to drills were given.

In a paper by James A. Ford on Processing Spline Shafts by a New Method, the tools used in overcoming the difficulty in hobbing to exact dimensions were described; these being a die constructed in a manner similar to that followed in the fabrication of automatic threading-dies, the cutters being in the position that the



K. L. Herrmann Thomas J. Litle, Jr. K. K. Hoagg
THREE MEN WHO WERE LARGELY RESPONSIBLE FOR THE SUCCESS OF THE MEETING

die chasers ordinarily occupy; and a cam ring of strong construction. The methods employed to obviate warpage during the hardening process and in re-centering shafts were also set forth. The paper appears on another page of this issue.

The paper by P. E. Haglund and I. B. Scofield presented the fundamental reasoning underlying the creation and operation of the Ford River Rouge Plant. Basic raw materials, coal, iron ore and limestone, are brought to this great industrial unit at minimum cost, conveyed mechanically to coke-oven, blast-furnace and foundry, all by-products being utilized at each step and industrial waste practically eliminated. Coke, combustible gases, tar, chemicals for fertilizer and benzol are used in plant operation or sold commercially. Blast-furnace gas is burned in the powerhouse. Slag will be used eventually in cement manufacture. Molten blast-furnace iron is conveyed direct to the foundry, mixed with cupola iron and poured, so that the heat generally wasted in casting pigs is conserved. Sand from the foundry molds returns automatically to the molders while still possessed of some of its heat from the previous casting.

Molds are assembled, cores made and set, castings poured and cooled, all on a conveyor system. Cylinder blocks are conveyed mechanically through shake-out, cleaning and inspection departments direct to a compact machining department where they are machined without a pause or return to the floor and passed out to the finished-stock loading platform. Truly, here we find American production genius at its height. This paper, fully illustrated, will appear in the December issue of THE JOURNAL.

President Bachman presided at the Friday morning session. H. P. Harrison presented the paper on some phases of construction of the Franklin automobile, this being entitled Some Unique Features of Automobile Construction, which is printed on p. 398. The production of air-cooled engine cylinders, crankshafts with case-hardened bearings, forged aluminum-alloy connecting-rods and hot-swaged rear-axle drive-shafts were discussed in turn.

L. C. Hill presented the contribution by H. J. Crain and J. Brodie, of the Packard company, under the title, Experience Notes from a Production Notebook. In this interesting narrative brief descriptions were given of machinery devised for the purpose of running-in brake-band assemblies and of tools for centering and driving pistons during external-grinding operations. In addition, points involved in the discovery of the causes of obscure noises in engines and gear oil-pumps were given, and in particular an account of the study the Packard company has been making of gear noises in general, especially in connection with transmissions and rear axles. The paper is printed in this issue of THE JOURNAL.

Many matters relating to gears, including noises, tooth spacing, variations in finished product, index errors and tooth-forms, as well as cutting machines, hardening and grinding, were treated by K. L. Herrmann in his paper, Some Causes of Gear-Tooth Errors and Their Detection, which is printed on p. 391. His purpose was to show that production variables have a much greater influence on gear sounds than pressure-angles, steel or tooth-forms. In this connection he sought the assistance of gear-cutting tool and machine designers. He urged strongly the importance of this, pointing to the large apparently unnecessary expense in the production of gears, and also expense involved after they are "produced."

It was evident not only that Mr. Herrmann had a comprehensive grasp of his subject, but that he had taken great pains to make clear essential matters involved in improving the conditions of the quantity production of gears. Supplementing his paper, he projected some views of gears in operation to illustrate tooth-spacing errors, interferences and other undue variations. It is earnestly hoped that Mr. Herrmann's efforts will stimulate incisive and clarifying discussion at future meetings of the Society and the Sections.

Two papers on the Friday program were the occasion of a most interesting incident in that two production authorities reached identical conclusions independently on the important matter of selecting machine-tools. A. J. Baker and R. K. Mitchell, of the Willys-Overland and the Maxwell plants, respectively, read the papers in question. They both believed that the efficiency of the specially designed and highly productive machine-tool has been greatly exaggerated and predicted a movement toward wider use of standard machine-tools that can be adapted to various classes of work by the employment of simple and inexpensive fixtures and attachments. Mr. Mitchell said that the most forceful argument against special machine-tools is the unstable design and continuous development of the automotive parts themselves. He recommended a more general use of the three fundamentals of jig and fixture work: the clamp, the V-block and the angle-plate.

Mr. Baker's conclusions were as follows:

(1) There is a surplusage of machine-tool equipment of the standard types, both actual and potential
(2) Machine-tool builders are devoting their thought to high-production single-purpose machines of standardized types
(3) The craze for special machinery is passing
(4) Special machinery will not always stand a financial comparison with standard machinery
(5) We are not, as an industry, facing our responsibilities in the matter of training operative help for tool and die help
(6) We cannot disregard the inventory value of existing equipment and the loss that will be shown on our balance-sheet by its conversion from productive machinery to excess machinery for sale when considering new equipment
(7) The only good reason for installing new machinery, old machinery or any machinery, apart from those cases where higher quality is demanded, is to reduce cost

FACTORY VISITS

The inspection trips through representative Detroit factories were a much appreciated feature of the meeting. Over 250 production men participated in the visit to the Ford River Rouge plant on Thursday afternoon. The party was taken through the by-products building where the reclamation of tar, benzol and ammonium-sulphate from the coking process was observed. The coke-ovens, blast-furnaces, foundry and machine-shops were all seen in operation. Sections of the tractor assembly, stamping and body plants were inspected. The visit was concluded with a trip through the central powerplant which has an interesting boiler equipment that utilizes all manner of by-products in its novel combustion system. The members were impressed by the extensive use of mechanical conveyors in all departments, the elimination of waste material and labor and the relatively small number of men required to operate this huge project.

Three simultaneous visits were made Friday afternoon to the Packard, the Dodge and the Cadillac factories. Approximately 60 members visited each of the plants. The party visiting the Cadillac factory was shown through the engine, clutch, transmission and chassis-assembly de-

A. B. C. Hardy	I. B. Scofield	H. W. Alden
A. J. Baker	E. K. Wennerlund	P. E. Haglund
H. P. Harrison	R. K. Mitchell	B. B. Bachman

AUTHORS AND PROMINENT SPEAKERS AT THE PRODUCTION MEETING

partments. Cadillac assembly methods are unusual in that the mechanical or gravity conveyor is not used extensively. Attention is given to the completion of a number of bench assemblies that are taken later direct to one general assembly group and distributed to the men. The men are moved rather than the work in making the progressive major assemblies. The party inspected the gear-cutting department and saw a group of new gear-grinding machines in operation. A trip through the heat-treating plant and an inspection of the fender enamelling system was included in the Cadillac visit.

The members who visited the Dodge factory were impressed particularly by the number of interesting stampings produced in the press-room. The body and fender enamelling system, which is the largest of its kind, was explained to the visitors. The members were shown parts of the engine-assembly line, the chassis-assembly and some of the more interesting machine operations. The efficient use of floor space to increase the productivity per square foot was one of the outstanding impressions left by the Dodge visit. It was also noticed that careful preparation of the steel surfaces before enamelling is largely responsible for the excellent finish given the enamelled body and parts.

The Packard visitors were shown through all of the major departments of the passenger-car division with the exception of the small-parts machine shops. The foundry, forge and heat-treating plants were inspected and the party taken through the cylinder and crankcase machining departments. Passing through the engine block-test room, the members were taken down the long conveyor line on which the chassis assembly is made. A group of men especially interested in truck construction were taken through the Packard truck shops.

The production staffs of the factories visited deserve special recognition for their hospitality and the efficient manner in which they conducted the visits. Inquiries were always answered with specific information, the unconventional practices of each factory were explained unselfishly and lack of time alone prevented the most scrutinizing type of inspection of each factory.

The Dinner

It is not too much to say that the Society never held a more interesting event, or one more important to the automotive industry, than the Production Dinner. President Bachman, in introducing Harold H. Emmons, the toastmaster, iterated emphatically that there is no essential difference in the work of those engaged in manufacturing and of those whose vocation is research or design. Moreover, that the man who feels that he in his own particular work is self-sufficient and not intimately connected with others in selling and keeping sold the articles on which our industry is founded, is sadly lacking in vision.

Pierre S. duPont, president of the General Motors Corporation, held the diners intently with very felicitous and forceful remarks. He said in part:

> If we work together in the industry, its problems will be easily solved. If we are not fearful of its problems, they will be still more easily solved.
>
> We are fortunate to be engaged in an industry that is an infant. The world is before us now; there is not only the realm of our own industry and its many ramifications and possibilities, but thousands of uses for the automobile that we do not picture to-day. We have only scratched the industry. This is the only country in the world that uses the automobile, practically. Yet the bulk of the population is elsewhere, with its own problems, peculiar if you will, but all to be solved; before you to be solved.

A. B. C. Hardy, president and general manager of the Olds Motor Works, was the principal speaker of the evening, his topic being "The Production Man's Place in Our Industry." Mr. Emmons said that Mr. Hardy presented the best analysis of this subject ever made.

Mr. Hardy pointed out that the production or manufacturing division of an automobile plant occupies from 80 to 90 per cent of the land acreage and from 70 to 80 per cent of the floor space involved, and directs the work of from 5 to 10 or even 15 times as many people as are required by all the other divisions of the plant combined. He listed in a suggestive way the things the production man must know, and spoke of how he must be schooled. He said, among other things

> No one accomplishes anything alone in our complicated fast-moving industry.
>
> The production man, if he be a wise man, is always on the lookout in his own plant, and also takes a side-look into other plants.
>
> The production man and his department heads are the means of conveying to the workers the standards and character of the company. And the workers judge the company accordingly.
>
> The real bosses of all our jobs are the motor-using public.
>
> I have seen some poor designing turned into fairly good cars by production work and refining.

By direction of Toastmaster Emmons Past-President Kettering gave the benediction. The benediction included the following remarks:

> When we engineers design a thing, and the production man starts to make it and puts it together, it is never what we designed.
>
> The time for the production engineer to "sit in" is before the thing is designed. The industry has spent hundreds of millions of dollars foolishly by not calling in the production man when the design was in process.
>
> The two closest fellows in any organization should be the production man and the engineer. A third man, who has never been there before, should come into the picture—the accounting man.
>
> This meeting marks an epoch in the history of the automobile engineer.

Committeemen and Co-Workers

The story of the Production Meeting would not be complete without mention of those whose efforts during the past three months contributed to its success. K. L. Herrmann and T. J. Litle gave much of their personal time in the solicitation and selection of the valuable papers. K. K. Hoagg not only arranged for the several factory visits but secured the transportation equipment and directed the efficient loading and dispatch of the vehicles. W. C. Keys, and the conscientious members of the Reception Committee, W. R. Strickland, Frank Ruoff, C. H. Brennan, A. L. Kimball and J. W. Spray, deserve thanks for undertaking and carrying out many detail responsibilities in a manner praised by all. E. F. Roberts, A. U. Widman, H. A. Coffin, George E. Goddard and C. E. Sorensen were the genial hosts who opened the doors of the factories to the members. The authors and speakers deserve much commendation, although they were amply repaid by the enthusiastic reception accorded their respective contributions. Acknowledgment is made to the General Motors Building Corporation for the provision of quarters for the meeting which were excellent in accommodations and facilities. The Detroit Section of the Society is thanked for the cooperation of its officers and

(Concluded on p. 408)

Some Causes of Gear-Tooth Errors and Their Detection

By K. L. Herrmann[1]

Detroit Production Meeting Paper *Illustrated with* Charts and Drawings

THE different gear noises are classified under the names of knock, rattle, growl, hum and sing, and these are discussed at some length, examples of defects that cause noise being given and a device for checking tooth spacing being illustrated and described. An instrument for analyzing tooth-forms that produce these different noises is illustrated and described.

Causes of the errors in gears may be in the hardening process, in the cutting machines or in the cutters. A hobbing machine is used as an example and its possibilities for error are commented upon. Tooth-forms are illustrated and treated briefly, and the hardening of gears and the grinding of gear-tooth forms are given similar attention.

MOTOR-CAR production-men, as a rule, do not lay claim to being specialists in all the various arts and sciences that enter into the finished motor-car; so, in the matter of gears, we do not pretend to know all the details that enter into their design and production. Very few of us are familiar with the mathematics relative to the involute gear-tooth forms that the engineering fraternity stresses considerably in connection with gear noise. We are much more familiar with the noises of the finished product, which are recognized as knock, rattle, growl, hum and sing. When these occur, naturally those men who have made a life study of gear subjects are called in, and they make recommendations.

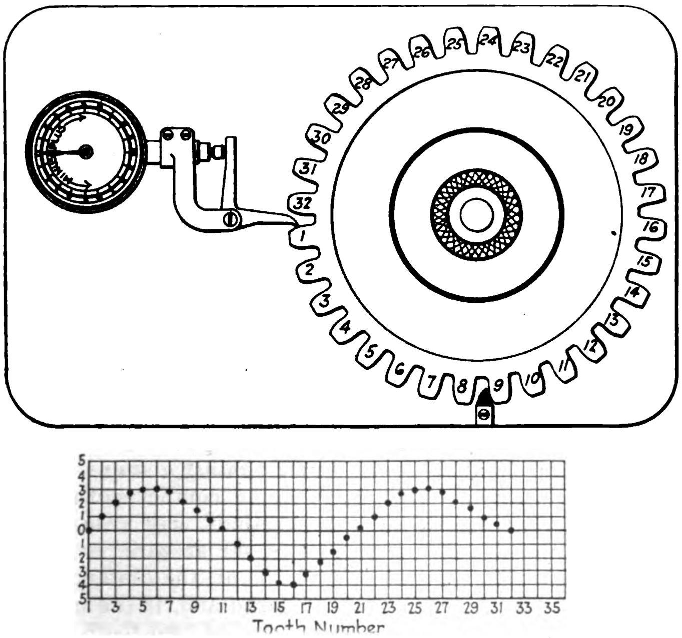


Fig. 1—Device Used for Checking Tooth Spacing and Underneath a Chart Showing the Results Obtained

As a rule our gear experts offer widely varying remedies for the same cause. If one is using a 20-deg. pressure-angle, we can advise a 14½-deg. angle and have a large number of supporters in both the engineering and the production fields. If both 20 and 14½-deg. angles have been tried, we can easily advise a stub tooth or full length. If all six combinations have been tried, it is easy to recommend topping-off or no topping-off; and, if this does not work, we recommend a different *steel, cutting machine* or *cutter*. While all of this goes on, sufficient time has elapsed to start all over, with the first recommendation. Of all of these recommendations we find many past and present supporters of national reputation.

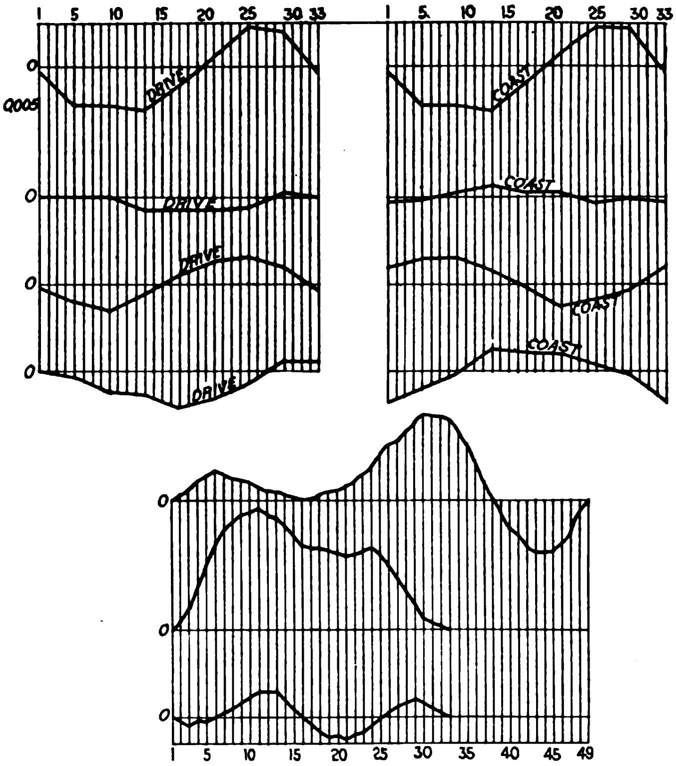


Fig. 2—Chart of the Variations in a Finished Gear

It is the purpose of this paper to show that production variables have a much greater influence on gear sounds than changing pressure-angles, steel or tooth-form details; also, by showing the errors present, to obtain definite help from the gear-cutting tool and the machine designer. We will confine this discussion, for the time being, to the transmission, which is the simplest type of gearing used in the motor car.

Gear Noises

We have already referred to the various kinds of gear noise. The first is a knock that might be caused by a nicked tooth or a single tooth of a pair of running gears

[1] M.S.A.E.—Engineer, Studebaker Corporation, Detroit.

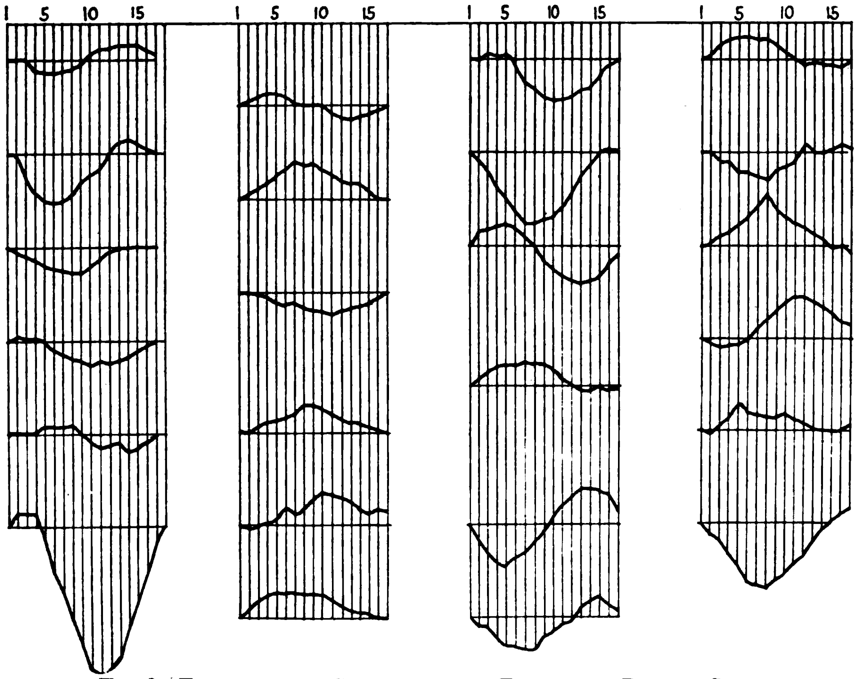


FIG. 3—ERRORS IN THE SPACING OF THE TEETH OF A DRIVING GEAR

that are in mesh. If this single tooth or nick happens to be in the transmission constant-mesh pinion and it is driven at 1000 r.p.m., there will be 16 distinct blows per sec. Elementary physics shows that, up to this point, these blows can be distinguished by the human ear as individual blows. With this same gear, rotating at 1000 r.p.m., if there are two nicks and they are a uniform distance apart in the gear, the gears will have 32 knocks per sec. This noise still will be distinguishable as individual knocks by some ears, but by others it will be noted as a distinct tune. However, if the two nicks happen to occur in this gear an uneven distance apart and the transmission is speeded-up, one can note readily the place

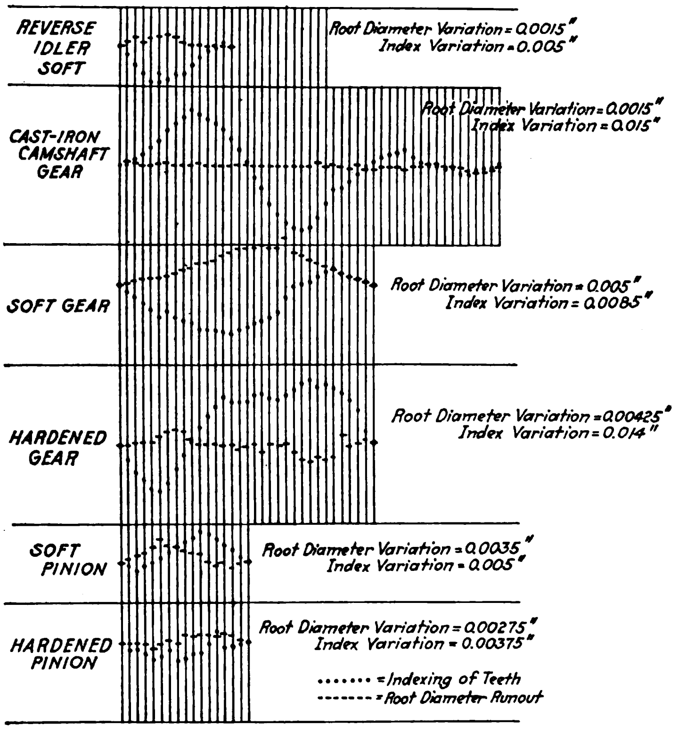


FIG. 4—CHART OBTAINED FROM A NUMBER OF GEARS AND PINIONS NONE OF WHICH HAS AN INDEX ERROR OF MORE THAN 0.005 IN.

where the gear noise changes from an individual knock into a sound that is usually designated as a rattle.

It is evident that influences similar to the nicks referred to above can be produced by inaccurate conditions in the gears. For example, should the tooth spacing in the transmission drive-pinion mentioned be such that the driven gear, instead of rotating at a uniform speed, is forced to increase and decrease its speed at every revolution of the pinion, very similar sounds will occur. If on one side of the driving pinion the teeth are 0.015 in. ahead of their proper position on the periphery of the pinion, the driven gear must gain and lose a corresponding amount in its steady motion. In so doing, it may strike on the back of its tooth rather than on the driven side, especially when idling, and produce a series of blows which still further increases and confuses the rattle.

It is not sufficient to check gears for spacing error from tooth to tooth. It is very desirable to check the accumulated error of a number of teeth, because a gear may vary 0.001 in. from tooth to tooth. With eight successive teeth each gaining 0.001 in. on the side of the gear and a similar number of teeth that may be losing 0.001 in., a total error of 0.016 in. might be imparted to the driven gears.

Fig. 1 illustrates a very simple device that has been used for checking tooth spacing. The gears are mounted on a bushing and one tooth comes against a stop. A dial indicator is arranged so as to be in contact with some tooth one-fourth, one-third or one-half way around the gear. When the dial indicator is set at zero, with the tooth against the stop at any one point, the distances between the two points can be measured and, if the gear be correct for indexing, placing any of the two teeth in the gear in similar positions should not cause the dial indicator to vary, especially if the gear runs true.

When the gear is first put on the indicating apparatus, the dial indicator is set at zero. We then put a mark at zero on the chart in Fig. 1 for tooth No. 1. The next step is to index the gear around one tooth. Any reading obtained is marked above the tooth number in the vertical line. We next index the gear around to tooth No. 3 and again mark the dial-indicator reading

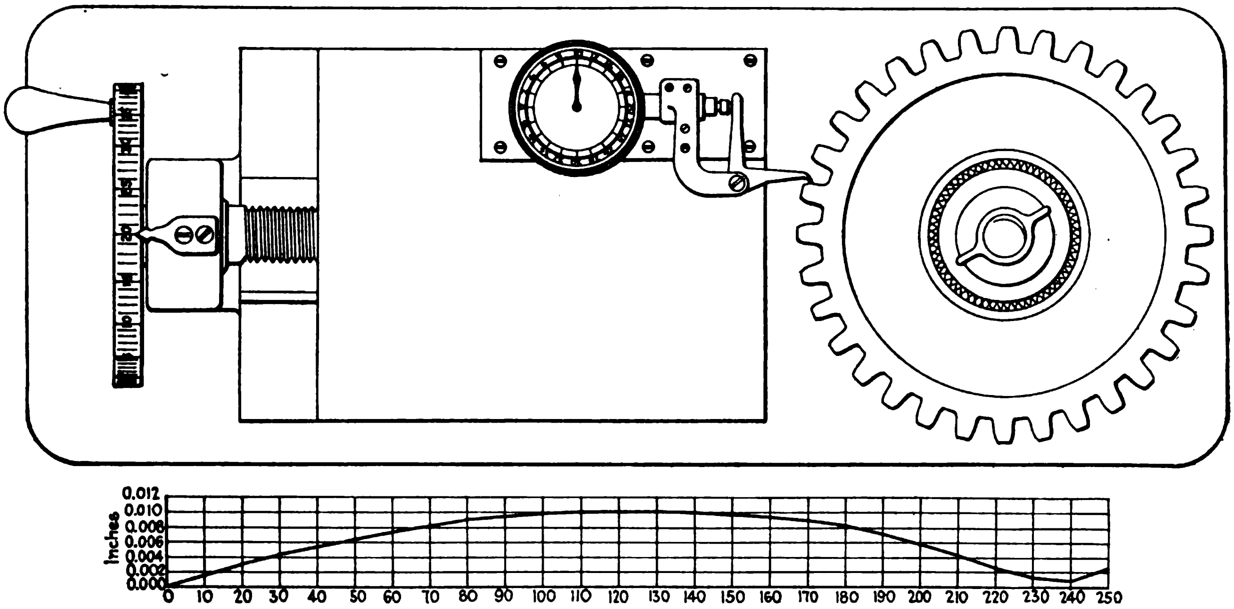


Fig. 5—Instrument for Obtaining an Analysis of Tooth-Forms Producing Noise and a Chart Showing the Outline of a Particular Tooth

opposite the number of thousandths of an inch that it may show. The gear is then indexed to teeth Nos. 5, 6, etc., until all the teeth on the gear have been indexed.

For the purpose of record, we now have a chart showing the accumulated variables. It will be seen from Fig. 1 that at no point is the spacing variable as great as 0.001 in. between any two teeth, but it can be in error a total of 0.008 in. or more when the error between the several teeth has accumulated. A better visual demonstration of this condition occurring in gears is made by means of a gear-tooth-form projector. When a gear-tooth form is projected upon a screen by this device, it will be noted that the magnification of the shadow is 100 to 1 and that, for every inch on the screen there is at least a 0.001-in. error somewhere in the gear. The shadow on the screen also shows the variation in uniform movement of the driven gear due to an error of this kind; that is, the driven gear, instead of having its tooth in the position of the outline on the screen, has been forced to advance a number of thousandths of an inch. It will be noticed further that this advance and retardation does not occur uniformly; that is, the advance may be confined to a very small number of teeth, remain there for a certain length of time and then be retarded slowly. A gear in this condition will give a rattle very similar to that which might be produced by unequal spacing. This condition can be studied best by charting it as described.

A number of variations as they occur in the finished gear are shown. Fig. 2 shows, first, a rapid drop, then a flat portion showing no change, then a rapid rise, another flat and then a return to the starting point. These curves require very little explanation.

It requires considerable imagination to determine just what happens when a driven gear such as those described is meshed with a driving gear having teeth misplaced as shown in Fig. 3; yet this occurs in daily practice. Fig. 4 shows a number of gears and pinions none of which has an index error of more than 0.005 in.

In addition to the two gears having a rather irregular action between themselves, their increasing and decreasing movement is carried on to the countershaft and the idler, which often have similar defects in themselves. Accumulating errors in gears often cause the fourth gear in a train to be as much as 0.025 in. away from its correct position and this change occurs in varying amounts

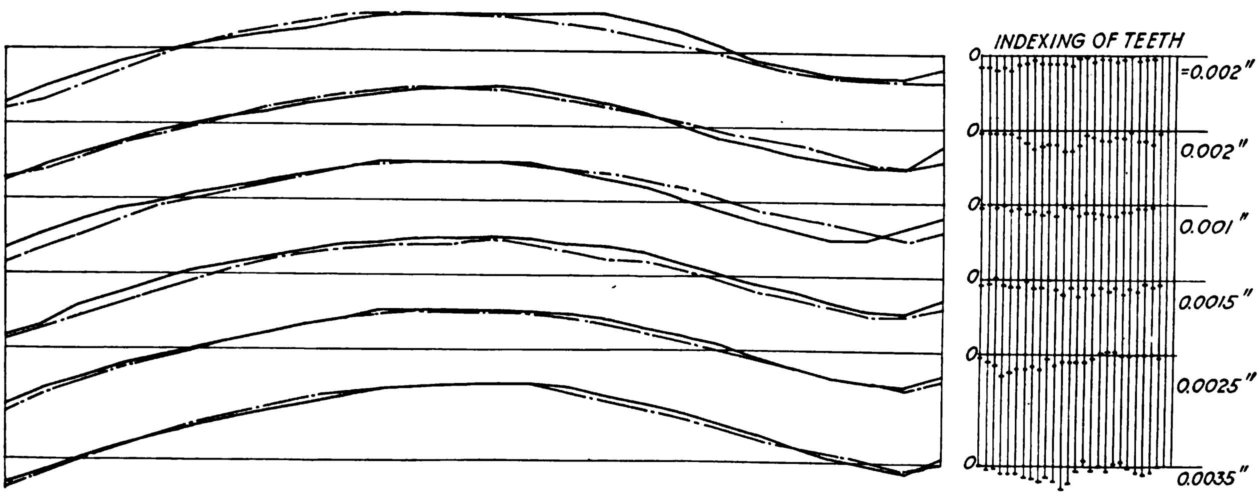


Fig. 6—Charts Showing Some of the Variations That Occur in the Production of Gears

depending on the number of teeth in the gears of the gear train. From this it is seen readily that the countershaft does not rotate smoothly and that the gear on the other end of the countershaft, in addition to the variable movement given it by the errors in the first two gears, imparts its error to the gear meshing with it. The action of the fourth gear will also lack uniformity in addition to that imparted to it by the driving gear, to the extent of the error on the fourth gear. If the ratios between the gears referred to are a direct proportion such as 2 to 1, these several shocks will occur uniformly, the error of the first uniformly with each revolution of the respective gears; however, if the ratios should be odd, such as 16 to 31, it will be difficult to estimate the change of speed that takes places.

Hum or sing is not nearly so difficult to analyze as the matter of rattle in a transmission gear. With this same transmission run at a speed of 1000 r.p.m. at the pinion shaft, if the pinion should happen to have 16 teeth, it will be found that 250 teeth per sec. go into and out of mesh. If the teeth are not correct in shape and the gears are under a slight load, there may be 250 blows per sec. under certain conditions, which we are told corresponds to the tone of middle C on the piano. Should fewer teeth go into and out of mesh, and this may be caused by a slower speed, a much lower pitch can be produced. In a similar way, because of the speed reduction that occurs in the usual type of transmission between the drive pinion and its countershaft, the tone produced by the reverse idler is very low and, instead of producing a hum or sing, it will produce what we usually call a growl. Errors in the sliding gears, because of their higher speeds, will produce higher pitch growls and approach a hum.

A great many instruments have been developed for the purpose of analyzing tooth-forms producing these sounds. The one that we have worked out and have chosen to use is shown in Fig. 5. It consists of a dial indicator mounted on a guided slide. We place the gear in a definite position with respect to the indicator, start at the

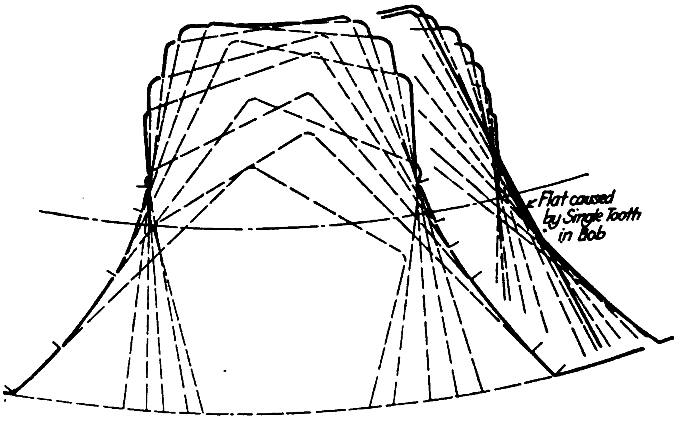


FIG. 9—A SINGLE HIGH TOOTH IN A HOB WILL PRODUCE A WIDER FLAT IN THE GEAR THAN IF ALL THE TEETH ARE OF UNIFORM HEIGHT

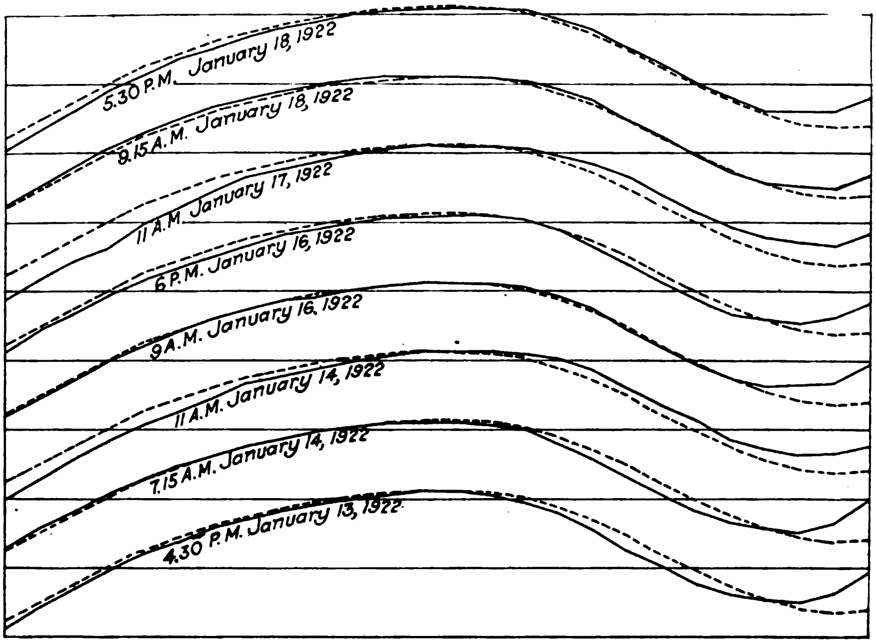


FIG. 7—VARIATIONS OCCUR IN THE PRODUCTION FROM THE SAME HOB ON DIFFERENT DAYS AND ALSO AT DIFFERENT HOURS OF ANY ONE DAY

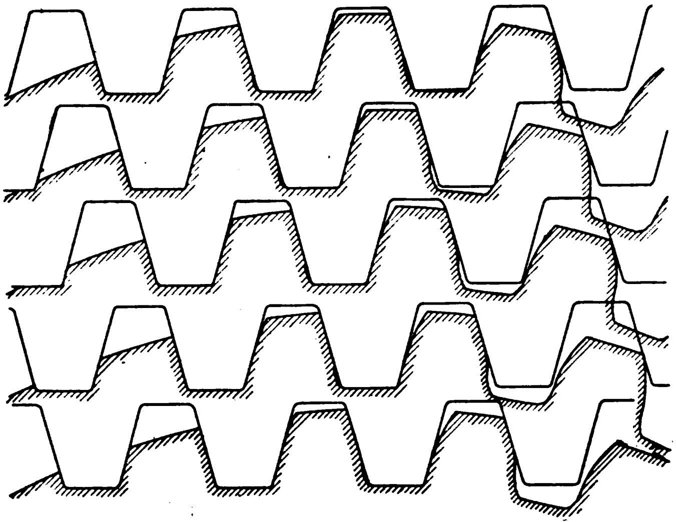


FIG. 8—OUTLINE OF A GEAR PRODUCED BY THE HOBBING METHOD

point of the tooth and set the indicator at zero. The slide is then moved toward the gear 0.010 in. and the indicator-reading marked on the chart shown in the lower portion of Fig. 5. The slide is then moved 0.010 in. more, the reading is marked again, and this is continued until the bottom of the tooth is reached. By taking the gear that has just been charted off the bushing and placing another gear in its place, other tooth-forms will be compared with the first.

The charts reproduced in Fig. 6 show some variations occurring in production, these particular charts having been made from gears with which special care had been taken to secure good gears. This was done by men who were not previously familiar with the method that was going to be used for the gear inspection. Fig. 7 shows some curves that occur in production from the same hob, and also curves made at different hours of the day.

Cutting Machines

The causes of the errors referred to are various. Some of them occur in hardening, some in the cutting machines and some in the cutters. We have found these errors in all of the types of machine that we have used. For the purpose of this discussion we are selecting a hobbing machine.

The hob is a generating tool that produces a gear such as is shown in Fig. 8. It will be seen readily that if all tooth-heights of the hob are the same, each hob-tooth

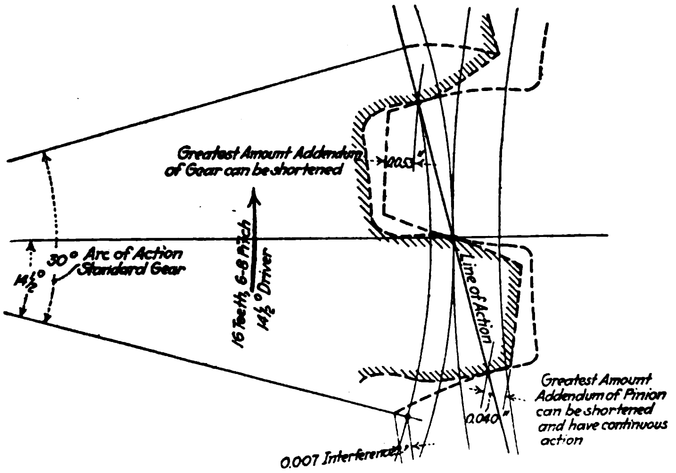


Fig. 10—Preliminary Layout That Is Made To Determine the Length of Tooth Necessary To Give a Complete Arc of Contact

generates, roughly, a flat in the tooth-form. If any of the teeth in the hob is high, a wider flat will be produced as shown in Fig. 9. Should the tooth-heights be correct and the hob be mounted in the machine with a run-out, a leaning tooth can be produced, depending on the sidewise setting of the hob with the gear. Also, should the hob be correct and the end-thrust collar in the hob spindle be out of parallel, giving the hob a slightly reciprocating motion with each revolution, an error can be produced that may compensate for the hob run-out or may add to it. Should the thrust collar at the rear end of the spindle be adjusted loosely so that the spindle may have end-play, the hob, as it cuts on one side, will be forced over and then back with each tooth of the gear and produce corresponding errors in the tooth-form. Again, if the gears in the hob-grinding spindle have inherent index-errors in them, or should the gears driving this gear be concentric or improperly spaced, their errors will be transferred to the different teeth of the gear being cut. Another important element in connection with the hob

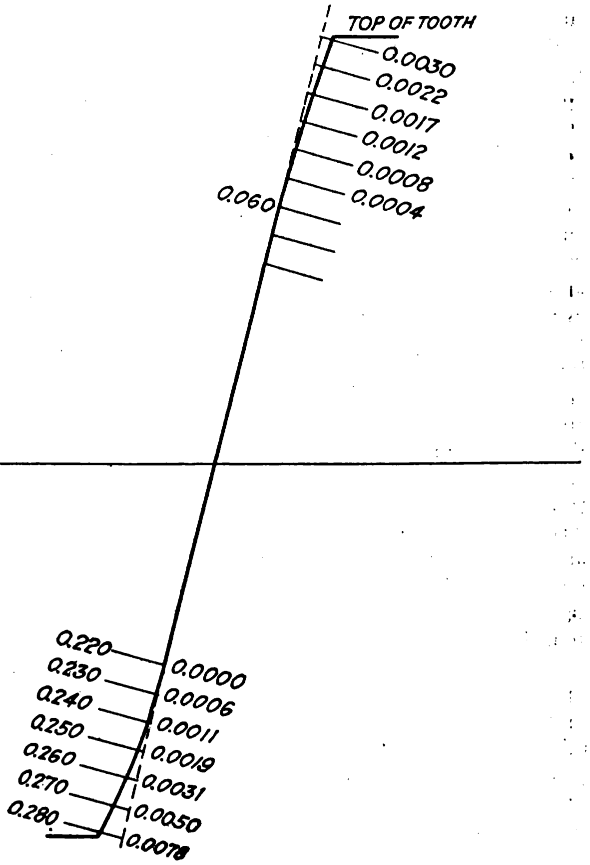


Fig. 11—Outline of the Hob Form Selected from the Layout Shown in Fig. 10

is the fit of the hob spindle. We will all agree that, should the hob spindle be tight for a certain portion of the revolution and loose for a certain other portion of the revolution, a sagging will occur in the driving-gear train which will be very detrimental to the tooth-form. Some hobbing machines are built so that the bevel gears in the hob drive-spindle give a thrust in the opposite direction to that given by a spiral pinion driving a hob-spindle gear. This permits a back-and-forth movement of a hob-

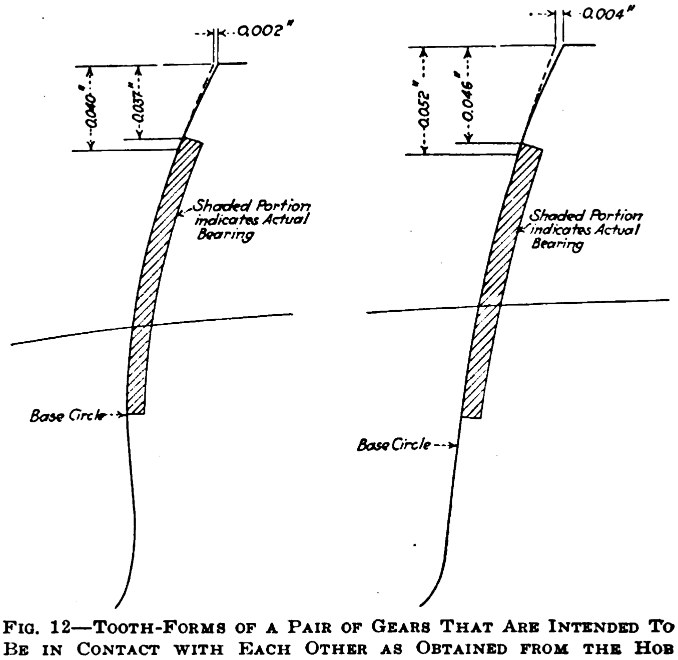


Fig. 12—Tooth-Forms of a Pair of Gears That Are Intended To Be in Contact with Each Other as Obtained from the Hob Outline Shown in Fig. 11

spindle drive-shaft and sometimes leaves its impression on the tooth-form.

Without going into the details concerning the other gears in the hob-spindle train, we might consider the influence of the thrust collar and the fits of the work spindle. In most hobbing-machines the bearings are kept fairly tight, and a great many operators insist that the hob spindle be kept warm. This also applies to the worm-shaft driving the wormwheel and, to a certain extent, to the work spindle. Unless the machines are extremely well adjusted, the thrust collars on the spindles so fitted score very easily and cause the spindles to be tight or loose, depending on various portions of its revolution. This causes a corresponding sag in the gears driving the index wormwheel and seems to be the main cause of the index errors already referred to. Another source of error is looseness in the gibs of the hob saddle. It is very difficult to move the hob slide across the face of the tooth without having some play between the gibs. This amounts, in a very similar manner, to the error that is obtained by having end-play in the hob spindle. Another contributing error to tooth-form is depth of cut. There will be considerable error in the tooth-form if the hob is sunk several thousandths of an inch too deep. Should the hob be straight-sided, the pressure-angle will increase with the depth of the cut and decrease as it is raised above the pitch-diameter. Another factor having considerable influence is the outside support for the hob spindle. We have had considerable difficulty in placing this outboard support of the hob spindle back in exactly the same place, giving us exactly the same condition of hob spindle as before.

When the work spindle is caused to rotate ahead of or behind its proper position, we necessarily have certain tooth-form errors in addition to index errors, and also in addition to those produced by the hob, its spindle and driving mechanism. There are conditions under which some of the errors referred to are counterbalanced by other errors. However, there are also conditions in which these errors accumulate. Considering the number of gears in a hobbing machine and the number of possibilities for errors outside of these gears, it is largely a matter of chance whether suitable combinations can be obtained to produce proper tooth-forms.

Tooth-Forms

Having all these liabilities to error in mind, the question often arises as to which is the noisiest tooth-form. Of the various kinds of gear that we have been able to cut and that we have had cut for us by a large number of different manufacturers, we have found that the errors from a general definite shape have apparently more to do with noise than any existing type of gear. The nearer to being correct a 14½-deg. pressure-angle-tooth gear is, the quieter it will be. We have never yet been able to secure 14½ and 20-deg. pressure-angle gears that had similar errors. The indications are, however, that the differences in sound due to the differences in the pressure-angle are very slight when compared with the differences in sound due to different errors in the same gear. We are using 14½ and 20-deg. pressure-angle gears regularly in production, and there seems to be little to choose between them.

The details of tooth-form are of some importance. Continuing with transmission gears, Fig. 10 shows the first layout which we make. This is with a view to determining the amount of tooth length necessary to give a 100-per cent arc of contact. With this information in hand, we select a hob form such as is shown in Fig. 11 and, using this on paper, we roll out two tooth-forms as shown in Fig. 12, one for each of the gears that are intended to be in contact with each other. The next step is to roll these gears on each other to determine the interferences, if any, and the amount that they are topped-off; then, if necessary, the hob form is modified and the same procedure carried through. When the hob form is established on paper in this manner, it is charted as shown in Fig. 13 and the hob supplier is asked to conform to this shape. Definite tolerances are given for the amount of variation from this form. On receipt of the hob, we inspect this form on a hob-checking apparatus very similar to that which we use for checking

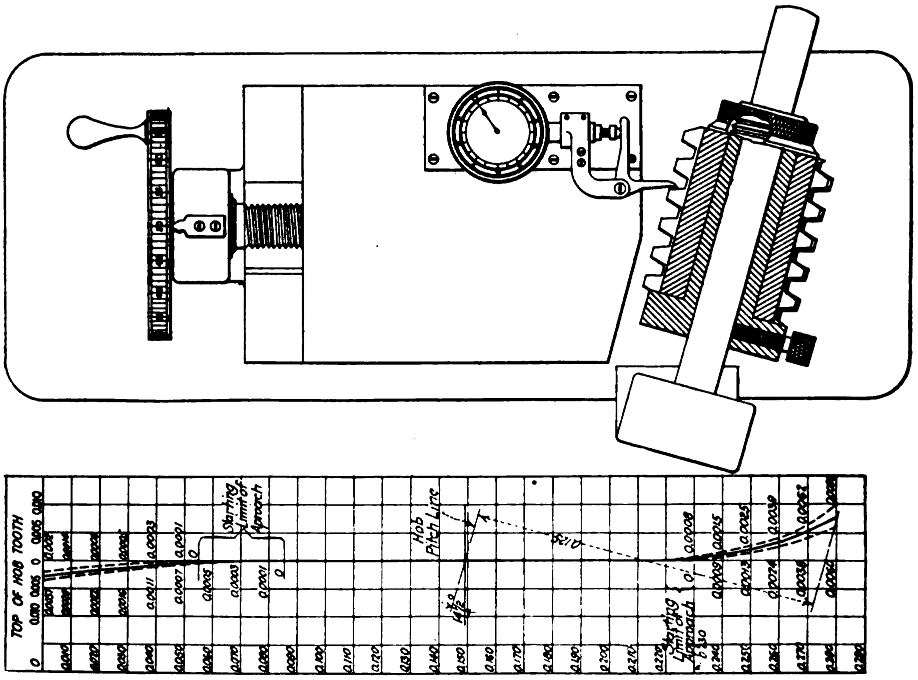


Fig. 13—Apparatus Used for Checking Hobs after Their Receipt and a Typical Diagram That Is Furnished Suppliers of Hobs Giving the Outline of the Hob Form

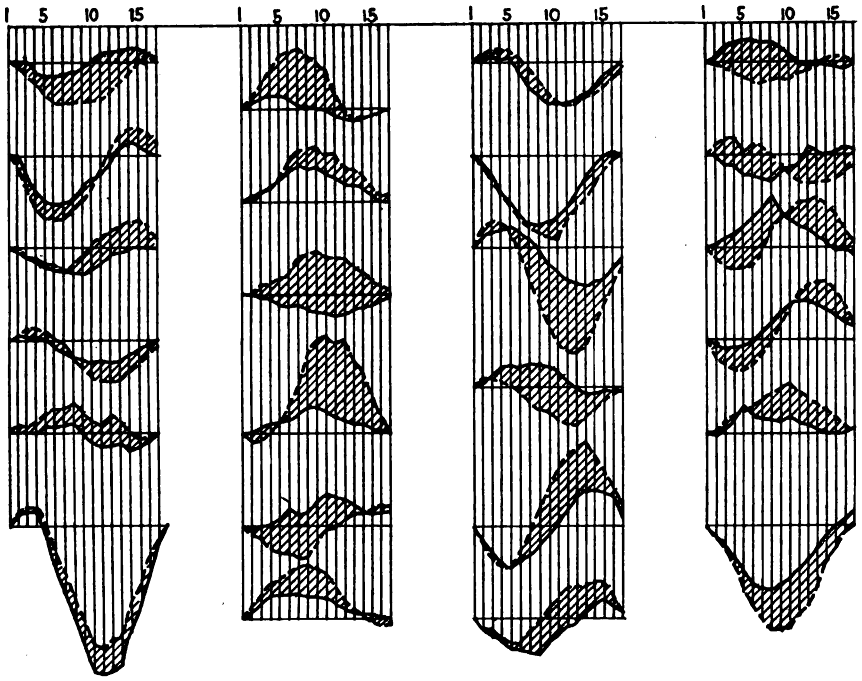


FIG. 14—CHART SHOWING THE VARIATION IN 24 GEARS, BEFORE AND AFTER HARDENING

gear-tooth forms, a drawing of which is shown in Fig. 13. If these conform to our standard requirement, it is expected that the hob will be satisfactory. We do not require any test of the hob in a cutting machine, because of the large number of errors that will be introduced by the machine, either in correcting hob errors or resulting in having an apparently correct hob rejected without cause.

HARDENING OF GEARS

Relative to the errors produced by hardening, we have prepared a number of charts showing the condition of the gear in the green and the condition of the gear in the hard. Fig. 14 shows 24 such gears and the variations occurring in them. Similar variations occur in gears mating with these gears. It will be seen that the cutting errors are considerable and that, at times, the hardening errors compensate for the cutting errors, while at other times the hardening errors go in the same accumulative direction.

Prof. John J. Keller gave a very interesting paper before the American Society for Steel Treating recently on why steel warps, showing what happens in a piece of steel when it is cooled and quenched. My impression from this paper is that freedom from warpage in steel is a matter of uniform hardening and cooling; also of removing all forging strains before machining. The question as to whether oil-treated steel is better than carburized is still unanswered as regards warpage. We have hardened more than 5000 gears of different brands of steel and carefully checked them. We find that there is very little difference in the warpage under the same hardening conditions.

GRINDING GEAR-TOOTH FORMS

The necessity for grinding gear-tooth forms depends largely on our ability to cut and harden gears, maintaining definite shapes. However, there is a large difference in the number of rejections that we have from gears ground by different processes. Our reports at this time show that out of 5000 gears ground by one method we have had a 14-per cent rejection. This is slightly greater than that which we have had from the hob gear without grinding. By another method of grinding of a similar number of gears, we have had less than a 0.5-per cent rejection, as well as more satisfactory gears. In the first case, four gears of the transmission were ground, and in the second case only two gears of the transmission were ground. All transmissions were passed by the same inspector and inspected to the same standards. We, of course, are looking forward with great interest to the continuation of our experiment on gear-grinding.

FAN DYNAMOMETERS

ON p. 45 of Vol. II. of the S. A. E. HANDBOOK is published information on the fan dynamometers. As Vol. II has not been issued since October, 1920, members of the Society joining since that date have not received copies of this volume containing general engineering information. Members using the volumes issued prior to 1920, however, should bear in mind that the note on p. 45a, reading "Radial setting figures appearing on the curves represent distance from center of hub to center of area of blades," is incorrect as the numbers on the curves actually indicate inches from the center of one blade to the center of the other. In using this information, it should be appreciated that results obtained by the use of fan dynamometers are highly unreliable. The Bureau of Standards has made tests which show that variations of as much as 20 per cent are possible under certain weather conditions.

Members desiring to supplement the information published in the Handbook on this subject should refer to a report on "Variable Torque Fan Dynamometers" issued by the National Advisory Committee for Aeronautics, an abstract of which appeared in *Automotive Industries* for March 24, 1921, and to p. 1688 of *Marks Mechanical Engineers Handbook.*

Some Unique Features of Automobile Production

By WILLIAM DUNK[1]

DETROIT PRODUCTION MEETING PAPER

THE author discusses the Franklin automobile designs in regard to the production of an air-cooled cylinder, the making of case-hardened crankshafts and the machining of duralumin connecting-rods, commenting also upon experiments made with a view to producing hot-swaged rear-axle drive-shafts.

The unit-cylinder construction of the Franklin engine, with the cylinder-head cast integrally with the body of the casting and steel cooling-fins cast in the outer surface of the cylinder-wall, presents a different problem from that of machining cylinders cast-in-block, and the procedure is described in some detail, inclusive of the machine tools used. The case-hardened crankshaft and the duralumin connecting-rod are treated in a similar manner. The experiments in the manufacture of a hot-swaged rear-axle drive-shaft have extended over the past 18 months, but have not yet justified quantity production by this method.

ALL engineers realize that there are a large number of problems to be met in the production of automobiles, and that the majority of these problems are much the same with the H. H. Franklin Mfg. Co. as with any other. I will endeavor, therefore, to confine this discussion to the following topics that are somewhat unique in Franklin automobile designs; namely, (*a*) the production of an air-cooled cylinder, (*b*) the making of case-hardened crankshafts, (*c*) the machining of a duralumin connecting-rod and (*d*) experiments with a hot-swaged rear-axle drive-shaft.

PRODUCTION OF THE AIR-COOLED CYLINDER

The design of the Franklin engine necessitates the use of unit-cylinder construction rather than the regular block-cylinder design. This fact makes the production of the cylinder a somewhat different problem from that of machining a block casting. In the Franklin cylinder, the head is cast integrally with the body of the casting and steel cooling fins are cast in the outer surface of the cylinder-wall.

The foundry practice in connection with the production of these castings is of interest. The mold is composed of three distinct sections, the lower section being an assembly of dry-sand cores for making the head of the cylinder and the valve ports. Next above this section is a green-sand section containing the cooling fins, and the upper section of the mold is a green-sand section for the neck of the cylinder between the cooling fins and the bottom flange, the cylinder of course being cast in an inverted position. The core assembly for the head of the cylinder is made up of five unit-cores that are made separately and assembled by pasting-in an accurate pasting jig so that the valve-port cores are held accurately in position while drying. The valve-port cores are provided with prints that are used to locate the upper end of the main cylinder-barrel core; for this reason it is necessary to be very accurate in the location of the port cores.

In making the green-sand section containing the cooling fins, the cylinder pattern is milled longitudinally with the proper number of grooves to receive the vertical cooling-fins. These grooves are 1/16 in. in depth, which corresponds to the amount that the fin is imbedded in the cylinder-wall. This section of the mold is made on an air-operated jar ramming-machine provided with a sand hopper directly above the mold. A very small amount of sand is used in this section of the mold, not over 2 in. of sand being provided outside of the diameter of the cooling fins. The fins are held in place in the cylinder pattern by simply wrapping a piece of soft iron wire around them and twisting the ends tight. This wire remains in the mold until the casting has been made, and serves to keep the section of green sand that lies within the cooling fins from sagging below the rest of the mold. The upper section of the mold is also made in green sand and is located on the center section by taper dowels. The last operation in making the mold is the lacing of the cylinder-barrel core and the strainer-gate core through which the metal is poured. The cooling fins are made of ordinary cold-rolled strip-steel which has been tinned and folded to a right angle about 5/16 in. from one edge. These cooling fins, when placed radially about the cylinder pattern, form a completely enclosed air-jacket.

The greatest source of casting loss that we experience in making this piece is caused by loose flanges. Unless the flanges extend at least 1/16 in. below the surface of the cast iron, they will not be properly held in place.

The analysis of iron used in making the Franklin cylinder is approximately as stated in Table 1.

TABLE 1—ANALYSIS OF IRON USED IN THE FRANKLIN CYLINDER

Constituents	Per Cent
Silicon,	2.00
Phosphorus,	0.50
Manganese,	0.50
Sulphur,	0.10
Carbon,	3.25

The machining operations are as follows: The cylinder casting is first mounted in a chucking fixture on the table of a small vertical boring-mill, and the bottom flange is faced and counterbored. This counterbore is used for locating the cylinder while drilling the four hold-down holes on the second operation. These hold-down holes are used for locating the cylinder for the boring operation, which is performed next. Our present method of boring the cylinder consists of making four separate cuts into the cylinder under a single-spindle Baker boring-machine. The necessity of making this number of cuts is caused by the fact that the cylinder-wall has no support to prevent its being split, if a heavy cut were attempted. The thickness of the metal of the finished cylinder between the cooling fins and the cylinder bore is only 3/16 in.

A new cylinder-boring machine has just been completed and tested which makes five separate cuts in the cylinder, operating on five separate cylinders simul-

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taneously. The first spindle of the machine rough-bores, and removes about 1/8 in. of metal from the cylinder-wall. The second spindle rough-bottoms the cylinder to within 1/32 in. of the finished depth. The third spindle semi-finish-bores, and leaves 0.035 in. of metal for finish-boring. The fourth spindle finishes the bottom of the cylinder to depth, and the fifth finish-bores to grinding size. We are leaving about 0.007 to 0.010 in. of metal for grinding at present, and we may find it possible to reduce this to a maximum of 0.005 in. The fact that we are dealing with unit cylinders allows us to center the cylinder very accurately for the grinding operation. The complete boring operation as performed by the new cylinder-boring machine occupies about 1½ min. per cylinder.

The cylinder is ground on the standard type of Heald cylinder-grinding machine equipped with a special small table for unit-cylinder work. It is held against an ordinary angle-plate grinding-fixture, and is located with a plug through a master bushing inserted in the grinding fixture. Cooling water is flowed on the outside of the cylinder during the grinding operation. The valve ports in the cylinder are reamed holes about 2½ in. in diameter and 1 in. in depth. They are held to a limit of 0.001 in. to provide a sufficiently tight joint with the extension pieces that attach them to the suction and exhaust manifolds. A special double-head machine was built to bore and ream the port holes as accurately as this.

Briefly, this machine consists of two three-spindle movable-heads operating toward a central holding-fixture on which four cylinders are mounted; there are three working positions and one loading position. The central fixture is arranged so that it can be indexed at the end of each stroke of the machine. The first pair of working spindles rough-bores the port holes to within 0.035 in. of the finished size. The second spindle does not remove any metal from the diameter of the hole but simply performs the bottoming operation. The third pair of spindles finish-reams the holes to size. By arranging the machine to operate on both ports at once, the thrust of the cutters has been neutralized almost exactly, because the ports lie nearly opposite each other across the cylinder casting. The cutters used in this machine for all operations are a standard type of shell-end mill, and they seem to give very satisfactory results. It is not necessary to change cutters oftener than 1 set to each 350 cylinders. The capacity of this machine is one complete cylinder every 40 sec. A particularly rapid means of loading and unloading the cylinder is necessary, to allow the operator to perform these operations during the cycle of the machine.

The valve-stem guides are separate castings machined complete and pressed into place in the cylinder casting, the final reaming operation being performed in the valve-stem holes after the guides are pressed-in. The seating and other minor operations on the cylinder are performed in the usual manner and do not require any special mention.

The Case-Hardened Crankshaft

Another interesting manufacturing detail is the Franklin case-hardened crankshaft, which we have been producing for the past 2 years. The primary object in incorporating this type of shaft is naturally to prolong the life, so far as possible, of the bearing surfaces. Preliminary experiments developed the fact that, if a crankshaft can be hardened to a scleroscope hardness of between 80 and 100, the life of both the main bearings and the crankpin bearings can be prolonged considerably. In some of our test cars as much as 50,000 miles was covered without any adjustment of the bearings being necessary.

In putting a proposition of this kind into production, a number of problems had to be met concerning which very few data were obtainable. The crankshaft forging has a carbon-content of from 0.15 to 0.25 per cent and the required depth of carburization is specified at 1/16 in.

In the manufacture of this shaft, the forging is first machined to within 0.040 or 0.045 in. of its finished diameter, with the exception of the flange and threaded ends which are left in the rough. The forging is then ready for the copper-plating operation, which is applied to prevent carburization of all parts of the shaft except the bearing surfaces. Several ideas were tried out in connection with preventing the copper-plating of these surfaces during the process, and we finally adopted the method of wrapping all bearings with pure gum rubber held in place with clips. This affords a very simple means for preventing the copper-plating of the surfaces; the rubber bands are easily removed and can be used again and again.

In carburizing the shafts they are packed, three in a box, in nickel-alloy carburizing-boxes, using either Q alloy or thermalloy. The approximate size of the carburizing boxes required is 12 x 44 in., and about 18 in. deep. A liberal amount of carburizing material is provided around each shaft. The boxes are sealed with the regulation cement and are loaded into the carburizing furnaces by a specially constructed charging truck, each furnace-chamber having a capacity of six of these boxes and each box having a total weight, loaded, of about 800 lb. The carburizing period required to get the required depth of penetration is between 22 and 24 hr. The use of nickel-alloy boxes seems to have a decided tendency to reduce the number of hours required to get this penetration.

At the end of the carburizing process, the shaft is returned to the crankshaft-machining department; the flange end and the threaded end are machined, roughly, to the finished dimensions, this operation having the result of removing all of the case-hardened surface produced by carburization. This allows for the machining of these parts after the shaft has been hardened.

The shaft is then returned to the heat-treating department and reheated in a gas-fired furnace to a temperature of 1420 deg. fahr. To keep the shaft as nearly straight as possible during the quenching operation, a hinged type of quenching die was devised that permits rapid loading for the purpose of quenching the shaft as soon as possible after removing it from the furnace. This disc is of very heavy construction, and is handled by an air hoist. The shaft is quenched in water at main temperature, and comes from the quenching operation usually not over 1/16 in. out of straight. It is necessary to straighten the shaft carefully at this point, before proceeding with the semi-finish and finish-grinding operations.

We have found it necessary to divide the finish-grinding operations into semi- and finish-grinding so as to use a rather coarse soft grinding-wheel for the semi-finish grinding in bringing the shaft to the finished size. The last 0.005 in. of metal is removed from the shaft with a very fine soft wheel to get the required smoothness of finish.

One somewhat uncertain feature that we always meet in the production of this shaft is the amount of shrinkage that will take place in the hardening operation. It is necessary to anticipate this, and to provide for it in machining the soft shaft. Our experience has shown that this shrinkage varies to a considerable degree. Our normal allowance in the overall length of the shaft and

the spacing of the bearings is about 0.035 in. for a 5½-in. center-distance between cylinders. We find that this allows us to finish the shaft within the limits required for length, in the majority of cases.

Our scleroscope-hardness requirements of 80 to 100 give us a shaft that has exceptionally long life. I neglected to state, however, that the point of contact of the quenching die with the shaft produces very small soft spots; but they do not seem to interfere with the general durability of the bearing surfaces in any way.

The Connecting-Rod

The Franklin connecting-rod is an aluminum-alloy forging, either of duralumin as furnished by the Baush Machine Tool Co. or of 17-S alloy furnished by the Aluminum Co. of America. This aluminum alloy contains the elements given in Table 2.

TABLE 2—ANALYSIS OF ALUMINUM ALLOY IN THE FRANKLIN CONNECTING-ROD

Constituents	Per Cent
Copper,	3.50
Manganese,	0.20
Magnesium,	0.25
Iron, not more than,	0.75
Silicon, not more than,	0.75

The elastic-limit of the forging is between 30,000 and 35,000 lb. per sq. in.; the ultimate-strength is between 50,000 and 55,000 lb. per sq. in.; and the elongation is from 15 to 25 per cent.

The forging is heat-treated by heating it to 920 or 940 deg. fahr., and is quenched in boiling water. It is then allowed to age about 1 week to bring it to a scelerscope hardness of 90 to 100.

We use about the following procedure in machining the duralumin forged connecting-rod. The forging is first coined under a heavy toggle-press, bringing the length of the wristpin and crankpin ends to within 0.025 in. of the finished size. It is then loaded onto a special fixture on a Blanchard grinding machine that has alternate roughing and finishing stations. It is ground first on one side, then removed from the roughing station, moved to the next adjacent station and finish-ground on the other side. A special multiple-spindle boring-and-reaming machine having an indexing table is used for boring and reaming the wristpin and crankpin holes. The operations which this machine performs are to rough-drill, rough-ream and semi-finish ream.

The action of the forging in these machining operations is similar to that of mild steel. The power required to drive the drills and reamers is approximately the same as the power we employed formerly with our steel connecting-rod, and the cutting speed does not exceed that for steel by more than 25 per cent. In machining this forging, it surprised us to find that it machined so nearly like a steel rod. We had anticipated being able to machine this rod at a much greater speed than we found possible. The extreme toughness and wire-like quality of the chips are convincing proof that the metal with which we are dealing is something more than ordinary aluminum.

The forging is next drilled for the rod bolts and is then put over a multiple fixture for parting the cap from the rod and facing the seats for the bolts and nuts. A broaching operation follows for providing slots in which the lower-end babbitt blocks are locked. These babbitt blocks are forced into place and broached to their semi-finished size in an ordinary two-spindle broaching-machine. The finishing of the wristpin end of the rod has been rather difficult, because it is impossible to ream a sufficiently good wristpin hole in the bearing end. Our present method of handling this operation is to semi-finish ream this hole, removing about 0.025 to 0.030 in. of the diameter, then to use a final reaming operation that removes about 0.005 in. and a final burnishing operation that is done with a high-speed burnishing-tool under a copious flow of regular medium automobile motor-oil. We find that by leaving a uniform amount of about 0.0005 in. on the diameter we can produce a highly burnished surface in the wristpin hole and maintain very uniform size. However, to obtain the fits required by our engineering department between the wristpin and the connecting-rod, it is necessary to do a certain amount of wristpin selection. When it is finally finished, the alignment of this rod is held to a limit of ±0.005 in., measured between the ends of arbors 12 in. long.

The finishing to size of the lower end of the rod is left until it is ready for assembly with the crankshaft. This operation is performed on a special type of boring machine in which we use boring bars of five separate diameters, increasing their diameters in steps of 0.00025 in. from the minimum size required up to the maximum; this allows us to cover the entire range in the crankpin bearings with the proper amount of oil clearance. At this time we use an oil clearance of about 0.001 in. for this bearing.

The weight of the finished connecting-rod runs very uniform, it being necessary to maintain but two standards of weight to keep within the limits specified by our engineers, which are a variation in weight of any two connecting-rods on an engine of not more than 0.2 oz.

Hot-Swaged Rear-Axle Drive-Shaft

We have been experimenting for the past 18 months with the production of drive-shafts by the hot-swaging process but, up to the present, we have not felt justified in following this method of manufacture in large production quantities. The saving in the amount of material used to make a rear-axle drive-shaft, and the reduction in the amount of labor required by this method, make it a very attractive proposition, and we are not inclined to give up the idea without a struggle.

Troubles have developed in the source of our experiments. We have been unable to obtain a swaging machine of a construction heavy enough to withstand the load put upon it by an operation of this kind. We had experienced continued breakdowns of our original machined, and we have practically rebuilt it. The problem of producing a swaged shaft, without having the finished product show any twisting effect whatever from the swaging, has been difficult.

In performing the second swaging operation, which consists of necking the shaft below the diameter of the splined end, we have not been able to reheat for this operation without excessive scaling, which mars the appearance and reduces the strength of the finished shaft. The procedure that we are following at this time requires three separate swaging operations and three separate heats for making them. If it were possible to take the shaft directly from the first swaging operation and bring it up to swaging temperature again, which is about 2000 deg. fahr. in our case, and to perform the second or necking operation immediately, we believe we could reduce the amount of twisting and the amount of scaling that we experience at present. The third swaging operation, that of producing the taper and the threaded end, is comparatively simple and gives us no trouble.

We are now considering the installation of a much heavier swaging machine. We find it can be built to order for us. We are considering also the installation of a second furnace to be used as a booster between swaging operations Nos. 1 and 2. With this proposed new equipment, we should be able to produce a very satisfactory product. We would be very glad, however, to learn of the experiences of some of the other automobile companies in producing a hot-swaged rear-axle drive-shaft.

EDUCATION FOR HIGHWAY ENGINEERING AND TRANSPORT

THE second national conference on education for highway engineering and highway transport was held in the City of Washington, on Oct. 26 to 28, under the auspices of the Highway Education Board. This Board has headquarters at the City of Washington and is composed of representatives of those agencies of the Government and industry having an intimate contact with and interest in the construction and maintenance of American highways. The personnel of the Board is as follows: John J. Tigert, United States Commissioner of Education, chairman; Thomas H. MacDonald, chief of the bureau of public roads, Department of Agriculture; Roy D. Chapin, vice-president of the National Automobile Chamber of Commerce; H. W. Alden, representing the Society of Automotive Engineers; Harvey S. Firestone, representing the Rubber Association of America; Col. F. C. Boggs, Corps of Engineers, United States Army, War Department; F. L. Bishop, dean of the School of Engineering, University of Pittsburgh, secretary of the Society for the Promotion of Engineering Education; and a representative to be selected by the American Association of State Highway Officials.

An important object of the conference was to discuss means of meeting the public demand for trained men in the construction and operation of highways, and for this reason the work of the 11 educational committees must be considered as constituting a vital part of the meeting. The program allotted much time to the sessions of these committees, whose final recommendations on details of curricula may be expected to be ready for announcement in about 4 months. One afternoon session of the general meeting was devoted to the subject of the content of prospective courses on highway engineering and highway transport in the engineering schools.

On the general program there were at least two addresses of direct appeal to the automotive engineer, and, curiously enough, both of them emphasized the success of the British system of collection and distribution of goods by motor truck. The first address, by Frederick C. Horner, on European and British Road Transport Practice, described more particularly the types and design features of the British trucks, while the second, by W. H. Lyford, on Cooperation versus Competition Between Motor Trucks and Railroad Transportation, discussed the application of the British system to American conditions. Mr. Lyford, who is vice-president of the Chicago & Eastern Illinois Railroad, is of the opinion that American method applied to the British system can make the motor-trucking company a valuable ally of the railroad with profit and advantage to both parties. This view was presented as the conclusion of a remarkably able and profound study of the situation.

The other addresses of the general meeting dealt with economic and sociological features of highway transport and with various phases of research. It is interesting to note that the need for more investigation in the field of automotive engineering is recognized as a part of the problem of highway transport, though provision for the prosecution of automotive research is not contemplated by the existing organization for highway research. This is implied, for instance, in the address of Dr. W. K. Hatt, director of the advisory board on highway research of the National Research Council, and is otherwise expressed in Prof. Arthur Blanchard's proposal that truck producers create for this purpose an organization similar to the Portland Cement Manufacturers' Association or the American Zinc Institute.

Research was a topic that pervaded the entire program. This may find an explanation in the fact that a large proportion of the 225 delegates to the conference represented engineering schools. Not the least effective of the arguments for research were the two excursions to the Arlington Experimental Farm and to College Park, University of Maryland. The arrangements for these excursions were excellent, and practically every delegate was enabled to get a good insight into the progress of the investigations. The motion pictures of the Pittsburg, Cal., road-test experiments, as explained by the engineer in charge of the work, also created favorable comment in this connection, but probably the most impressive illustration of the value of research was afforded in the reading of the circular just issued by the Illinois State Highway Department, announcing new specifications for the cross-section of concrete pavements. This modified cross-section is a direct result of the Bates experimental road work, and its timely appearance on this occasion had a dramatic effect, because the new cross-section will provide a pavement that not only is better able to support trucks loaded to the legal limit, but will also reduce the cost per mile.

The National Automobile Chamber of Commerce must be credited with having added much to the interest of the general meeting, not only by providing conspicuously able speakers at strategic points of the program, but also by sponsoring the exhibit of the Eugene B. Clark collection of original paintings dedicated to the automobile industry, and entitled The Spirit of Transportation. This exhibit attracted the attention of the general public, as well as that of the delegates to the convention.

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The Group-Bonus Wage-Incentive Plan

By E. Karl Wennerlund[1]

Detroit Production Meeting Paper *Illustrated with* Charts

The author states that the purposes of every plan of wage incentive are to stimulate the worker to a greater effort than is generally obtained on a straight day's-work basis; to reward him somewhat in proportion to his effort; and the gaining of other advantages such as greater attention to conditions that curtail production, more uniform labor costs and the elimination of inefficient employes. He states further that nearly all industries engaged in repetitive work are now on an incentive basis.

After outlining the most successful wage-incentive plans and enumerating some of the conditions that must be met, inclusive of four specific fundamental principles of industry that are stated, the group-bonus plan is explained and the application of group standard-time is discussed at some length, supplemented by tabular data. Experience with grouping is then related and conditions favorable to grouping are mentioned. The summary states that the group plan, like any other wage-incentive plan, has for its main objective intensive production.

The subject of wage incentive is an important one to factory executives. During the past few years it has received considerable study. Nearly all factories now have their own specialists who make time-studies of operations and recommend or set production rates. Even where outside industrial engineers are employed for this purpose their services are regarded as temporary, with the idea of developing a local staff as rapidly as possible to handle the regular routine. The question of the particular form of wage incentive, therefore, becomes important. No single plan can be recommended for all factories. Local conditions and the character of product will have to govern in each case. It is, of course, desirable to have an incentive plan handled as simply as possible with a minimum of clerical effort, and with the least liability to error in quantity check.

The purposes of every plan of wage incentive are the same:

(1) To stimulate the worker to a greater effort than is generally obtained on a straight day's-work basis

(2) To reward him somewhat in proportion to his effort

(3) The gaining of other advantages such as the attention given to conditions that curtail production, more uniform labor costs and the elimination of inefficient employees

The result has been that nearly all industries engaged in repetitive work, with very few exceptions, are on an incentive basis. Until recently, the plan followed has been generally of the type known as the individual-effort plan, with either straight piece-rates or some form of premium or bonus based on time measurements. Grouping of employes was not often resorted to, except for such operations as made it impracticable to keep track of the output from individuals. However, the principles of grouping, with a division of the earnings among several workers, have been well known and have been applied for a long time to such work as steam hammering, riveting and handling materials; but, for work done distinctly by an individual, the usual practice has been to set a rate for that particular detail operation and to pay the individual worker for his production at that operation. The application of grouping of employes throughout an entire factory organization, wherever work is measurable, is rather a new development in industry.

In automobile factories and most accessory plants, the character of product is such as to make parts move progressively from one operation to the next to a very great extent. Processes have been standardized and divided into minute detail and, being of such repetitive nature, operations have been segregated into progressive flow-lines wherever practicable. This was a natural result from the desire to minimize the expense and delays incident to transporting materials in process. The consequent physical arrangement of these factories has had an important bearing on wage-incentive problems.

It must be taken into account that an individual-effort system of payment to workers entails considerable routine expense. The earliest method, after rates had been set, was for each worker to make out a report at the end of the day stating how many pieces he had completed under each operation; or a shop checker came for this information and inserted the quantities on a prepared form. A systematic check on quantities after each operation was, of course, impracticable with large-volume production unless some special means were provided; so, there was much opportunity for collusion between shop checkers and workers and there is no doubt that many took advantage, particularly on parallel operations, of the management's inability to obtain an actual check. One safeguard often resorted to was to discourage the worker from turning in more than a certain maximum rate of earnings; but, even then, it was no guarantee that amounts up to the maximum might not be fraudulently claimed. It was just about as logical to continue to operate a factory under these conditions as it would be to eliminate the time-clocks at the entrance gate or to do away with cash-registers. To do so would require a lot of faith in human nature. There is no doubt that the time-clock and cash-register have both helped many men to remain honest.

Well-organized factories, operating under the individual-effort plan in recent years, have developed various methods for obtaining physical checks on quantities after each operation. One of these is to have a set of coupons, numbered serially, attached to each large piece or to a container of a pre-determined number of pieces. The worker is then required to present these coupons as evidence of quantities. By another method known as the lot system, the worker obtains a job ticket for each lot. The start and finish time of the operation for each lot is stamped on the job ticket which bears the same serial number as the lot quantity. Later, when these tickets come to the office, each is checked off on a master lot-card, then extended and credited to the individual account.

These methods are all good in theory. No doubt they pay their way when compared with having no definite check, and they may be entirely practicable in small shops or for certain classes of work, but let us consider the

modern automobile factory. The lot method not only ties up a large inventory of parts in process, but to check off and extend thousands of job tickets, daily, also involves a great expense. One such factory had in excess of 25,000 job tickets for each working day, each one of which had to be carefully audited as to quantity and rate, extended for total and then credited to the account of the employe.

It was the desire to simplify the factory routine, and to escape from this mass of detail management without sacrificing the principles of quantity checks, that prompted the development of the group-bonus plan. Certain fundamental principles of industry had to be recognized and taken into account. The most important of these are that

(1) Under any incentive plan, the worker must continue to maintain an individual interest in the rate of production
(2) He should be guaranteed a satisfactory hourly wage
(3) He should be paid for all time saved instead of for only a part of it
(4) He should be able to compute his own earnings, accurately and quickly

Under the group plan, it would not make much difference in theory whether a fixed group-price were established or some form of bonus or premium payment were used. Our past experience, however, indicated that it was preferable to set standards of output in *time* rather than in money value. With changing labor scales, we should then have no disturbance of the existing time-standards, but would make the adjustments on the hourly-base wage-rates on which bonus earnings are computed. Time standards also have the advantage of a very flexible wage-rate, by which means new employes can enter our service at a minimum rate subject to advancement through length of service and personal ability. Such revision in wage rates can be made at any time without in the least disturbing shop standards or affecting other employes.

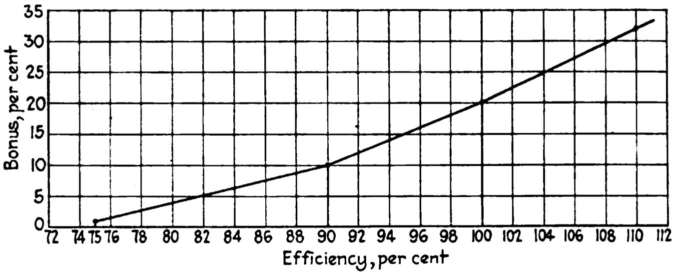


Fig. 1—Chart Showing the Relation between Efficiency and the Percentage of Bonus

On the other hand, a fixed group-price in money value would offer some very definite objections if adopted as a general plan, since the

(1) Division of the earnings among the group members would have to be on an even basis per hour worked; or
(2) Group earnings would have to be pro-rated on the basis of the hours worked by the individuals, multiplied by the predetermined base-rates. A change in an individual base-rate would affect all other members of the group, or the transfer of an employee with a different base-rate would upset their relative earnings

In regard to a uniform base-rate for all members of a group and their division of the earnings purely on an hourly basis, it is obvious that this method would prove unsatisfactory as a general policy; because, even in the same group, some operations may require more skill than others. Then there is the added consideration that employes who have been long in the service should be on a higher wage plane than beginners. The standard-time bonus-table as adopted offers every facility for differentiating between base-rates and permits a flexible system to suit the varying labor and factory conditions.

Table 1 is based on the theory that a bonus incentive of 20 per cent should be paid to a worker, in addition to his hourly base-rate, for attaining an efficiency of production of 100 per cent; and that, for a greater speed of production, he should be paid for all time saved. The table is constructed so as to give a gradually disappearing percentage of bonus, as efficiency of production falls below the standard. During the past 11 years, this bonus table has been used extensively in our plants; first as an individual-effort incentive and, in later years, as a group incentive. A graphic chart showing the relation between efficiency and percentage of bonus is reproduced in Fig. 1.

The standard time of a single operation is determined by time-study. For this purpose the decimal stop-watch is used. By standard time is meant the time required by the average competent workman, taken repeatedly, over an extended period, eliminating lost and waste time. It will contain allowances for legitimate tool-changes and for rest and delay to the worker. It is our practice, for convenience, to designate standard time in decimal hours rather than in minutes. For this purpose a conversion table has been devised and is presented as Table 2, because the detail time-study is observed in decimal min-

TABLE 1—BONUS TABLE FOR PRODUCTIVE WORKERS

Percentage Efficiency	Bonus	Percentage Efficiency	Bonus	Percentage Efficiency	Bonus
75	1.0	117	40.4	159	90.8
76	1.6	118	41.6	160	92.0
77	2.2	119	42.8	161	93.2
78	2.8	120	44.0	162	94.4
79	3.4	121	45.2	163	95.6
80	4.0	122	46.4	164	96.8
81	4.6	123	47.6	165	98.0
82	5.2	124	48.8	166	99.2
83	5.8	125	50.0	167	100.4
84	6.4	126	51.2	168	101.6
85	7.0	127	52.4	169	102.8
86	7.6	128	53.6	170	104.0
87	8.2	129	54.8	171	105.2
88	8.8	130	56.0	172	106.4
89	9.4	131	57.2	173	107.6
90	10.0	132	58.4	174	108.8
91	11.0	133	59.6	175	110.0
92	12.0	134	60.8	176	111.2
93	13.0	135	62.0	177	112.4
94	14.0	136	63.2	178	113.6
95	15.0	137	64.4	179	114.8
96	16.0	138	65.6	180	116.0
97	17.0	139	66.8	181	117.2
98	18.0	140	68.0	182	118.4
99	19.0	141	69.2	183	119.6
100	20.0	142	70.4	184	120.8
101	21.2	143	71.6	185	122.0
102	22.4	144	72.8	186	123.2
103	23.6	145	74.0	187	124.4
104	24.8	146	75.2	188	125.6
105	26.0	147	76.4	189	126.8
106	27.2	148	77.6	190	128.0
107	28.4	149	78.8	191	129.2
108	29.6	150	80.0	192	130.4
109	30.8	151	81.2	193	131.6
110	32.0	152	82.4	194	132.8
111	33.2	153	83.6	195	134.0
112	34.4	154	84.8	196	135.2
113	35.6	155	86.0	197	136.4
114	36.8	156	87.2	198	137.6
115	38.0	157	88.4	199	138.8
116	39.2	158	89.6	200	140.0

TABLE 2—EQUIVALENTS OF MINUTES IN DECIMAL HOURS

Min.	Equivalent Hours	Rate of Production per Hour	Min.	Equivalent Hours	Rate of Production per Hour	Min.	Equivalent Hours	Rate of Production per Hour
0.01	0.000167	6000.0	0.34	0.00567	176.5	0.68	0.01133	88.2
0.02	0.000333	3000.0	0.35	0.00583	171.4	0.69	0.01150	87.0
0.03	0.000500	2000.0	0.36	0.00600	166.7	0.70	0.01167	85.7
0.04	0.000667	1500.0	0.37	0.00617	162.2	0.71	0.01183	84.5
0.05	0.000833	1200.0	0.38	0.00633	157.9	0.72	0.01200	83.3
0.06	0.001000	1000.0	0.39	0.00650	153.8	0.73	0.01217	82.2
0.07	0.001167	857.1	0.40	0.00667	150.0	0.74	0.01233	81.1
0.08	0.001333	750.0	0.41	0.00683	146.3	0.75	0.01250	80.0
0.09	0.001500	666.7	0.42	0.00700	142.9	0.76	0.01267	78.9
0.10	0.001670	600.0	0.43	0.00717	139.5	0.77	0.01283	77.9
0.11	0.001830	545.5	0.44	0.00733	136.4	0.78	0.01300	76.9
0.12	0.002000	500.0	0.45	0.00750	133.3	0.79	0.01317	75.9
0.13	0.002170	461.5	0.46	0.00767	130.4	0.80	0.01333	75.0
0.14	0.002330	428.6	0.47	0.00783	127.7	0.81	0.01350	74.1
0.15	0.002500	400.0	0.48	0.00800	125.0	0.82	0.01367	73.2
0.16	0.002670	375.0	0.49	0.00817	122.5	0.83	0.01383	72.3
0.17	0.002830	352.9	0.50	0.00833	120.0	0.84	0.01400	71.4
0.18	0.003000	333.3	0.51	0.00850	117.6	0.85	0.01417	70.6
0.19	0.003170	315.8	0.52	0.00867	115.4	0.86	0.01433	69.8
0.20	0.003330	300.0	0.53	0.00883	113.2	0.87	0.01450	69.0
0.21	0.003500	285.7	0.54	0.00900	111.1	0.88	0.01467	68.2
0.22	0.003670	272.7	0.55	0.00917	109.1	0.89	0.01483	67.4
0.23	0.003830	260.9	0.56	0.00933	107.1	0.90	0.01500	66.7
0.24	0.004000	250.0	0.57	0.00950	105.3	0.91	0.01517	65.9
0.25	0.004170	240.0	0.58	0.00967	103.4	0.92	0.01533	65.2
0.26	0.004330	230.8	0.59	0.00983	101.7	0.93	0.01550	64.5
0.27	0.004500	222.2	0.60	0.01000	100.0	0.94	0.01567	63.8
0.28	0.004670	214.3	0.61	0.01017	98.4	0.95	0.01583	63.2
0.29	0.004830	206.9	0.62	0.01033	96.8	0.96	0.01600	62.5
0.30	0.005000	200.0	0.63	0.01050	95.2	0.97	0.01617	61.8
0.31	0.005170	193.5	0.64	0.01067	93.8	0.98	0.01633	61.2
0.32	0.005330	187.5	0.65	0.01083	92.3	0.99	0.01650	60.6
0.33	0.005500	181.8	0.66	0.01100	90.9	1.00	0.01667	60.0
			0.67	0.01117	89.6			

utes, while the standard time is converted into hours.

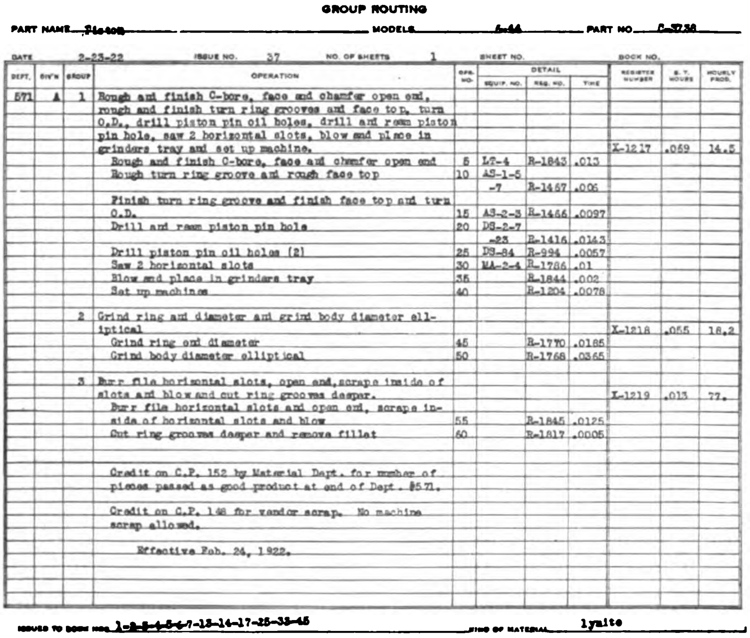
Group standard-time is obtained by totalling the individual standard-times for all operation embraced by a group. It thus always appears as one-man hours, independently of how many workers may subsequently be assigned to a designated group. All detail operation standard-times and the total group standards are listed on a shop routing for ready reference as shown in Fig. 2.

Application of Group Standard-Time

To obtain a clear idea of the application of group standard-time and its use as a basis for a wage-incentive plan, let us consider a single production line manufacturing and assembling pistons in an automobile-engine plant. This unit has its operations arranged in sequence, in what is known as a progressive line of manufacture. It starts with the rough-machining of the casting and ends with the finished product with piston-rings inserted. The production line includes both machine and bench-assembly operations. Parts move without a break from one operation to the next in a steady flow. Some operations may be done by a single workman; others may have several workers in parallel, depending on the volume of production and with what detail it is practicable to subdivide operations.

It is proposed to keep no check on quantities passing intermediate operations or on those completed by individual workers on operations in parallel but to give credit for finished pistons passing final inspection. This particular production line may be located in a department also producing connecting-rods or any other line of manufacture. Such a line embraces a production unit of the factory, and is technically known under the group plan as a *division*. Its workers are primarily interested in the production of pistons; they know little or nothing about conditions on the connecting-rod line and should not be grouped with it.

A study of the piston-line division indicates that there are two distinct classes of machine work, besides bench operations. The machine operations are lathe turning and grinding. There are thus three major operations, and the workers group themselves naturally into these three classes of work. It is thus seen that a division may consist of any number of distinct groups. They are bound together because they receive credit only for the finished product at the end of the division, while they are distinct and independent so far as efficiency measurement and bonus reward are concerned. A group is debited with the actual man-hours of the workers in the group and is credited with the group standard-hours multiplied by the number of pieces passing the division. That is, the quantity count is not taken at the group operations but at the end of the division of work. The ratio between standard hours and actual hours constitutes the efficiency measurement for which a bonus percentage is paid. For convenience, the entire pay-period of 1 or 2 weeks is run on a cumulative basis, and bonus is computed on the average efficiency attained for the period.


GROUP ROUTING

PART NAME Piston MODELS 6-44 PART NO. C-3738

DATE 2-23-22 ISSUE NO. 37 NO. OF SHEETS 1 SHEET NO. BOOK NO.

Dept.	Div'n	Group	Operation	Opr. No.	Detail: Equip. No.	Detail: Reg. No.	Detail: Time	Register Number	S. T. Hours	Hourly Prod.
571	A	1	Rough and finish C-bore, face and chamfer open end, rough and finish turn ring grooves and face top, turn O.D., drill piston pin oil holes, drill and ream piston pin hole, saw 2 horizontal slots, blow and place in grinders tray and set up machine.					X-1217	.069	14.5
			Rough and finish C-bore, face and chamfer open end	5	LT-4	R-1843	.013			
			Rough turn ring groove and rough face top	10	AS-1-5					
					-7	R-1467	.006			
			Finish turn ring groove and finish face top and turn O.D.	15	AS-2-3	R-1466	.0097			
			Drill and ream piston pin hole	20	DS-2-7					
					-23	R-1416	.0143			
			Drill piston pin oil holes (2)	25	DS-84	R-994	.0057			
			Saw 2 horizontal slots	30	MA-2-4	R-1786	.01			
			Blow and place in grinders tray	35		R-1844	.002			
			Set up machines	40		R-1204	.0078			
		2	Grind ring and diameter and grind body diameter elliptical					X-1218	.055	18.2
			Grind ring end diameter	45		R-1770	.0185			
			Grind body diameter elliptical	50		R-1768	.0365			
		3	Burr file horizontal slots, open end, scrape inside of slots and blow and cut ring grooves deeper.					X-1219	.013	77.
			Burr file horizontal slots and open end, scrape inside of horizontal slots and blow	55		R-1845	.0125			
			Cut ring grooves deeper and remove fillet	60		R-1817	.0005			
			Credit on C.P. 152 by Material Dept. for number of pieces passed as good product at end of Dept. #571.							
			Credit on C.P. 148 for vendor scrap. No machine scrap allowed.							
			Effective Feb. 24, 1922.							

ISSUED TO BOOK NOS. 1-2-3-4-5-6-7-13-14-17-25-33-45 KIND OF MATERIAL lynite

Fig. 2—Typical Routing Sheet Giving the Standard Time and the Tool-Group Standards for the Detail Operations on a Particular Part

Obviously, for a given product, the fewer actual man-hours there are in a group, the higher the efficiency of production will be and the greater the per cent of bonus earnings. The incentive is, therefore, to obtain a high rate of production per man per hour consistent with good quality of product.

Each member of a group receives the same percentage of bonus at the end of the pay period, but it is computed on his wage earned while assigned to the group; and the total amount earned by each worker will, therefore, depend on his hourly base-rate and number of hours worked. No job tickets are used. A shop timekeeper will handle from 300 to as high as 600 group workers. A list of employes is tabulated daily for each group, and elapsed time is taken at the end of the day from the entrance time-clock. If a worker is transferred out of or into a group, a transfer slip noting the time is recorded by the timekeeper and the elapsed hours are charged accordingly. Standard-hours credit is obtained from the finished inspection reports of quantities, multiplied by the group standard-time. The forms used in this connection are reproduced in Fig. 3. The method of tabulating group records daily in the time office, and of computing bonus, is indicated by the form shown in Fig. 4.

A group may produce various parts having different time-standards, and therefore different direct-labor costs and, since we use no individual elapsed-time job-tickets, labor costs are computed from the group cost per standard-hour. For given base-rates, the labor cost will be constant per standard-hour and per piece for all efficiencies above 100 per cent. If the average efficiency falls to 90 per cent, the labor cost will increase less than 2 per cent.

In the above discussion, we have assumed that the production line has its operations arranged in sequence. Such lines are, of course, ideal for the grouping of workers. But in the class of factories we are considering, there may be anywhere from 5 to 40 per cent of the direct-labor operations that cannot be arranged progressively. Such departments usually consist of miscellaneous small machines, automatics, grinders, or minor

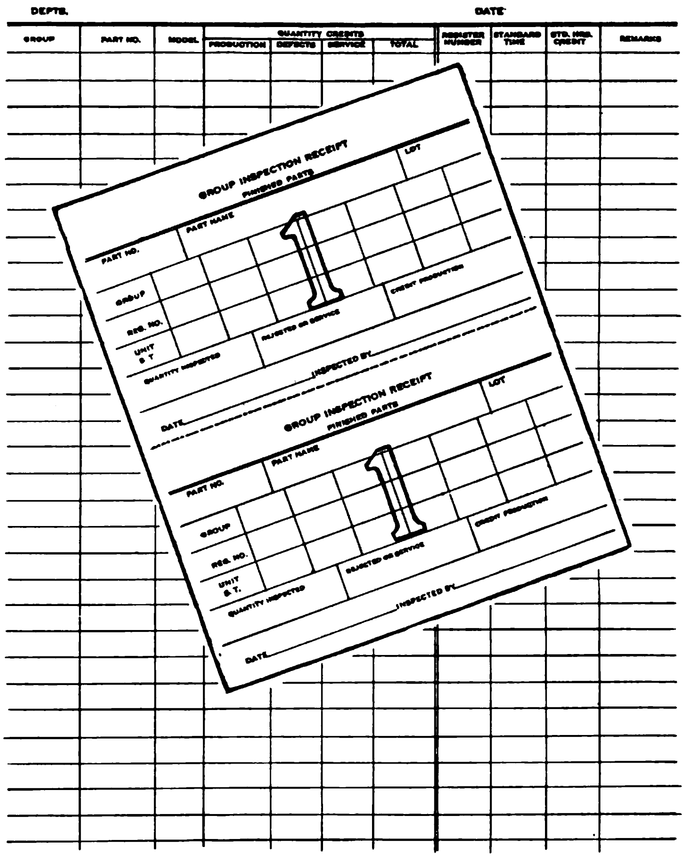


FIG. 3—FORMS EMPLOYED IN CONNECTION WITH THE SECURING OF THE STANDARD-HOURS CREDIT FOR A GROUP OF OPERATORS

bench operations. However, it has been our practice up to the present to group such departments according to similar operations, being careful to retain the personal interest of the worker in the group production. Credit is given only for finished parts as they leave the department. So far, such grouping has been entirely successful. If there were any special departments of work in

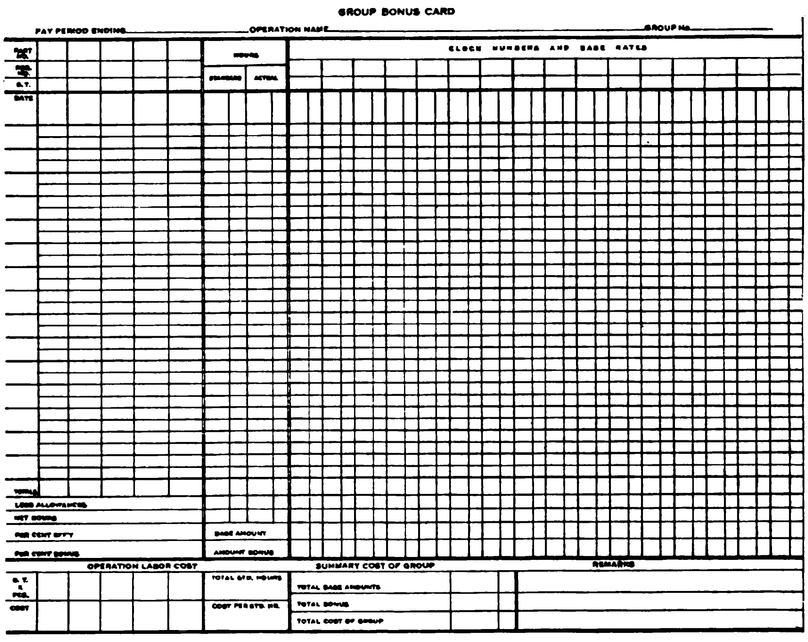


FIG. 4—CARD ON WHICH THE GROUP RECORDS ARE TABULATED DAILY IN THE TIME OFFICE AND THE BONUS COMPUTED

TABLE 3—WAGE TABLE FOR PRODUCTIVE WORKERS; COMPUTED FROM TABLE 1

Employe's Hourly Base Rate	Hourly Earnings at 85 to 130 Per Cent Efficiency									
	85	90	95	100	105	110	115	120	125	130
$0.25	$0.268	$0.275	$0.288	$0.300	$0.315	$0.330	$0.345	$0.360	$0.375	$0.390
0.26	0.278	0.286	0.299	0.312	0.328	0.343	0.359	0.374	0.390	0.406
0.28	0.300	0.308	0.322	0.336	0.353	0.370	0.386	0.403	0.420	0.437
0.30	0.321	0.330	0.345	0.360	0.378	0.396	0.414	0.432	0.450	0.468
0.32	0.342	0.352	0.368	0.384	0.403	0.422	0.442	0.461	0.480	0.499
0.34	0.364	0.374	0.391	0.408	0.428	0.449	0.469	0.490	0.510	0.530
0.35	0.375	0.385	0.403	0.420	0.441	0.462	0.483	0.504	0.525	0.546
0.36	0.385	0.396	0.414	0.432	0.454	0.475	0.497	0.518	0.540	0.562
0.38	0.407	0.418	0.437	0.456	0.479	0.502	0.524	0.547	0.570	0.593
0.40	0.428	0.440	0.460	0.480	0.504	0.528	0.552	0.576	0.600	0.624
0.42	0.449	0.462	0.483	0.504	0.529	0.551	0.580	0.605	0.630	0.655
0.44	0.471	0.484	0.506	0.528	0.554	0.581	0.607	0.634	0.660	0.686
0.45	0.482	0.495	0.518	0.540	0.567	0.594	0.621	0.618	0.675	0.702
0.46	0.492	0.506	0.529	0.552	0.580	0.607	0.635	0.662	0.690	0.718
0.48	0.514	0.528	0.552	0.576	0.605	0.634	0.662	0.691	0.720	0.749
0.50	0.535	0.550	0.575	0.600	0.630	0.660	0.690	0.720	0.750	0.780
0.52	0.556	0.572	0.598	0.624	0.655	0.686	0.718	0.749	0.780	0.811
0.54	0.578	0.594	0.621	0.618	0.680	0.713	0.745	0.778	0.810	0.812
0.55	0.589	0.605	0.633	0.660	0.693	0.726	0.759	0.792	0.825	0.858
0.56	0.599	0.616	0.644	0.672	0.706	0.739	0.773	0.806	0.840	0.874
0.58	0.621	0.638	0.667	0.696	0.731	0.766	0.800	0.835	0.870	0 905
0 60	0.642	0.660	0.690	0.720	0.756	0.792	0.828	0.864	0.900	0.936

such factories that could not be grouped to advantage, there would be no objection to leaving them on an individual-incentive basis or even on a straight day-rate basis.

Indirect labor has been grouped extensively wherever a "community of interest" can be maintained between workers. They must have a common interest in the results of their own efforts. Storeroom labor, unloading materials from cars, boxing and loading automobiles for shipment and similar classes of work where the effort is measurable, have been grouped with very good results.

In summarizing our experience during the past 4 years with the group-bonus plan, it should be borne in mind that the particular feature involved is the principle of grouping employes and not the wage-incentive table that happened to be selected. Very likely any one of several incentive plans could have been used in connection with grouping and have produced satisfactory results. This one was selected because we thought it would be more adaptable to changing factory conditions than a system of fixed-group piece-rates having their value in dollars rather than in time. It also offers the same incentive to high production as could be obtained from piece rates.

Although the primary purpose of grouping was to simplify the factory system and to reduce the amount of clerical detail, it developed that there were many advantages from an operating standpoint. Much less material is tied-up in process, and full advantage can be taken of the mechanical arrangement of progressive lines whereby parts are made to flow from one operation to the next in a steady stream through single or parallel operations. Under the continuous-flow method of production, the checking of quantities after individual operations becomes difficult, if not impracticable; so, the logical method seems to be to count from the end of the line and credit groups of workers. Nearly every such production line has its "neck of the bottle" or several of them. If these can be speeded up, the whole line benefits.

One of the early advantages noted was the speeding-up, or elimination, of slow workers. Hence, it has been our experience that more production per man-hour has been obtained under grouping than under a previous individual-incentive plan. From the viewpoint of factory operation it has meant the elimination of job tickets and elapsed-time records for group operations. This has meant saving in clerical detail. It has added to productive time, because there are always some delays if employes are required to keep count on quantities, obtain job tickets or furnish information to shop checkers.

CONDITIONS FAVORABLE TO GROUPING

It may be taken for granted that factory workers cannot be grouped indiscriminately, or under all sorts of conditions, with any hope of success. The individual incentive must not be lost sight of. If a group worker feels that others will carry his burden whether he exerts himself or not, there is no incentive for him to put forth his best efforts. Consequently, conditions must be such as will enable the individual to realize that the results are going to depend on his own interest in the work. Progressive lines of sequence operations, grouped according to major operations, are ideal for this purpose. On the other hand, where operations are not progressive but have a community interest, equally good results are obtained. As an example, a group of 20 men snagging miscellaneous castings on an average tonnage basis gave a remarkable performance because they had one interest, the output in pounds of castings per man per hour. Storeroom attendants, caring for and delivering materials, were successfully grouped by placing the incentive on the basis of hours time per 100 units produced by the factory.

The essentials for grouping are that workers must be placed close together as a team, and also be able to see the results of their own efforts. Operations should be largely repetitive. The results of each day's group-work should be made known as early as possible the following day to maintain the individual interest. The best results are obtained by tabulating this information cumulatively from day to day over a pay period of 1 to 2 weeks. It is essential that the computation of earnings be made so simple that the average factory worker can figure out his own earnings quickly, just as he would on straight day's-work. For this purpose a wage table, such as Table 3 but more in detail, is posted in the shop. Most employes use a copy of the bonus scale such as is shown in Table 1, and compute earnings directly from the average group efficiency.

SUMMARY

The group plan is primarily applicable to repetitive work arranged in progressive production lines of sequence operations, but it is also applicable to non-progressive operations where the individual interest can be centered on definite results per man-hour of labor. Like any other wage-incentive plan, it has for its main objective the speeding-up of the production rate per employe; that is, *intensive production*.

It is being used because it simplifies the factory detail where a high rate of production is desired on repetitive work, and where a wage incentive is employed as the means of obtaining intensive production. Its application has gained favor among factory executives because it speeds-up production as compared with the individual plan on the same work, ties up less material in process and minimizes clerical detail in the factory.

To have a group plan remain in successful operation over an indefinite period, it is necessary to maintain the interest of the individual worker in the group effort. It must be simple for computation of individual earnings, flexible so as to meet changing factory conditions and easily understood by each group member.

The Control of Operating Tool and Supply Cost

By F. A. MANCE[1]

DETROIT PRODUCTION MEETING PAPER

BY allotting to each department the precise amounts of tools and supplies that are to be used for each operation a control is established throughout the factory that must accord with the number of cars built. Over a period of 4 years a system of this kind in the plant cited has shown a decrease in the cost of renewals of 71 per cent. The methods of procedure are given for instituting such a system and for charging off the costs of tools used in one department when the tools are "salvaged" from other departments. A preferential list of sources of supply is recommended, which should contain the names of firms supplying the tools that have given the most satisfactory results in actual use.

A CONTROL over expenditures for perishable tools and operating supplies has been successfully established by rating or allotting the amounts to be used by each department for every operation. By this allotment we are able to regulate the amounts of tools and supplies that are used throughout the factories in accordance with the number of cars built. Each item is priced and extended, and the aggregate is totaled. The total represents the amount of money allotted to the department to produce a definite number of cars in a given period of time. In figuring departmental percentages of efficiency this amount is included with labor and production.

This system has been in operation for nearly 4 years and during this time the cost of renewals of perishable tools has decreased approximately 71 per cent. It is admitted, of course, that some of this reduction was due to a decrease in the cost of the tools.

To anyone interested in adopting this method we would suggest the following procedure: Start with the selection of a competent and aggressive tool-trouble man for making the survey. Provide work sheets on which to list the department, the part number, the operation, the description of both tools and supplies and the amounts allotted for any given number of cars per month. The allotment can be based on the actual withdrawals over a given previous period, after eliminating the waste on account of abuse and improper methods of use where instances of this kind are found. By this we mean that the allotment should be based on the amounts that should be used. These work-sheets are afterward to be used in making up a standard form in quadruplicate. The standard form should contain additional information as to price, the extensions and the total, as previously explained. The standard allotment-sheets when completed are distributed to the foreman interested, the tool or supply stores, the superintendent of the plant and the supervisor of tools or the methods and standardization department.

Deliveries from stores are made upon approved requisitions, provided the article wanted has been previously allotted and appears on the allotment sheets. Departmental operating tool and supply reports are made up for every 10-day period by the supervisor of stores and show the amounts withdrawn during that period. They are then forwarded to the accounting department for pricing. From copies of these sheets each foreman is checked as to the amount he is running over or under his allotment and he is allowed to see these copies during each period in order to acquaint himself with the cost of the articles that he uses. It is amusing at times to listen to the protests registered by the foremen against the prices paid for some of the more expensive items, but it gives one the satisfaction of knowing that he is vitally interested at all times in keeping down his operating costs.

Salvaged or restored tools are carried in stock and are given out on requisition in the same manner as new tools, with the exception that the requisition is stamped with the word "salvage" in red ink. This signifies that there is to be no charge against the department drawing out this material. When a drill has become too short for use in one department it is turned in as "salvage" and reconditioned for use in another department. This is also done with cutters, reamers and grinding wheels. The allotment sheets show where the salvage should be used and each item so salvaged is listed "no charge."

The department first drawing out a new tool bears the entire cost of the tool. In the case of special tools made to conform to a blueprint, the tool-design department notes on the tracings whether the tools can be salvaged from either a standard or another special tool of similar design, and these tools are also listed "no charge" on the allotment sheets.

The tool-salvage department keeps in touch daily with the various department foremen, advising them of available salvage tools, especially when these are in addition to the tools specified on the allotment sheets. The use of salvage tools so far this year has amounted to $1 per car. This sum represents what the cost would have been had we purchased new tools. The amount of money spent to recondition the tools amounted to 30 per cent of the original cost. This 30 per cent is charged off as expense, and is pro-rated over the entire portion of the plant that is benefited by the use of the tools.

One important factor that has a direct bearing on tool cost is the listing of sources of supply. This list is made up by the methods and standardization department and contains the names of the firms whose tools have given us the best results in actual use over a period of from 1 to 4 years.

The smallest possible number of firms is listed that is consistent with good business, and the relative merits of the firms can be determined from the order of their listing. When only one firm is given, no other satisfactory source has been found. When several are grouped, each firm is considered as good as any other of that particular group. Changes and additions are made by the methods and standardization department only when satisfactory

[1] Production department, Studebaker Corporation of America, Detroit.

results have been obtained from comparative tests and from use. This approved list of sources is distributed to the different purchasing departments for reference in buying.

So vast an amount of operating supplies, such as grease, solder, paints, oils, enamels, and gasoline, is used in the automotive industry that no effort should be spared to eliminate waste. On these particular items our allotment has accomplished a considerable saving in the quantities used from year to year.

In connection with the allotment of tools, we have begun recently a study of the life of perishable tools. Among drills this study consists, first, of a microscopic inspection of the steel structure to determine good heat-treatment. Various sizes of drill that show proper heat-treatment are selected and used for drilling steel forgings of a given Brinell hardness. The actual amount of metal removed per grinding is noted, and the drills are compared as regards life per inch of flute length and number of feet of metal removed.

Several charts have been made and an average has been computed for use as a reference in determining the actual life of drills for a given number of feet of steel of a certain range of Brinell hardness.

In all non-productive departments the expenditures for operating supplies are controlled by a fixed sum or budget and are based on car production in the same manner as in the productive departments.

The tool and supply stores are controlled by requisitioning only such items as appear on the allotment sheets. All purchase requisitions from stores must state the amount used in the previous 3 months, the approximate total cost, the amount in stock at all plants and whether other material can be substituted or salvaged. It is essential that the man who supervises the allotment should check all requisitions before they are approved.

OCTOBER COUNCIL MEETING

The meeting of the Council held in Detroit on Oct. 25 was attended by President Bachman, Past-President Beecroft, Vice-Presidents Watts and Young, Councilors Brush, Crane, Strickland and Scott; and W. A. Chryst and A. J. Scaife.

A preliminary financial report for the fiscal year ended Sept. 30 last was considered. This showed that the deficit for the year was about $15,000. This deficit is approximately half as great as that specified in the budget. The result of the year's operations was considered very satisfactory under the circumstances, the deficit being practically equal to additional expense incurred during the last fiscal year for activities consequent upon the enlarged scope of the Society's work. The total income for the last fiscal year was $248,000 this being $16,000 more than contemplated by the budget for the period, and $5,200 more than the actual income during the fiscal year ended Sept. 30, 1921. The net assets of the Society on Sept. 30 amounted to $122,000.

It was decided to issue a 1923 Roster of the Society in the same form as heretofore, copies thereof being furnished without charge to members requesting them.

A long discussion was had of proposed programs for the 1923 Annual and Summer Meetings of the Society. The chief difficulty in scheduling sessions is to allow enough time for discussion and still have enough of the available desirable papers presented without holding too many sessions at a meeting. It was decided definitely to discontinue the Carnival that has been held in recent years in connection with the Annual Meeting. The sentiment at the Council meeting was not in favor of a boat trip for the Summer Meeting.

Thirty-eight applications for individual membership were approved. The following transfers in grade of membership were approved: Junior to Member, R. E. Bissell, F. L. Creager; Associate to Member, W. J. Bryan, Joseph V. Petrelli, E. Burnell and Thomas A. Clark.

It was reported that up to Oct. 24, 1922, 611 applications for membership had been received during 1922, as compared with 625 received during the first 10 months of 1921, and 1033 during the first 10 months of 1920. A material gain in membership will be shown at the beginning of 1923. At the present time the net membership of the Society is about the same as it was at the corresponding period of 1921, the Council having recently dropped a large number of members for non-payment of dues.

The following subjects were assigned to the respective Divisions of the Standards Committee:

Engine Testing Forms—Motorboat Division

Motorboat Trial Performance Forms—Motorboat Division

Isolated Electric Lighting-Plant Storage-Battery Rating—Storage Battery Division (reassignment)

It was decided not to assign the subject of engine-bearing oil-grooves for consideration in connection with standardization.

THE PRODUCTION MEETING

(*Concluded from p.* 390)

members and the service of its local office. The meetings, visits and dinner were conducted punctually and smoothly because of the whole-hearted support of the entire group of members actively interested in the arrangements.

1923 Production Meeting

There will be a 1923 Production Meeting of the Society; that is a certainty. The spontaneous reception and representative attendance given the Detroit Meeting assure the closest attention being given to the production phases of the industry by the Society at National and Sections Meetings. Out of this new activity will come a better understanding between the man who designs and the man who produces. This can only mean a further strengthening of our industry. No one who attended the Detroit Production Meeting can have failed to secure much of value from it.

Experience Notes from a Production Notebook

By H. J. Crain[1] and J. Brodie[1]

Detroit Production Meeting Paper *Illustrated with* Photographs and Drawings

WHILE investigating the sources and causes of noise in automobiles during an extensive connection with one of the largest automobile companies, the authors recorded their experiences in the shop in the form of notes. Some of these are offered with a view to stimulating the discussion of the subject and with the hope that additional information will be brought out by an exchange of ideas, particularly on the problem of eliminating gear-noises. In many cases they found that noise was caused by failure to allow sufficient clearance for an adequate oil-film. And it was noted frequently that when one noise had been located and silenced another appeared that had not been apparent before. The topics that have been considered include the running-in of brake-bands, engine knocks, oil-pump gear-noise and that of gears in general, the clearances of ball bearings, backlash, and rear-axle bevel-gears.

THE experiences recorded in this paper have not been selected in accordance with a specific plan. No attempt has been made to cover any particular subject fully or to arrange the different descriptions with regard to a related sequence. It is possible that some of the experiences have been encountered or the methods have been used by other production executives but we believe that knowledge of these methods is not general and that it will be of interest to factory men. We hope that the discussion will bring out additional information on some of the matters treated, particularly on the perplexing problem of eliminating gear noise.

Running-in Brake-Bands

The increasing congestion on city streets and the seriousness of the automobile accident and collision situation should be convincing evidence of the need of proper adjustment of motor-car brakes. It would seem important that cars should be shipped from the factories with the brakes seated, and adjusted to overcome the rapid wear that usually occurs in driving the first few hundred miles. This rapid wear is caused by the ironing or smoothing of the brake-lining surface until the high spots have been worn down to the level of the rest of the lining face. It may be due also to slight imperfections in the contour of the brake-band. Figs. 1 and 2 illustrate two motor-driven machines designed and built by the Packard company for the purpose of running-in brake-bands. The machine shown in Fig. 1 handles the internal or expanding brake and that in Fig. 2 the external or contracting brake. The brake-drums rotate at a speed of approximately 1000 r.p.m. in both cases. The drums are cooled by water, circulated about the peripheries of the drums, so that the temperatures are never excessive. Pressure is exerted on the brake-bands by a weighted lever, which can be seen clearly in Fig. 1, the weight being adjusted so that the pressure is only great enough to assure a full bearing of the band on the drum.

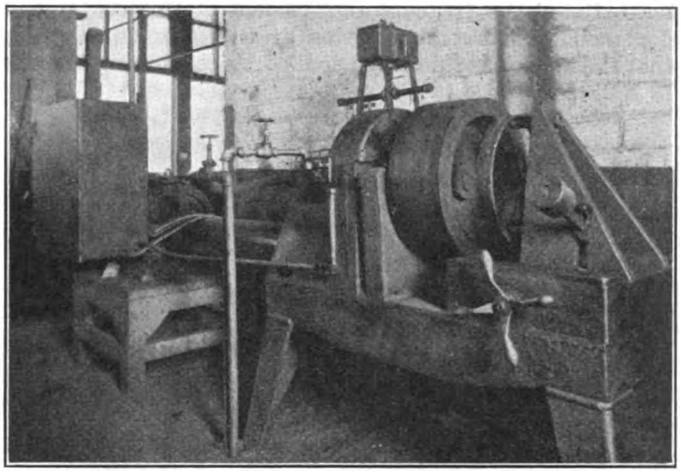


Fig. 1—Motor-Driven Machine for Running-In Internal Brake-Band Assemblies

Each band is run for about 1 min. The two machines are located so that the same operator can handle both; the band of one being run-in while the operator is loading the other. It will be found that the bands acquire a polished surface on these machines, and that the irregularities sometimes existing around the rivet-holes and throughout the lining surface are smoothed-out. By taking this precaution at the factory the maximum brake efficiency is attained at the beginning of operation of the car, and the adjustments usually required in a new car after a few days of service are unnecessary.

The importance of accuracy in grinding a piston skirt is recognized by all production men. The center, shown in Fig. 3, was originally used in the Packard shops for centering and driving the pistons during the external grinding operation. Excessive wear of the surface *a*

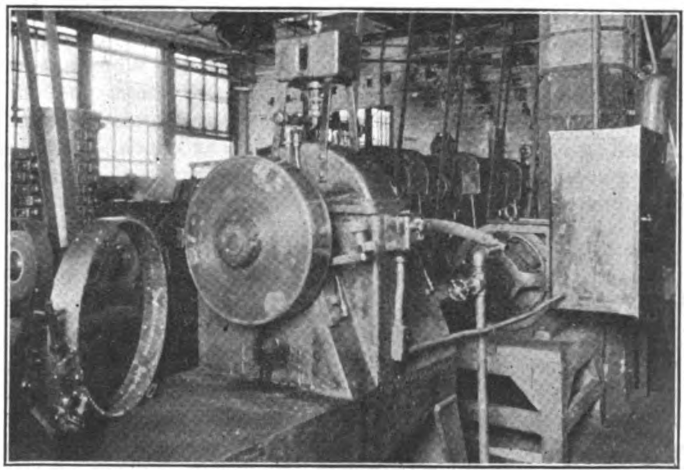


Fig. 2—A Special Machine Developed by the Packard Company for Running-In External Brake-Band Assemblies

[1] Production department, Packard Motor Car Co., Detroit.

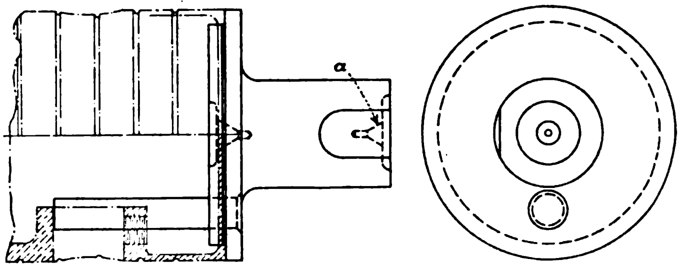


FIG. 3—FORM OF CENTER ORIGINALLY USED IN THE PACKARD SHOPS FOR CENTERING AND DRIVING THE PISTONS DURING THE EXTERNAL GRINDING OPERATION

necessitated the frequent replacement of this center and demanded constant supervision by the foreman in order that the work should not be spoiled by continuing the use of a center that had passed the permissible stage of wear. The study given to this small but puzzling problem has resulted in the adoption of the center shown in Fig. 4. In this instance the thrust and driving load are taken by a taper-roller bearing of heavy load-capacity, the wear is distributed over a very large surface, lubrication is easily maintained and the life of the center is greatly prolonged. This design has proved very successful, and, no doubt, other tool designers could apply it to advantage. This particular center is used with Brown & Sharpe Nos. 12 and 14 external grinding-machines.

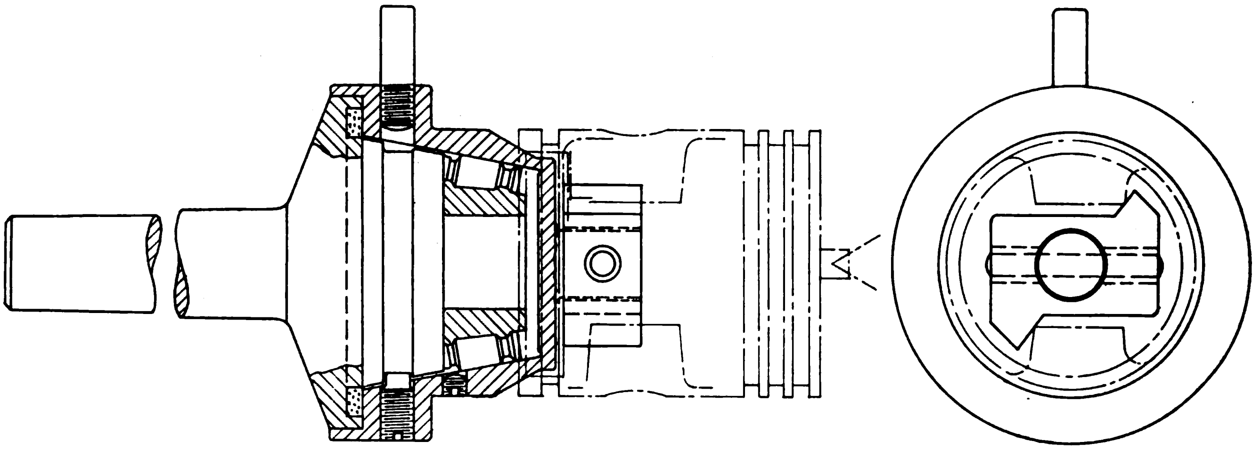


FIG. 4—A NEWER FORM OF CENTER IN WHICH THE THRUST AND THE DRIVING LOAD ARE TAKEN BY A TAPER ROLLER BEARING

A PUZZLING ENGINE KNOCK

All production and inspection departments have had the displeasure of running down peculiar engine knocks. The following note from Packard experience may shed some light on this trouble. A few years ago when a new model was started through the Packard shops the engines of the first run received at the test-stands were all found to have a perceptible piston slap or knock. Numerous remedies were tried but eventually the real cause of the noise was found largely through accident.

The click came only at the time of the explosion. Investigation revealed the fact that the valves were not centering properly in the conical surface in the cylinder. The condition is shown in exaggerated form in Fig. 5. It was found that the tool used to form the valve-seat was centered by a spindle inserted in the valve-stem guide. This spindle was too much undersize and allowed the tool to float just enough to throw the conical seat out of alignment with the valve-stem guide. As a result, the valve-spring would not bring the valve fully into the seat; the valve would hang on one side of the valve-seat until the explosion snapped the valve into the seat with a very noticeable click. Of course this part of the noise was then obviated without difficulty.

But this correction did not stop all the noise. A more annoying knock was eventually found to come from the piston-rings, which were of the diagonal-cut type with a slight clearance between the ends. As the explosion-pressure reached the rings they were compressed and their ends snapped. Filing several grooves in the upper edge of the top ring would let part of the explosion in behind the ring, expand the ring and overcome the slap until the grooves had filled with carbon. A ring that slaps in the manner described generally shows bright polished ends. When the proper end-clearance had been determined by experiment, the noise ceased. The rings in general use to-day have overlapping joints; the end clearance can be very large and, of course, this trouble is not encountered.

OIL-PUMP GEAR NOISE

Oil is circulated in Packard engines by a gear-pump similar to that illustrated in Fig. 6. When this particular design was first adopted it was found to produce a very irritating noise, which sounded like the blades of a fan striking a sheet of paper. Naturally this was

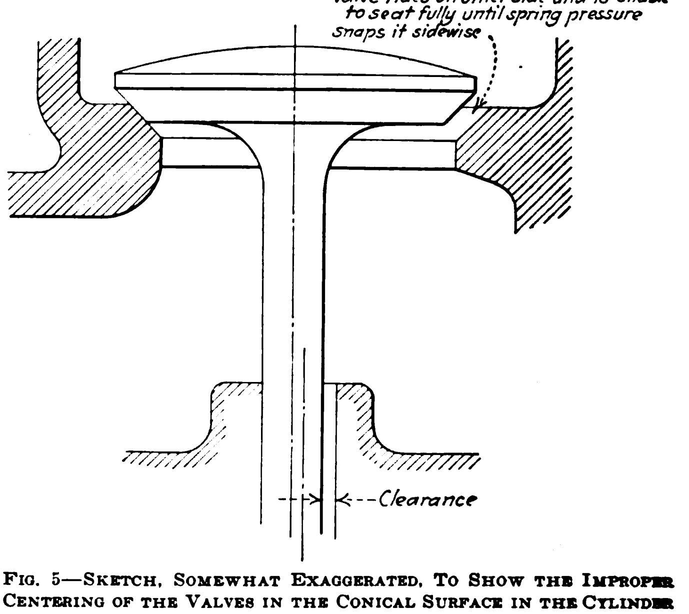


FIG. 5—SKETCH, SOMEWHAT EXAGGERATED, TO SHOW THE IMPROPER CENTERING OF THE VALVES IN THE CONICAL SURFACE IN THE CYLINDER

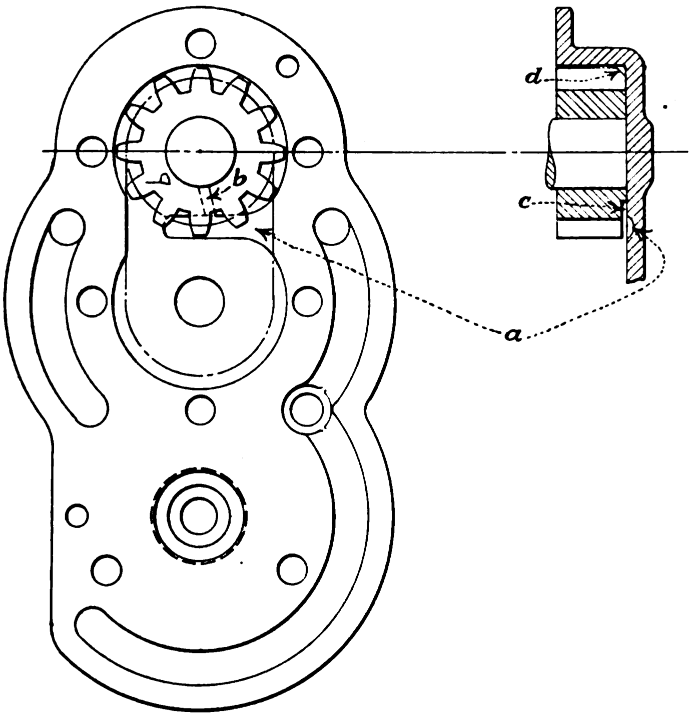


FIG. 6—THE GEAR-PUMP THAT CIRCULATES THE LUBRICATING OIL IN THE PACKARD ENGINE

attributed to imperfections in the gears. In the experiments made to abate this nuisance tooth-forms and pressure-angles were varied, and helical and herring-bone gears were fitted, but the clatter persisted. All degrees of backlash were tried but without avail. It was noticed that a certain run of pumps were more quiet than the others. These were inspected carefully to ascertain what variation was responsible for the lessening of the noise. The only difference found was a slight relief on the lower face of the idler gear. At *a*, in Fig. 6, is shown a feeder channel that is cut in the base of the pump for the purpose of carrying oil to the idler bearing through the cross channel *b*. Note that this channel is open to the pressure side of the pump but ends at the point where tooth contact ceases on the suction side. It was found that when the tooth corners were beveled, as shown at *c*, the noise was reduced. The possible effect of these changes was the basis of a careful study, which resulted in the discovery of the real source of the noise. Both these schemes eliminated the sharp cutting off of the oil stream that would naturally attempt to escape at *d* from the pressure side to the suction side of the pump. By relieving the pressure in the groove *a* the oil was not able to spurt against the tooth faces and rattle the unloaded idler gear in the backlash space in the driving gear. When, as an experiment, the groove *a* was filled with solder, the altered pump became quiet. The design of the pump was changed, the groove *a* was omitted, and no further trouble was experienced. This case is cited as an example because it indicates that gear noises are not always attributable to the gear-teeth themselves.

GEAR-NOISE INVESTIGATIONS

The production and engineering staffs of the Packard Motor Car Co. have been studying the matter of gear noises for many years. This work is still being carried on but no panacea has been definitely discovered for gear troubles. Each case seems to have its peculiarities and to require special modifications to become silent. Alterations that are effective in one case might not be effective in another that to all appearances is similar. It is more than likely that the study being given to gear-tooth wear and noise in our own and other industries will lead eventually to a better understanding of the fundamental causes of the difficulties. For the present, however, it is only possible for us to exchange experiences for the common benefit of those who are interested.

A large number of investigations made over a period of years have led us to believe that gear noise does not always originate in variations of the gears themselves. Such variations undoubtedly contribute to the gear growl or chatter but the noise can often be cured or dampened by an alteration of another part. We have concluded that the proper mounting of the gears on rigid shafts is a paramount requirement if noise is to be avoided. The most perfect tooth-forms, ground, shaped or milled, will not run quietly if they are carried on shafts that spring or are not in perfect alignment. It is assumed that this essential fact is recognized in the engineer's design of transmissions and axles. Given a properly designed mounting, it is the factory man's problem to reduce noise. The factors controlling it are largely independent of blueprints. Drawings can only give the characteristics of gear-teeth; the production man must see that the actual contours and allowable variations keep them within an acceptable range of quietness.

The front end of a typical transmission is shown in Fig. 7. Particular attention is directed to the mounting of the main drive-gear *a*, in which it will be seen that this is carried on a roller bearing, the inner race of which is formed by the shaft itself and the outer race is mounted in the transmission case. In the final inspection of a certain model at the Packard factory it was noticed that the degree of gear noise varied from very

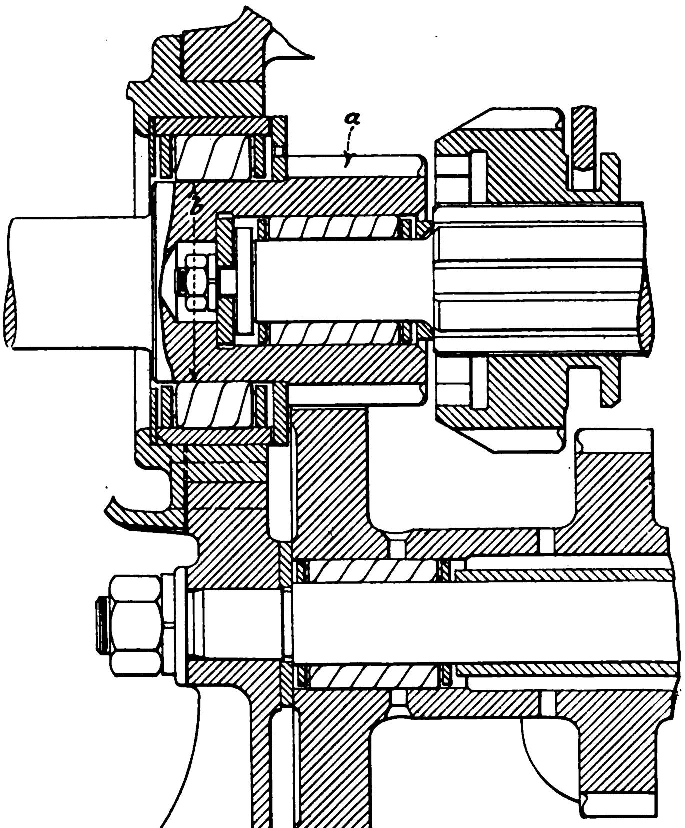


FIG. 7—FRONT END OF A TYPICAL TRANSMISSION SHOWING HOW THE MAIN DRIVE GEAR IS CARRIED ON A ROLLER BEARING

quiet to objectionably loud. Attention, naturally, was centered on the noisy gears. These were returned to the transmission department and torn down for careful inspection, adjustment and reassembling. Invariably the inspection revealed gears, bearings and shafts that were as near perfection as it seemed possible to approach. This led to the assembling of special gears in which perfection was carried to the utmost degree, a state far be-

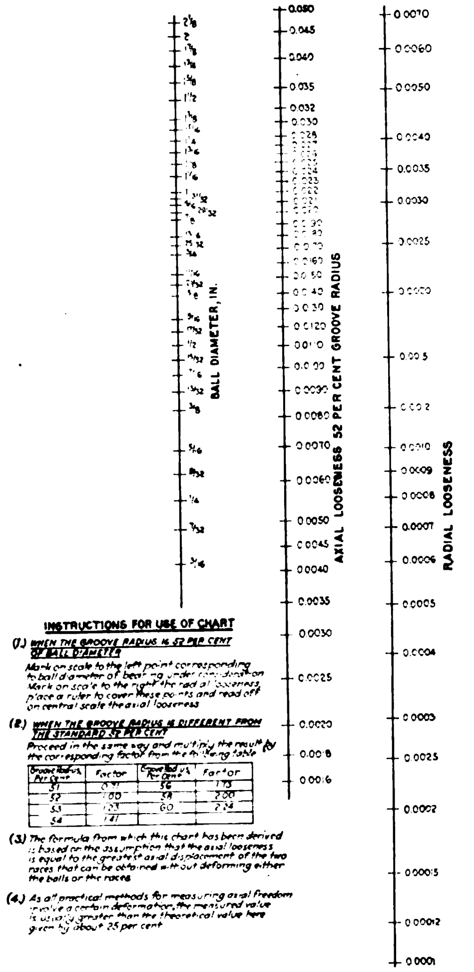


Fig. 8—Nomographic Chart Designed To Convert the Desired Radial Clearance of a Ball Bearing into the Equivalent Axial Play

yond that possible under even unreasonable inspection practice. But the noise, if anything, was worse.

It remained for us to tear down and to inspect several transmissions that were passed in the final car-test as being quiet. When this was done it was found that the roller race on the shafts had been ground to the low-limit diameter, while the roller race in the shells had invariably been ground to the largest or extreme high limit specified for these holes. For purposes of comparison, six noisy transmissions were then torn down, the bearing diameter *b* was ground approximately 0.001 in. under the former low limit and the transmissions were reassembled and tested. This change caused the noise practically to disappear.

Experiments were made repeatedly with noisy transmissions and, in every case, when this alteration was made and the bearing clearance was increased, the noise was either entirely eliminated or was reduced to a degree that was not objectionable. We concluded that this result was produced by providing sufficient space for an adequate oil-film. Further experiments along similar lines have substantiated this conclusion. This explanation seemed logical since we have always found such a clearance to be necessary in crankshaft and connecting-rod bearings.

The remedy seemed a simple one to supply but we were quite concerned about mounting a bearing under conditions that simulated those it would assume after several months' wear. We had always supposed that bearings of the anti-friction type must be mounted snugly. Before definitely adopting the new practice, wisdom demanded that we check the effect of the greater diametral clearance on the wear of the bearing. Transmissions were run under similar conditions with the standard or snug bearing and with the increased clearance. We found that the snug bearing wore rapidly during the early stages of the test and eventually reached the state of looseness with which the other bearing started. The loose bearing, on the contrary, practically retained its original clearance. We concluded that the wear of the snug bearing was accelerated because of the absence of an oil-film sufficient for complete lubrication. The loose bearing apparently accommodated an adequate oil-film and the wear was normal. The test was continued for some time and frequent examination showed that the snug or full-fitting bearing continued to wear faster than the loose one. This, we believe, is due to the heavy initial wear that breaks or distorts the ground surface instead of glazing it as seems to be the case when the bearing is assembled with a proper clearance at the start.

Ball-Bearing Clearances

After the transmissions using the roller bearing had been changed to conform to the practice just described, it became apparent that the same gear-noise existed in the transmission used in one of the other Packard models which had the transmission gears mounted on ball bearings. We altered a few experimental ball bearings by deepening the grooves in the races to allow a minute clearance for the balls, instead of assembling them to a good rolling-fit or to the fit of the standard stock. This was done to provide oil clearance, as had been done with the roller bearings. When the loose ball bearings were substituted for the tight ones in noisy transmissions, our previous experience was repeated and the noise decreased. Careful experiments determined the desirable clearance to allow in the races for the reduction of noise, and the manufacturers of ball bearings agreed to supply bearings with various amounts of clearance so that we could determine the requirements of this work.

It was a comparatively easy matter to determine the clearance needed in the case of the roller bearings since diametral clearances could readily be measured. It was found to be difficult, however, to measure radial clearance in the ball-bearing races and we were forced temporarily to determine the degree of clearance by the end-play or axial looseness. A maker of ball bearings prepared the nomographic chart, reproduced in Fig. 8, for the purpose of converting a desired radial clearance into the equivalent axial play. This enabled us to order bearings with a selected radial clearance, which could be held uniform without demanding a finer tolerance from the bearing maker, and cured the gear noise most effectively.

We selected three groups of 10 bearings each and assembled the transmissions with the respective groups. Group No. 1 represented the average clearance in the bearings found in stock; No. 2 contained more radial clearance; and No. 3 still more or about 0.0006-in. radial clearance added to the old standard. These bearings

(*Concluded on p.* 437)

NATIONAL AERONAUTIC ASSOCIATION

THE National Aeronautic Association of the U. S. A. was formally organized in Detroit last month at the time of the holding of the Second National Aero Congress there and the national airplane races at Selfridge Field, Mount Clemens, Mich. Over 300 delegates from various parts of the Country attended the congress, and a large number of them took part in many arduous long committee and general sessions with the avowed determination to establish a national organization to foster the advance of the arts and sciences of aeronautics and allied interests. In this purpose they were successful.

The delegates represented groups of States divided into districts on the Government national defense basis. The congress was constituted of an unusually impressive body of men, and was conducted in a businesslike expeditious manner. The work of Lieut.-Col. Harold E. Hartney preliminary to the convention was very highly commended. The Detroit Aviation Society gave excellent cooperation in connection with the congress, as well as in the conduct of the races.

The following were elected as officers of the National Aeronautic Association for the ensuing year:

President, H. E. Coffin, Detroit
Vice-President, B. M. Mulvihill, Pittsburgh
Treasurer, B. F. Castle, New York City
Secretary, J. B. Coleman, Sioux City, Iowa

Governors

S. D. Waldon, Detroit
P. H. Adams, Boston
G. L. Cabot, Cambridge, Mass.
J. D. Larkin, Jr., Buffalo
M. J. Cleary, New York City
L. F. Sevier, Pittsburgh
R. F. Walters, Baltimore
A. S. Fliery, Birmingham, Ala.
F. H. Burgin, Atlanta, Ga.
G. L. Martin, Cleveland
D. M. Outcalt, Cincinnati
C. S. Rieman, Chicago
Ralph Cram, Davenport, Iowa
Howard Wehrle, Kansas City, Mo.
Edgar Tobin, San Antonio, Tex.
William Long, San Antonio, Tex.
P. G. Johnson, Seattle, Wash.
C. H. Messer, Spokane, Wash.

Races

The winners of the races were as follows:

Curtiss Marine Flying Trophy—Navy T R 1 plane (Lieut. A. W. Gorton) Average speed, 112.65 m.p.h.

Detroit News Aerial Mail Trophy—Army Martin Transport (Lieut. E. H. Nelson) Average speed 105.1 m.p.h.

Aviation Country Club of Detroit Trophy—Army Honeymoon Express (Lieut. H. R. Harris) Average speed 135.1 m.p.h.

Liberty Engine Builders' Trophy—Army Lepere observation plane (Lieut. T. J. Koenig) Average speed 128.8 m.p.h.

Pulitzer Trophy—Army Curtiss high-speed pursuit plane (Lieut. R. L. Maughan) Average speed 206 m.p.h.

Air Institute Meeting

An interesting session was held on the morning of Oct. 11 under the auspices of the Detroit Aviation Society. Prof. H. C. Sadler, of the University of Michigan, presided; the program having been arranged by a committee representing the National Advisory Committee for Aeronautics, the American Society of Mechanical Engineers, the Aeronautical Chamber of Commerce and the Society of Automotive Engineers. John R. Cautley took an unusually active part in the preparations for the meeting.

Dr. Joseph S. Ames, of the National Advisory Committee for Aeronautics, sent a communication on the Importance of Scientific Investigation in a General Aeronautical Program. L. E. Pierson, president of the Merchants Association of New York City, contributed remarks on Commercial Aviation and the Commercial Bank. C. F. Redden, president, Aeromarine Airways, had as a topic Flying-Boat Transportation. W. P. MacCracken, Jr., chairman of the Aviation Committee of the American Bar Association, gave a comprehensive outline of Aeronautical Legal Problems. Edmund Ely, president, National Aircraft Underwriters Association, spoke on the status of Aircraft Insurance. Prof. E. P. Warner, of Massachusetts Institute of Technology, prepared for the session a paper reviewing developments in Europe from the standpoint of Commercial Flying. C. G. Peterson presented an analysis of a proposed Contract Air-Mail Route between Chicago and New York City.

The Air Mail

Col. Paul Henderson, second assistant postmaster general, in connection with The Air Mail said:

> Our service at present consists of a relay advance of mail from New York City across the Continent, and vice versa. That is to say, we do not take any particular mail for a complete trip across the Continent. We advance into Cleveland certain mail which misses the late night-trains out of New York City. We take from Cleveland into Chicago mail which, if we did not carry it, would go into Chicago on a train too late for delivery in the afternoon. This process is repeated in relays across the Continent with the net result that we advance approximately 12,000 lb. of first-class letter mail each day by some 3 or 4 hr. It should be noted that this may in certain instances mean a real advance of 15 to 18 hr., inasmuch as it may result in the delivery late in the evening of mail which might otherwise have not been delivered until the following morning.
>
> The planes that we are now using are remodelled DeHavilands, which we procure free of charge from the Army. As of this date, we have 70 such planes in flying condition. Twenty are in the air each day and about 24 are in process of being overhauled and rebuilt. Our engineers have found it necessary to make some 200 changes in the design of the plane to make it suitable for carrying mail.
>
> We use Liberty engines, also procured without charge from the Army. Our experience has shown us that at the end of 100 hr. flying service it is necessary to overhaul each engine. This we do at an average cost of about $250 per engine. At the end of 300 or 400 hr. flying service we overhaul the planes themselves. The major portion of this overhauling and rebuilding is done in our shops at Chicago, which are rather complete, approximately 100 persons being employed in them.
>
> From July 16, 1921, until Sept. 7, 1922, we flew approximately 2,000,000 miles without a fatal accident. During the fiscal year ended June 30, 1922, we maintained an efficiency of 94.39 per cent. This means that out of every 100 trips scheduled, 94.39 were finished on schedule time. Our records show that two-thirds of our trips were made in clear weather; one-third in foggy, cloudy or stormy weather.
>
> On Sept. 16 we finished 10 consecutive weeks of flying the entire Transcontinental route with 100 per cent efficiency; that is to say, during those weeks each of our scheduled trips was started and finished exactly on schedule time. It should be remembered that our daily route includes the crossing of three mountain ranges, the Alleghanies, the Rockies and the Sierras.
>
> There are employed in the Air Mail Service 390 people, of whom 39 are pilots. With three or four exceptions, our pilots are all "ex" Army or Navy flyers. They are exceptionally high-grade young men

and to them is due much credit for the success of the Air Mail Service.

NIGHT FLYING

It is obvious that to get from the airplane all that it may offer in the shape of postal service it will be necessary to fly at night. With this thought in mind, we have for the past 4 months conducted an intensive series of experiments on this subject. I think we have reached the stage where it is safe to conclude that it is entirely possible to fly at night. We expect within a few weeks to light, as an experiment, our Chicago field, and I am optimistic enough to predict that within 6 or 8 months we will be able to fly from Chicago to Cheyenne at night.

If we are successful in this, it will mean that we will be able to make a Transcontinental flight from New York City to San Francisco in one continuous movement, flying from New York to Chicago in the daytime, from Chicago to Cheyenne at night, and from Cheyenne to San Francisco during the early part of the second day. We should be able to establish and maintain a schedule of from 28 to 30 hr. between New York and San Francisco if this night-flying experiment proves out.

Our plan for night flying includes an emergency landing-field every 25 miles, with the proper field-lights, and a beacon-light visible for a distance in excess of 25 miles.

It is my personal opinion that within 2 or 3 years the Air Mail Service will have developed to a point where it will undoubtedly be thought wise to turn over the service to private contractors and make it nation-wide in scope, with higher postage than is now charged on ordinary letters.

The position of the Post Office Department in the matter of the Air Mail Service is that such information as we are able to develop and such experiments as we are able to follow through to a conclusion are for the benefit of the Country at large. If in our work we shall be able to add our share toward the prompt advancement of aeronautics, we will have done our duty.

COMMERCIAL AIR-TRANSPORT

J. Rowland Bibbins, manager, department of transportation and communications of the Chamber of Commerce of the U. S. A., pointed out that

the American Nation is now spending for transportation about $100 per capita per year, far more than the whole prewar national debt, and nearly half the present national debt. If all business transactions could be done at one place and time, there would be no need for long-haul transport. But during the year, commerce requires a movement of 500,000,000,000 passenger-miles and 50,000,000,000 ton-miles on the railroads alone, neglecting entirely the enormous movement, yet unchartered, of 10,500,000 motor vehicles.

Outside of the actual transport cost, a great and unknown cost is the time element in the transaction of business, not only of personnel, but of mail, specie, securities, bank clearings and urgent merchandise, as well as less valuable and urgent freight and express. Every added hour or even minute in transit adds to the cost of doing business, in personnel, interest carrying charges, additional equipment needed in transport, and additional working capital assets of business. This conception is not visionary, it is an actuality, and has given rise to various methods and agencies for expediting business, for which service additional rates are paid, and paid gladly. Here is an open field, and a fair field, for air transport.

I think it is not an exaggeration to say that railroad investment other than main-line represents perhaps one-half of the total investment to-day, or $10,000,000,000. And we have estimated roughly a total investment in marine terminal facilities, other than railroad tide-water facilities, included above, of at least $1,000,000,000. No man knows how much of this tremendous investment could have been avoided by starting right, with a reasonable unification of effort and cooperative organization such as now have come about in the 50-odd railroad belt-line companies in the United States, as exemplified in the Railroad Terminal Association of St. Louis, and the plan now underway for the New York industrial district; but we can draw from past experience and see to it that in this great development that is to come, terminal duplication with its super-burden of cost and inconvenience to the traveling public and traffic, should be reduced to the practical minimum. In fact, while it is a far look ahead, this may possibly become the key to maximum future success of air transport.

There is no profit in speculating what ultimate relation will be established and proved desirable between air transport and other transport agencies. We are learning rapidly, however, through the tragic experience of war, and bald necessity to-day, that our National transportation plant is in essence a single great problem and not an unrelated series of separate problems. And experience is rapidly being accumulated which will enable us to determine in the not distant future just what special field and radius of operation as well as economics will develop the best that lies in each form of transportation, rail, motor, trolley, canal and ultimately air.

Our department has estimated roughly the present-day investment in our national transportation system. It is $50,000,000,000, exceeding manufactures, mining and all others except agriculture. This investment has doubled within about 10 years, and previously within about 15 years. Of course, the recent great activity has been in highway and automobile transport. What of the future?

The basic rail-tonnage per capita has risen steadily to 25 tons in 1920, hardly without interruption. Railroad investment per ton carried has reduced through economic methods to practically a stable level of the last decade. Our population and tonnage demand will increase by a substantially known amount. During the next 20 years this combination of events will demand at least $25,000,000,000 new capital for transportation in all forms.

In the meantime probably half of the whole transportation plant representing the depreciable elements, will pass through the renewal cycle, a polite term for the scrap-heap, $25,000,000,000 more. A total of $50,000,000,000 in transportation funds will thus have to be raised and spent somehow. This is twice our national debt to-day. Looking backward as well as forward, the real potentiality of a new speed-service appears in its true light, for every additional increment in speed of transit will release just so much existing capital for non-speed service. I am no prophet, but the facts are irresistible.

Chassis Friction Losses

By E. H. Lockwood[1]

Metropolitan-New England Sections Paper *Illustrated with* Photographs and Charts

THE loss of power due to the friction of the various parts of the chassis has been carefully and elaborately investigated by a dynamometer, the dual purpose being the determination of the amount of internal frictional resistance of the front or rear wheels and the measurement of the power that can be delivered at the rear wheels with the concomitant rate of fuel-consumption.

The rolling-friction due to the resistance of the wheels as a whole is taken up first and afterward the separate resistances of the tires, bearings and transmission are studied under varying conditions of inflation-pressure and load. The five frictional resistances that were chosen as giving the most useful information are those of the front tires, the rear tires, the front bearings, the rear bearings and the engine.

Among other topics considered are the ratio of the total friction less the weight of the vehicle; a comparison of the resistance of passenger cars and trucks, of solid tires with pneumatic and of fabric tires with cord; the ratio of tire-friction to bearing friction; the rules for determining the total friction of the chassis; the effect of variations in the load and the inflation-pressure on the rolling-resistance of pneumatic tires; the development of resistance formulas for fabric and cord tires; a comparison of the wear of pneumatic tires with that of solid tires under the same load, the effects due to variations of the speed and to heavy-wall tubes, roughness of the tread, non-skid surfaces, large and small sizes, age and duration of wear; the development of heat in the tires; and the influence on friction of the rise of temperature of the gearbox lubricant.

The apparent decrease of friction during the last few years and the uniformity of the products of certain manufacturers are noted. The results of the tests are shown in detail by numerous charts and tables.

THE dynamometer drums consisted of metal-shrouded paper cylinders, mounted on a heavy shaft hung from the basement ceiling on ring-oiled babbitt-bearing hangers. The tops of the drums projected slightly above the main floor through openings in the concrete. The diameter was about 67 in.; the faces were 15 in.; and the overall width was 71 in.

Dynamometer measurements of a chassis were made by placing one pair of wheels on the tops of the drums, so that the wheels and drums revolved in rolling contact. Two distinct ends were sought: (*a*) the determination of the amount of internal frictional resistance of the front or the rear wheels; (*b*) the measurement of the power that the engine can deliver at the rear wheels, and the rate of fuel-consumption.

The power to drive the drums was obtained from a 15-hp. variable-speed electric motor which was belted to the drum-shaft. The power delivered to the drum-shaft by the engine of the car was measured on a prony-brake pulley of 100-hp. capacity. The drum-shaft rotation was measured by two ratchet counters and by an electric tachometer reading in revolutions per minute. All the apparatus could be observed and controlled from an operator's table on the main floor near the drums.

Fig. 1 is a view of the dynamometer from the basement, showing the driving motor and the prony brake. Fig. 2 shows the operator's table with the starting rheostat and meters for the electric motor, drum tachometer, revolution counters and scales for weighing the prony-brake load. The entire apparatus is shown in vertical section in Fig. 3 to which reference may be made by the following letters:

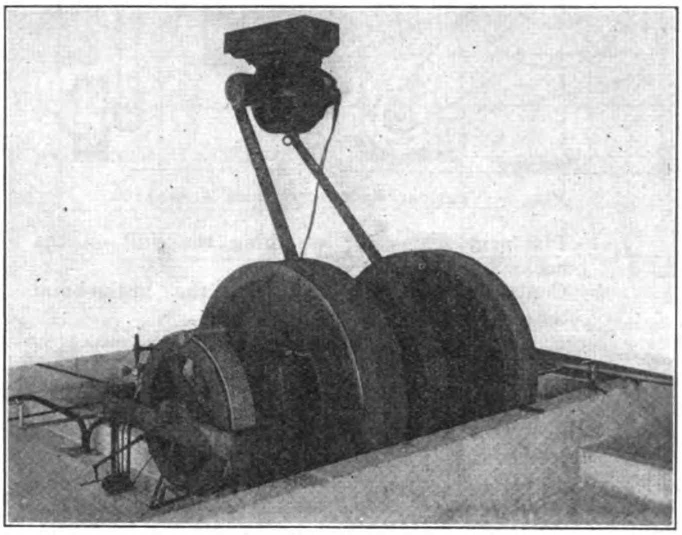


Fig. 1—View of the Dynamometer from the Basement Showing the Prony Brake

a, a, Main drums
b 15-hp. variable-speed electric-motor belted to the drums
c Prony-brake pulley on the main shaft
d Electric tachometer, geared to the main shaft
e Indicating dial of the electric tachometer
f, g Direct-current meters for the 15-hp. motor
h Starting rheostat for the 15-hp. motor

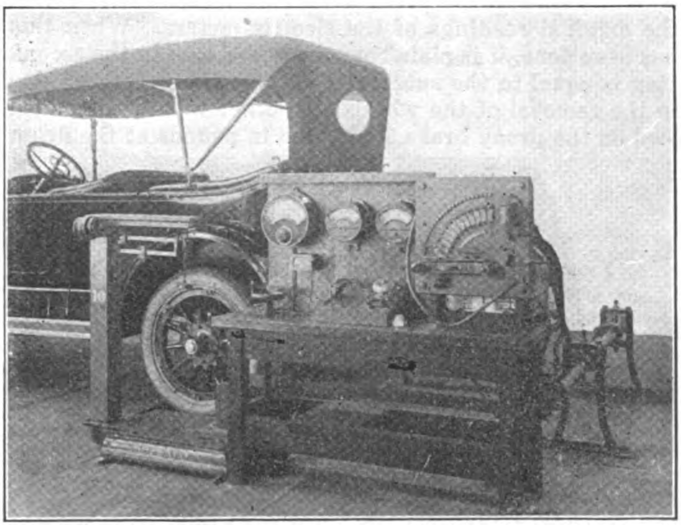


Fig. 2—Operator's Table

[1] M.S.A.E.—Assistant professor of mechanical engineering, Yale University, New Haven, Conn.

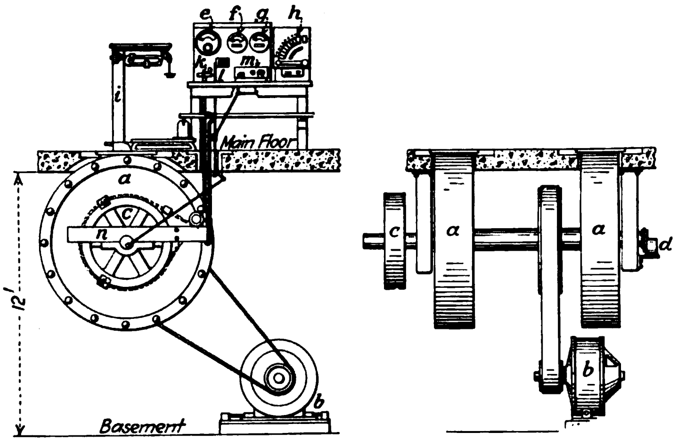


FIG. 3—VERTICAL SECTION OF THE APPARATUS

i Platform scales for weighing the pull on the brake-arm
k Control-handle for adjusting the brake-band tension

MEASUREMENT OF ROLLING FRICTION

The simplest measure of rolling friction is the tractive force that must be applied to the axle of the wheel to overcome the resistance. The tractive force can be measured by a spring-balance when the wheel is being towed, as shown in Fig. 4 at the left. The towing method is not practicable except at very slow speeds. The drum method, shown in the right-hand portion of Fig. 4, produces the same result as the towing method, but avoids its drawbacks. In this case the wheel rotates with the axis stationary, while the tractive force required to revolve the drum is being measured. The tractive force at the drum circumference can be measured conveniently and accurately by the dynamometers on the Mason Laboratory drums. The rolling friction, or rolling resistance, is measured in pounds, and is the tractive force required to revolve the drums against the resistance of the wheels. The measurement of rolling resistance requires two separate steps: (*a*) the drums are revolved with the wheels in place, while the electric input is carefully read and recorded; during this stage, the current input represents the total power required to drive the drums and the vehicle wheels; (*b*) the wheels are removed from the drums, after which the idle drums are rotated at the same speed as before. The operator then adds the load to the prony brake until he has duplicated the original readings of the electric meters. When this has been done, it is plain that the *added load* in the second step is equal to the *subtracted load* in the first step, due to the removal of the wheels. In other words, the added load on the prony brake, expressed in pounds at the drum circumference, is a measure of the rolling resistance of the wheels.

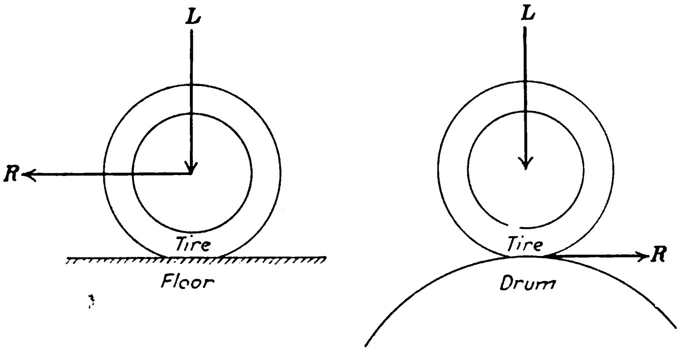


FIG. 4—DIAGRAM SHOWING THE TOWING AND THE DRUM METHODS OF MEASURING ROLLING FRICTION

This method can be varied to give the tire resistance alone, without the resistance of the bearings. To do this the wheels are not removed from the drums, but are jacked up until the rubber surface rests lightly upon the drums with contact enough to cause the wheels to revolve. The second step is then repeated, load being added to the prony brake until the original readings of the current meters are duplicated. The increase in the prony-brake load in this case will be found to be smaller than before, since it represents not the total resistance of the wheels, but only that portion of it that is due to the flattening of the tires on the drums. In jacking up the wheels for tire-resistance measurements, no definite pressure between the tire and the drum is required. Tests prove that any light pressure will suffice, provided it does not cause appreciable flattening of the rubber, yet drives the wheel.

This method was adopted because of its simplicity. It has proved sensitive in practice and readily allows small variations to be detected when the inflation-pressure or the load is changed. The method, however, involves a slight error for which no allowance has been made. This lies in the assumption that the drum-bearing friction and the wheel-bearing friction remain unchanged when the wheel is jacked up for the second reading. It is evident that the bearing friction must be less when the wheels are jacked up, hence the prony-brake reading represents not only the tire resistance but also the slight change in the bearing resistance. The final result, therefore, is slightly too large. It has been deemed safe to neglect this error since it is very small and does not affect the validity of comparative conclusions.

TOTAL FRICTIONAL RESISTANCE

Friction measurements have been made at intervals since 1916 on cars having great variations of weight and tire equipment. About 50 typical examples have been chosen for the comparison of the ratio of the total friction loss to the weight of the vehicle. These cars had pneumatic tires and ranged in weight from 1800 to 5300 lb. Fabric tires were used on the lighter cars, in most cases, and cord tires on the heavier ones. The inflation-pressure was from 60 to 80 lb. per sq. in. Observations were made at speeds of 20, 30 and 40 m.p.h. The results were averaged because the friction was practically constant at all speeds.

The front-wheel and the rear-wheel resistances were measured separately, and the results were plotted with the weight as the base line. Variations were found in cars of the same weight, as might have been expected. These are shown in Fig. 5 by the dots surrounding the line marked "front." An average line was plotted by taking the mean of the results for a group of cars having similar weights. These points are indicated by the encircled dots, the number showing the size of the group. A straight line was found to represent the tractive frictional resistance of both the front and the rear wheels of the car, after which they were combined into a single line marked "total."

The total frictional resistance of the average car as shown by these tests can be represented by the simple formula

$$R = 30 + 0.012\,L \qquad (1)$$

where

R = Total frictional tractive force, in pounds
L = Total weight of the vehicle, in pounds

The results given by the formula and the diagram in

Fig. 5 refer to the total frictional resistance of the tires, bearings and transmission on a smooth road. The resistance due to the wind is not included. The results given by the formula are for the average car and a considerable variation from this average can be expected. An easy-rolling car with cord tires may have, perhaps, 15 lb. less resistance than that given by the formula, while a hard-running car may have more.

The frictional tractive force of vehicles frequently has been expressed in the special unit "pounds per ton." This unit is convenient when the tractive force varies in direct proportion to the weight of the vehicle; otherwise it is not convenient. For example, in the group of 50 cars referred to, the "pounds per ton" varied from 54 to 34. This unit is somewhat awkward and open to ambiguity since the ton has two recognized values, 2000 lb. and 2240 lb., one being used mostly in America and the other in England.

It seems proper to suggest a new unit that is not ambiguous, namely, "pounds per thousand pounds," or "pounds per M." This unit is merely one-half the pounds per ton when the net ton is used. For example, 40 lb. per ton is the same as 20 lb. per M. Moreover, the new unit is directly comparable with the usual coefficient of friction by changing the decimal point. Thus 20 lb. per 1000 lb. becomes 20 lb. per M; and the frictional coefficient is therefore 0.020.

Resistance of Trucks

The frictional resistance of a number of heavy trucks equipped with both solid-rubber and pneumatic tires has been measured. These results, shown in Table 1, are interesting for comparison with passenger cars. The trucks have additional interest, as they belong to a group that is now being used for the measurement of tractive resistance under actual road conditions. The road tests are being carried out under the direction of Major Mark L. Ireland, of the Quartermasters' Corps, as a part of the extensive program of the advisory board of highway research of the National Research Council. It is expected that the tractive resistance of these trucks, as determined on the road, will soon be available for comparison with the laboratory dynamometer tests of the same trucks:

The trucks have been chosen as examples of the widest variation in internal friction. Yet the total tractive resistance, expressed in pounds per M, lies between the limits of 19.0 and 26.0. The higher values are clearly the fault of excessive friction in the transmission system, due probably to newness of the vehicle; hence they are likely to decrease with use.

These tests tend to prove that the rolling resistance of heavy trucks in good working order might be expressed as 20 to 21 lb. per M for all sizes. This figure differs from that of the group of 50 cars referred to in the previous paragraph and is expressed in equation (1). It agrees better with the group of seven cars mentioned in the next paragraph. These cars were tested recently and were known to be in free-rolling condition. Their rolling resistance was expressed by the figure, 19 lb. per M. This fairly close agreement among widely differing vehicles may lead to the further generalization that the rolling resistance of all rubber-tired vehicles, pneumatic and solid, when in best running condition, lies between the narrow limits of 19 and 21 lb. per M.

The lack of agreement with the 50 cars tested from 1916 to 1921 has been noted. This may be explained partly by the fact that most of the lighter cars were equipped with fabric tires, while the heavy cars were equipped with cord tires. Equation (1) yields 27 lb. per M for light cars but, on the other hand, it gives 19 lb. per M for a car weighing 4500 lb. It seems likely

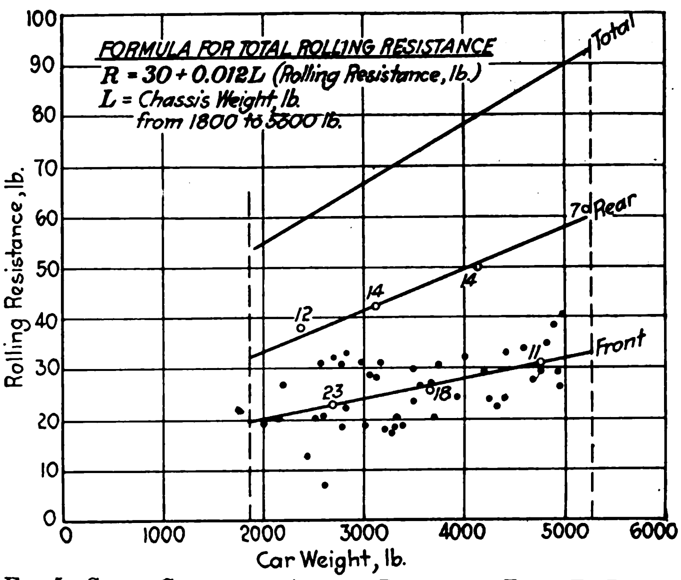


Fig. 5—Chart Giving the Average Results of Tests to Determine the Rolling Resistance of the Chassis of 50 Cars Equipped with Pneumatic Tires

TABLE 1—FRICTIONAL RESISTANCE OF TRUCKS

Specification	Quartermaster Standard Type B Truck No. 432,799	Quartermaster Standard Type B Truck No. 44,913	Mack Chain-Drive 7½-Ton Truck
Front Wheels			
Weight, lb.	4,665	4,395	5,200
Tires, Solid Rubber Single, in	36x5	......	36x7
Tires, Pneumatic Single, in.	......	38x7	......
Total Tractive Force, lb.	80.0	63.0	98.0
Tractive Force per 1,000 Lb. of Weight, lb.	17.5	14.4	19.0
Rear Wheels			
Weight, lb.	6,875	6,300	7,115
Tires, Solid Rubber Dual, in.	40x6	......	40x7
Tires, Pneumatic Single, in.	......	44x10	......
Total Tractive Force, lb.	155.0	215.0	195.0
Tractive Force per 1,000 Lb. of Weight, lb.	22.5	34.0	27.5

Loaded Truck, Complete

	Weight, Lb.	Total Tractive Force, Lb.	Tractive Force per 1,000 Lb. of Weight, Lb.
Type B Truck, No. 432,799	11,540	235	20.3
	14,540	292	20.0
	17,540	344	19.6
	20,540	396	19.3
Type B Truck, No. 44,913	10,695	278	26.0
	13,695	349	25.5
Mack 7½-Ton Truck	12,315	293	23.8
	15,315	340	22.2

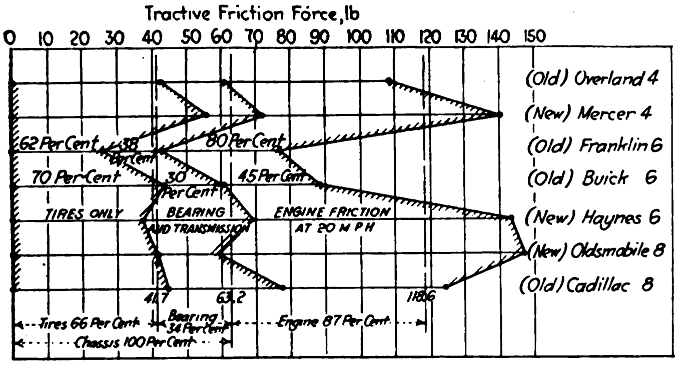


Fig. 6—Internal Friction of Automobiles

that with the increasing use of cord tires future tests will show less resistance for the lighter cars, thus bringing equation (1) nearer to the simpler value of 19 to 21 lb. per M for all vehicles.

Friction of Parts

Results have been given for the total chassis friction and its relation to the weight of the vehicle. It is now proposed to subdivide the total friction into several components to observe their relative importance better.

Five friction items have been chosen as giving the most useful information: (*a*) the front tires, (*b*) the front bearings, (*c*) the rear tires, (*d*) the rear bearings, including the transmission in the neutral gear; and (*e*) the engine friction in the direct drive when the rear wheels are turning at the equivalent of a car speed of 20 m.p.h.

Speed has not been mentioned in the first four items inasmuch as the friction is nearly constant at all speeds. Engine friction has been measured by driving through the rear wheels from the dynamometer, with the ignition shut-off. Care was taken to have the engine warm before beginning the measurements; also to have the throttle wide-open, as this was found to reduce the resistance materially. This last fact suggests that engine resistance is due in part to compression action in the cylinders or to air friction in the valves. It was observed that the engine resistance, measured in terms of tractive force at the rear wheels, increased directly as the speed. The frictional resistance of the engine at 40 m.p.h. was *double* that at 20 m.p.h. This result is significant by contrast with other friction measurements including those of tires, bearings and transmission, of which the increase of friction with speed is negligible. Granting that the engine friction determined in this way is too large, it may at least be used for comparative purposes.

Seven representative automobiles were chosen for the friction measurements. Table 2 gives the dimensions, weight, and the like, of the several cars, together with the friction items already mentioned. Considerable variation was found in the different cars, as might have been expected from the difference in weight. To bring this out more clearly, the total friction, divided into the three parts of tires, bearings and transmission, and the engine, has been plotted in Fig. 6.

A study of the friction diagram shows that, in spite of variations of other factors, the ratio of tire friction to bearing-transmission friction remains constant in the order of 2 to 1. In other words, the friction loss in the tires is two-thirds the total chassis friction, exclusive of the engine.

Examination of the internal-friction chart reproduced in Fig. 6 shows that the item of engine friction is large, being on the average seven-eighths the remaining chassis friction. Admitting that the measured engine frictions are too large, for reasons already mentioned, it is probably true that comparisons can be made fairly between them. Three of the engines had been used but little since leaving the factory and showed higher friction than the others. Two had run over 12,000 miles since the last overhauling and were in free-running condition. Comparing the two groups, the new engines had nearly three times the friction of the old ones.

A true comparison of the frictional resistance of the chassis requires that all values be reduced to the same car-weight. Table 2 is made out for the seven cars, giving the resistance of the tires, bearings and transmission, and chassis without the engine, in terms of pounds per thousand pounds. The items for the different cars are by no means constant, yet they are not widely apart. The average of the three columns can be considered as fairly representative of the friction items for the cars weighing from 3000 to 5000 lb. From these results the following conclusions can be drawn: In cars weighing from 2500 to 5000 lb. the average friction of cord tires only was 12.5 lb. per M; of bearings and transmission only, 6.5 lb. per M; and of total chassis without the engine, 19 lb. per M.

TABLE 2—FRICTIONAL DISTRIBUTION IN THE CHASSIS

Date of Test	April 3, 1922	April 4, 1922	April 10, 1922	April 11, 1922	April 11, 1922	April 12, 1922	April 12, 1922
Name of Car	Overland	Mercer	Buick	Cadillac	Oldsmobile	Franklin	Haynes
Car Model	85	......	44	......	47 FS	9A	55
Number of Cylinders	4	4	6	8	8	6	6
Engine Bore and Stroke, in.	$4\frac{1}{8}$x$4\frac{1}{2}$	$3\frac{3}{4}$x$6\frac{3}{4}$	$3\frac{1}{4}$x$4\frac{1}{2}$	$3\frac{1}{8}$x$5\frac{1}{8}$	$2\frac{7}{8}$x$4\frac{1}{2}$	$3\frac{1}{4}$x4	$3\frac{1}{2}$x5
Weight: Front Wheels, lb.	1,360	1,850	1,350	2,010	1,495	1,250	1,585
Rear Wheels, lb.	1,650	2,250	1,460	2,400	1,735	1,450	1,815
Total, lb.	3,010	4,100	2,810	4,410	3,230	2,700	3,400
Front Tires: Size, in.	32x4	32x$4\frac{1}{2}$	32x4	35x5	32x4	32x4	33x4
Kind	1 Fabric 1 Cord	Goodyear Cord	1 Fabric 1 Cord	Revere Cord	Federal Cord	Goodyear Cord	Goodyear Cord
Rear Tires: Size, in.	32x4	32x$4\frac{1}{2}$	32x4	35x5	32x4	32x4	33x4
Kind	Cord	K-S Cord	1 Fabric 1 Cord	Yale Cord	Federal Cord	Goodyear Cord	Goodyear Cord
Rolling-Resistance:							
Front Tires, lb.	23.2	25.3	22.8	21.0	16.4	9.5	14.6
Front Bearings, lb.	4.4	2.4	3.2	6.2	4.0	7.4	3.9
Total Front Wheel, lb.	27.6	27.7	26.0	27.2	20.4	16.9	18.5
Rear Tires, lb.	19.0	30.0	20.3	23.5	25.2	16.0	22.8
Rear Bearings and Transmission, lb.	14.0	14.0	14.6	26.5	13.8	10.0	27.8
Total Rear Wheel, lb.	33.0	44.0	34.9	50.0	39.0	26.0	50.6
Total, Front and Rear Wheels, lb.	60.6	71.7	60.9	77.2	59.4	42.9	69.1
Tires Only, lb	42.2	55.3	43.1	44.5	41.6	25.5	37.4
Bearings and Transmission, lb.	18.4	16.4	17.8	32.7	17.8	17.4	31.7
Engine, at 20 M.P.H., lb.	47.5	69.0	27.4	47.5	88.0	34.0	74.4
Total, Including Engine, lb.	108.1	140.7	88.3	124.7	147.4	76.9	143.5

Resistance Formulas

Two rules for determining the total chassis friction have been deduced from the dynamometer tests. Their agreement logically should be compared. The first was deduced from 50 tests of various cars where the front and the rear resistances were measured separately. This embraced a variety of tire equipment that was mostly fabric tires on the lighter cars and cord tires on the heavier ones. These tests were spread over a period of 5 years. The formula for the first series, where L = the weight of the chassis, is

$$\text{Total chassis resistance} = 0.012\,L + 30$$

The second series consisted of seven cars, where the tire and the bearing resistances were separately measured. Practically all the cars were equipped with cord tires and the measurements were made at one time by the same observers. The formula for the second series is

$$\text{Total chassis resistance} = 0.019\,L$$

TABLE 3—INTERNAL FRICTION AND TOTAL FRICTION, IN POUNDS OF TRACTIVE FORCE PER 1000 LB. OF WEIGHT

Name of Car	Weight, lb.	Internal Friction, Lb.		Total Friction, Lb.
		Tires	Bearings and Transmission	
Buick	2,810	15.4	6.3	21.7
Cadillac	4,410	10.1	7.5	17.6
Franklin	2,700	9.5	6.4	15.9
Haynes	3,400	11.0	9.3	20.3
Mercer	4,100	13.6	4.0	17.6
Oldsmobile	3,230	13.0	5.4	18.4
Overland	3,010	14.0	6.0	20.0
Average	3,380	12.4	6.4	18.8

The divergence of the two formulas can be seen best by comparing the values for the same weights of car. For L = 2500, the resistance by the two formulas is: first, 60 lb.; second, 47.5 lb.; difference, 12.5 lb. For L = 4000, the two formulas give first, 78 lb.; second, 76 lb.; difference, 2 lb. From this comparison it appears that the first formula gives larger values for the light-weight cars, while both agree well for the heavier cars. The theory has been advanced that the general use of fabric tires on the lighter cars was the cause of their increased resistance.

The internal friction of several heavy trucks has been separated into tire and bearing elements as shown in Table 4.

In Table 4, Truck B 432,788 was known to have the excessive transmission friction incident to a new truck, which is shown by its bearing and transmission friction of 12.0 lb. per M or nearly double that of its companion truck. The other two trucks can be considered as representatives of the heavy class. Their average tire friction is 13.5 lb. per M, or 61 per cent; average bearing and transmission friction, 8.5 lb. per M, or 39 per cent; total, 22.0 lb. per M, or 100 per cent.

The corresponding figures for passenger cars with cord tires were: average tire friction, 12.5 lb. per M, or 66 per cent; average bearing and transmission friction, 6.5 lb. per M, or 34 per cent; total, 19.0 lb. per M, or 100 per cent. In general, the friction distribution in trucks does not differ greatly from that in lighter vehicles.

TABLE 4—INTERNAL FRICTION OF HEAVY TRUCKS

Name of Truck	Tractive Resistance in Percentage and Pounds per 1000 Lb. of Weight		
	Tires Only	Bearings and Transmission	Total
Quartermaster Standard, Type B, No. 432,799	13.7 lb.	6.6 lb.	20.3 lb.
	67%	33%	100%
Quartermaster Standard, Type B, No. 432,788	15.1 lb.	12.0 lb.	27.1 lb.
	56%	44%	100%
Mack 7½-Ton, Type-AC, Chain Drive	13.4 lb.	10.4 lb.	23.8 lb.
	56%	44%	100%

Fabric Tires

The influence of load and inflation-pressure on the rolling resistance of pneumatic tires becomes evident when the results of tests are studied. Some typical examples will now be presented for a line of fabric tires from the same maker, sizes 32 x 4 in., 33 x 4½ in. and 35 x 5 in. In making tests of the rolling resistance of these tires, an arbitrary schedule of loads and inflation-pressures was adopted, designed to cover a sufficiently wide range for the size of the tires. Three different speeds were used for each test-load and inflation-pressure, resulting in as many as 36 independent readings of rolling resistance for one tire. The changes due to speed were very slight; hence it was possible to eliminate the speed as a variable, giving an average value for each load and inflation-pressure without regard to the speed. The rolling resistance of each tire, arranged in columns under the respective loads, is given in Table 5. The variation of the figures in each column is a measure of the effect produced by the change of inflation pressure.

To bring out the variation more clearly, a parallel column has been added to Table 5 giving the results in

TABLE 5—ROLLING-RESISTANCE OF FABRIC NON-SKID TIRES AT 20 TO 40 M.P.H. SPEEDS

Test No. 28; Size, 32x4 In.; Weight of Tire, 19.5 Lb.; Outside Diameter 32.8 In.

Inflation-Pressure, Lb. per Sq. In.	Load, 460 Lb.		Load, 700 Lb.		Load, 975 Lb.	
	Rolling-Resistance		Rolling-Resistance		Rolling-Resistance	
	Lb.	Per Cent	Lb.	Per Cent	Lb.	Per Cent
30	8.03	186	14.45	165	22.3	163
55	5.35	124	10.60	121	16.8	121
80	4.33	100	8.78	100	13.7	100

Test No. 60; Size, 33x4½ In.; Weight of Tire, 23.5 Lb.; Outside Diameter, 33.7 In.

	Load, 585 Lb.		Load, 935 Lb.		Load, 1,285 Lb.	
30	9.05	124	18.40	152	27.7	145
45	8.65	118	15.60	130	23.0	121
65	8.65	118	14.20	118	21.2	111
90	7.32	100	12.03	100	19.1	100

Test No. 102; Size, 35x5 In.; Weight of Tire, 33.5 Lb.; Outside Diameter, 36.0 In.

	Load, 750 Lb.		Load, 1,050 Lb.		Load, 1,650 Lb.	
40	12.70	123	19.80	128	37.0	148
55	11.40	109	17.60	115	32.4	129
70	10.50	102	15.90	103	28.2	113
90	10.30	100	15.40	100	25.1	100

TABLE 6—COMPARATIVE ROLLING FRICTION OF FABRIC AND CORD TIRES

Size of Tire, in.	32x4	33x4½	35x5
Speed, m.p.h.	20 to 40	20 to 40	20 to 40
Inflation-Pressure, lb. per sq. in.	30 to 80	30 to 90	40 to 90
Load, lb.	460 to 975	585 to 1,285	750 to 1,650
Number of Readings	54	72	32
Average of Rolling-Friction Readings			
Fabric, lb.	11.30	15.20	21.10
Cord, lb.	7.60	9.06	13.45
Fabric, per cent	100.00	100.00	100.00
Cord, per cent	67.00	60.00	64.00

percentages, using 100 per cent for the smallest value of the resistance. The increase of rolling resistance due to lower inflation-pressure is apparent in every case. A general conclusion from the figures in Table 5 can be stated thus: The rolling resistance of a fabric tire, fully loaded and inflated, may be increased by more than 50 per cent when the inflation-pressure is dropped from 90 to 30 lb. per sq. in., and by 25 per cent when the pressure is dropped to 50 lb. per sq. in. A word of caution is added in this connection, namely, that the total car resistance will not be increased in this same ratio, because

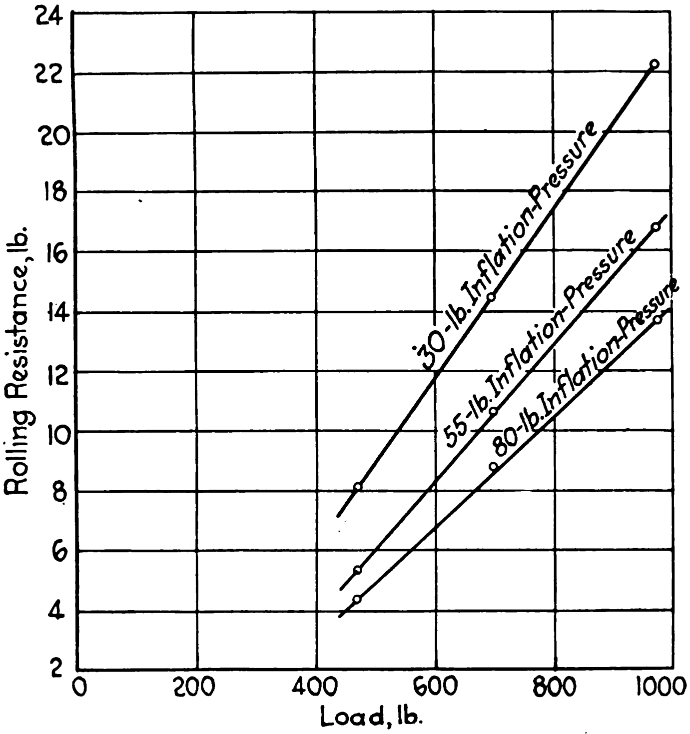


FIG. 7—AVERAGE ROLLING RESISTANCE OF A 32 X 4-IN. FABRIC PNEUMATIC TIRE AT SPEEDS OF FROM 20 TO 40 M.P.H. AND DIFFERENT INFLATION-PRESSURES

the tires produce only a part of the total friction of the chassis; also, that this result applies only to *fabric* tires.

The diagrams shown in Figs. 7, 8 and 9, contain a curve for each inflation-pressure, plotted with the rolling resistance and the load as coordinates. The plotted points show the existence of slight observational errors, yet fairly satisfactory curves can be drawn through them. An interesting fact, shown on all the diagrams, is that the line for all the inflation-pressure curves is straight. Another is that, at the same inflation-pressure, these straight lines practically coincide on each diagram, showing that the rolling resistance depends solely upon the load and not upon the size of the tire. Whether this conclusion can be extended to the larger sizes, such as pneumatic tires for trucks, will be discussed in another paragraph.

The rolling resistance of the fabric tires, when fully

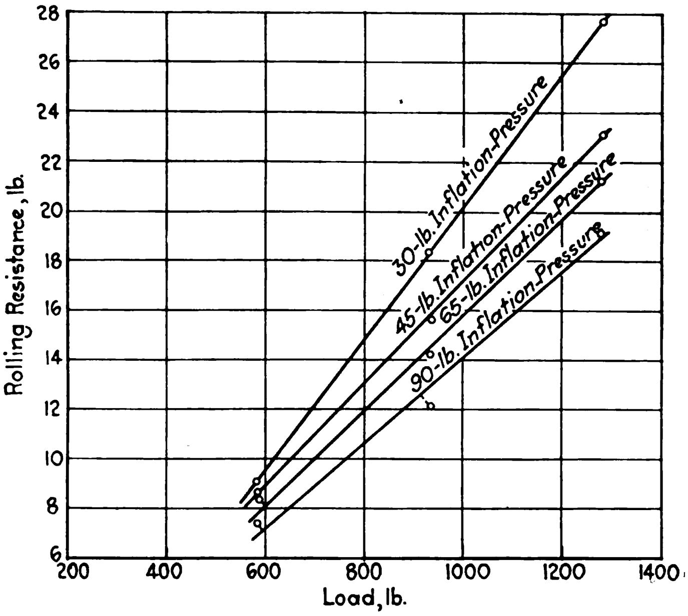


FIG. 8—AVERAGE ROLLING RESISTANCE OF A 33 X 4½-IN. FABRIC PNEUMATIC TIRE AT SPEEDS OF FROM 20 TO 40 M.P.H. AND DIFFERENT INFLATION-PRESSURES

inflated, can be expressed by a simple algebraic formula which applies to all three sizes:

$$R = 0.018L - 3.0 \quad (2)$$

where,

R = the rolling resistance, in pounds
L = the load on the tire, in pounds

Another way of expressing the result is in the simple form of pounds per thousand pounds. This number

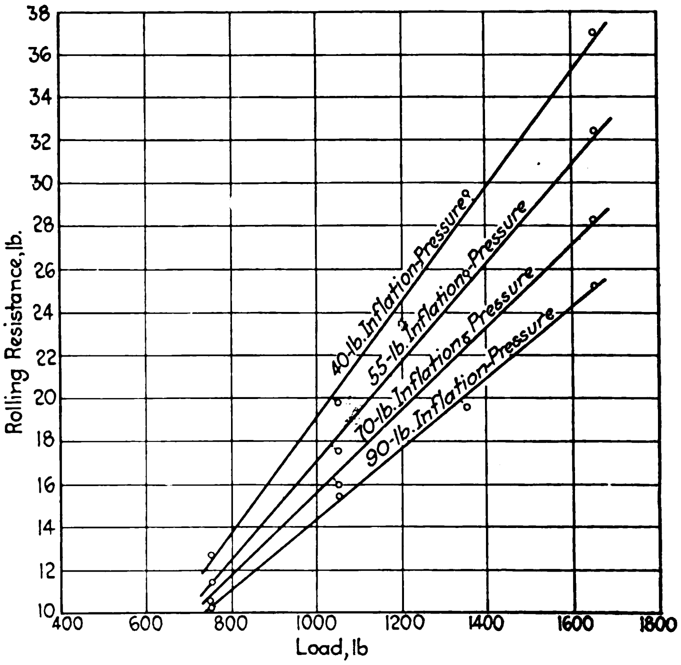


FIG. 9—AVERAGE ROLLING RESISTANCE OF A 35 X 5-IN. FABRIC PNEUMATIC TIRE AT A SPEED OF 20 M.P.H. AND DIFFERENT INFLATION-PRESSURES

varies slightly with the load. Its average value is 15 lb. per M for the tire sizes referred to in this article.

Cord Tires

Dynamometer tests of cord tires show less rolling resistance than those of fabric tires. The difference can be stated as being approximately one-third. In other words, the rolling resistance of a standard make of cord tire is only two-thirds that of a fabric tire. This conclusion has been proved many times during the past 5 years with different sizes and makes of tire. The figures in Table 6, based on tests of Fisk and Goodyear tires, both of which have shown practically identical rolling resistances, are presented as typical.

The deductions from these figures are confirmed by curves plotted from the test results. An example is given in Fig. 10, where two rolling-resistance curves are

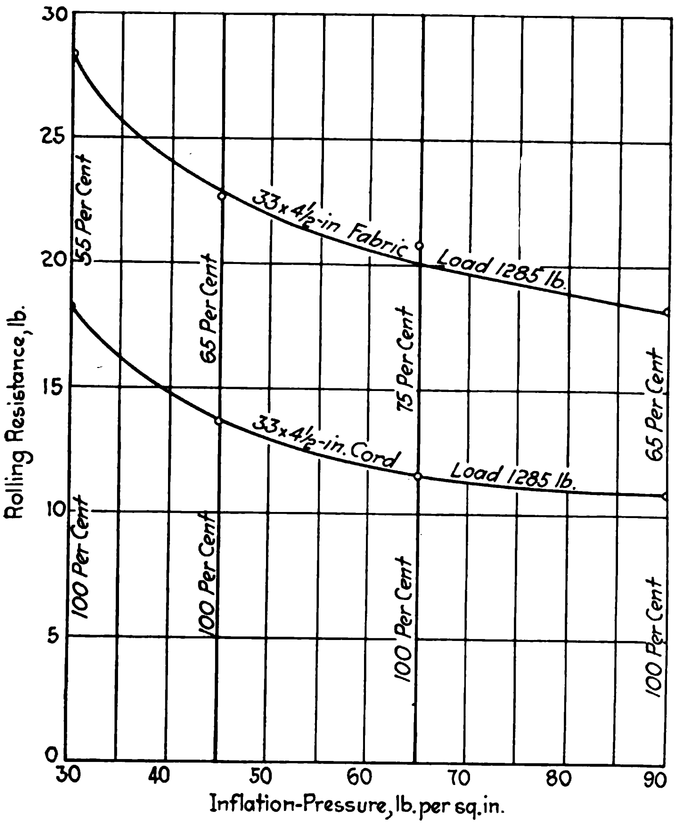


Fig. 10—Typical Rolling-Resistance Curves of Fabric and Cord Tires at a Constant Load and Varying Inflation-Pressures

plotted, one for fabric and one for cord tires. The curves are nearly parallel, showing that the difference in friction is about the same at all inflation-pressures.

The cord tire has a valuable characteristic that is clearly shown in Fig. 10. A low friction is reached at a moderate inflation-pressure. In this case the resistance is 11.5 lb. at an inflation-pressure of 65 lb. per sq. in., and 11.0 lb. at a pressure of 90 lb. per sq. in.; hence, the cord tire can be run at a moderate inflation-pressure with only a slight loss of power. On the other hand, the fabric-tire resistance diminishes steadily as the inflation-pressure is increased; hence, high pressures are demanded for the saving of power.

Another illustration of the difference between fabric and cord-tire resistances is given in Fig. 11. In this example the curves of rolling resistance are plotted for a constant inflation-pressure. The results are given for several sizes of fabric and cord tires. There are differences between the various sizes, but the points lie in a band, or strip, in each case. These curves show clearly that the rolling resistance of any size of cord tire, under any load, is about two-thirds that of a fabric tire under the same conditions.

Table 7—Average Rolling-Friction for Seven Ribbed and Non-Skid Firestone, Fisk, Goodrich and Goodyear Tires

Inflation-Pressure, Lb. per Sq. In.	Load, Lb.	Speed, M.P.H. 20	30	40	Average Rolling-Friction, Lb.
30	585	6.03	6.33	6.91	6.42
45		5.16	4.98	5.26	5.13
65		4.68	4.70	4.65	4.68
90		4.17	4.29	4.54	4.33
Average					5.14
30	935	10.99	11.75	12.37	11.70
45		9.04	9.76	10.01	9.60
65		8.21	8.34	8.43	8.33
90		7.49	8.01	8.22	7.90
Average					9.38
30	1,285	17.39	18.17	19.01	18.19
45		14.06	14.71	14.89	14.55
65		12.16	12.32	12.64	12.37
90		10.87	11.30	11.70	11.29
Average					14.10
General Average					9.54

The rolling resistance of cord tires is further analyzed in Table 7, which gives the rolling friction of 33 x 4-in. tires arranged for three speeds, three loads and four inflation-pressures, a total of 36 readings. The figures in Table 7 are compiled from tests of seven cord tires of well-known makes, and therefore can be considered as fairly representative of new cord tires.

The tabular values have been plotted in the diagram, Fig. 12, with one curve for each 10 lb. of inflation-pressure. The spacing of the curves shows clearly the change of resistance with any change of pressure. After reaching an inflation-pressure of 60 lb. per sq. in., but

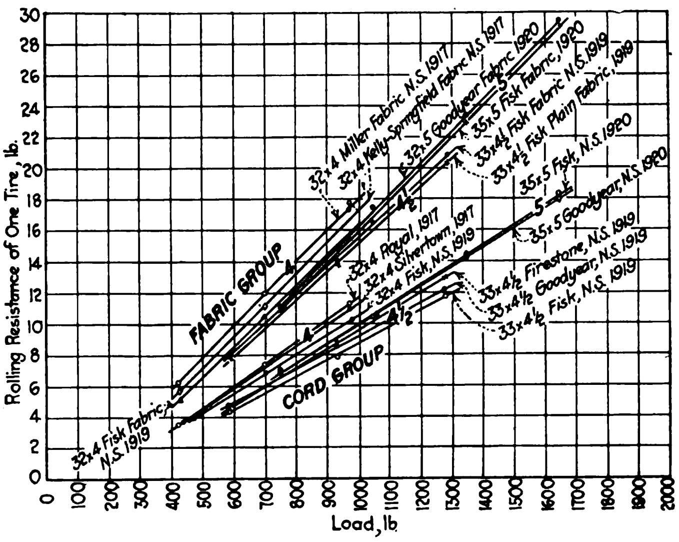


Fig. 11—Comparative Rolling-Resistance Curves of Fabric and Cord Tires at an Inflation-Pressure of 65 Lb.

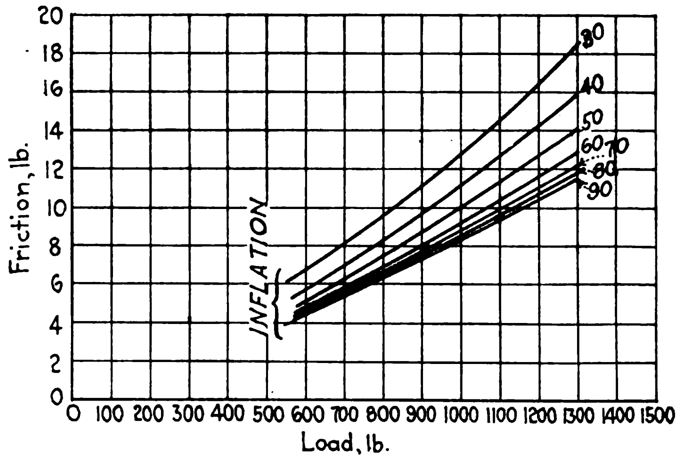


FIG. 12—ROLLING-FRICTION OF 33 x 4½-IN. CORD TIRES AT DIFFERENT LOADS AND INFLATION-PRESSURES RANGING FROM 30 TO 80 LB.

little further decrease in rolling friction is found, again showing that in cord tires a high degree of inflation is not required for easy rolling.

The resistance curves are practically straight lines of varying slope. An empirical formula has been derived for these curves that gives the rolling resistance for any load and inflation-pressure within the limits covered by the experimental work

$$R = [(L - 200) \div 100] \times \{ [1.06 + (90 - P)^2] \div 7000 \} \quad (3)$$

where

$R =$ Rolling-resistance
$L =$ Load
$P =$ Inflation-pressure

The limits of load are from 500 to 1300 lb., and those of inflation pressure, 30 to 90 lb. per sq. in. This formula can be put into simpler form by inserting a normal inflation-pressure of 60 lb. per sq. in., causing it to take the form

$$R = 0.012\,L - 2.0 \quad (4)$$

A similar formula was deduced for fabric tires, where the constants are exactly 50 per cent larger than for the cord-tire formula. This is further evidence that cord tires have two-thirds the rolling friction of fabric tires.

SOLID-RUBBER TIRES

Tests of solid-rubber tires were made at intervals from 1919 to 1921 on ¾ to 2-ton trucks. It was found that solid-tire resistances could be measured by the jacking-

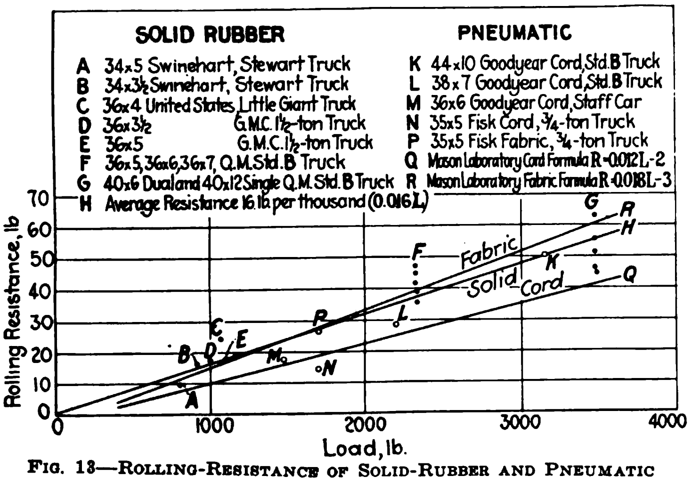


FIG. 13—ROLLING-RESISTANCE OF SOLID-RUBBER AND PNEUMATIC TIRES FOR DIFFERENT LOADS

up method and that, in general, the resistance was between those of cord and fabric tires under the same load.

In 1921, in cooperation with the Committee on the Tractive Resistance of Roads Research, the drums were widened to accommodate the dual tires of heavy trucks. Six trucks were brought to the Mason Laboratory for the investigation of internal friction by the dynamometer method. As planned by Major Ireland, the tests were carried out with empty and with loaded trucks, with different temperatures of the lubricant, with several kinds of new solid-rubber tires, and with a few well-worn tires. The tire-resistance measurements are summarized as shown in Table 8.

TABLE 8—SUMMARY OF TIRE-RESISTANCE MEASUREMENTS

		Tire Size, In.	Load per Inch of Tire Width, Lb.	Tire Resistance per 1,000 Lb. of Weight, Lb.
New Front Tires	Single	36x4	583	21.7
		36x5	466	19.0
		36x6	375	20.2
		36x7	375	19.0
		Average..	450	20.0
Worn Front Tires	Single	36x5	466	15.5
		36x6	375	16.5
		36x7	375	16.2
		Average..	400	16.1
New Rear Tires	Dual	40x[illegible]	2[illegible]5	18.0
		40x6	275	15.7
	Single	40x12	275	14.0
		40x12	275	15.8
		40x12	275	19.0
		Average..	275	16.5
Worn Rear Tires	Dual	40x6	285	12.5
		40x6	255	11.0
		Average..	270	11.8

The following conclusions are apparent from the figures in Table 8:

(1) Partly worn tires roll with less friction than new and more resilient tires, the difference being from 20 to 25 per cent; hence, increased cushioning must be paid for by increased power

(2) Front tires show more friction than rear tires, both when new and when worn. This is to be explained by the relatively heavier load carried by the front tires. The rear-tire tests were made with the truck empty; hence, the load per inch of tire width was considerably less than that for the front wheels

(3) The rolling resistance of the solid-rubber tires varies considerably, depending upon the wear, the construction and the like, but the values are all between the cord and the fabric-tire figures. Taking the average of all the readings as an index, the solid-tire resistance is 16.0 lb. per M

A final comparison of solid and pneumatic tires has been made in Fig. 13. All the solid-rubber tire resistances have been plotted with respect to the load in three groups; for light trucks, and for the front and rear wheels of heavy trucks. The average of all the solid-

rubber-tire results has been represented by the line marked 16 lb. per M, as mentioned above. The results for several large-sized pneumatic-tires also are plotted in Fig. 13 and the Mason Laboratory formulas for fabric and cord tires are stated. Fig. 13 confirms the previous statement that solid-rubber-tire friction is approximately equal to fabric-tire friction.

Dynamometer tire-friction measurements have been made at speeds of 20, 30 and 40 m.p.h. Over this range the observed readings have been nearly constant. Stated more correctly, the variations due to speed have been of about the same order as the observational error of a single reading; hence they are not easily observed. When the averages of many different readings are taken,

TABLE 9—AVERAGE ROLLING-RESISTANCE AT DIFFERENT SPEEDS FOR 11 33x4½-IN. TIRES

Speed, M.P.H.	Series No. 1		Series No. 2	
	Inflation-Pressure, 65 Lb. per Sq. In.		Inflation-Pressure, 90 Lb. per Sq. In.	
	Load, 935 Lb.		Load, 1,285 Lb.	
	Average Resistance		Average Resistance	
	Lb.	Per Cent	Lb.	Per Cent
20	9.4	100	12.6	100
30	9.8	104	13.1	104
40	9.9	106	13.6	108

a slight increase of the rolling resistance with the speed is found. The figures in Table 9 show that, by doubling the speed, the rolling resistance may be increased by 6 or 8 per cent.

In view of the small change produced by the speed, it has been deemed most satisfactory to ignore it and to give the average value as constant for all speeds. This assumption is allowable for rolling on smooth, hard surfaces. For rough, uneven or soft roads, the rolling resistance doubtless will increase materially with the speed.

Careful measurements have shown that the non-skid type of tread produces slightly more resistance than the smooth tread. The increase produced by the non-skid tread is small and is presented as a matter of interest rather than importance. The figures in Table 10, based on tests of three well-known tires, will indicate the influence of the non-skid tread.

The figures in Table 10 seem to show that the effect of the non-skid tread is to increase the rolling resistance about 5 per cent. The increase is caused, probably, by

TABLE 10—INFLUENCE OF NON-SKID TREAD ON ROLLING-RESISTANCE[2]

Kind of Tire, Cord	Rolling-Resistance	
	Ribbed Tread, Lb.	Non-Skid, Tread, Lb.
Fisk................	8.86	9.23
Firestone............	8.84	9.52
Goodyear............	8.80	9.98
Average..........	8.83	9.58

[2] Average of 36 readings on 33 x 4½-in. cord tires at different speeds, loads and inflation-pressures. The increase in resistance due to the non-skid tread is 8 per cent.

TABLE 11—ROLLING-RESISTANCE OF LARGE PNEUMATIC TIRES

Tire Size, In.	Rolling-Resistance per 1,000 Lb. of Weight, Lb.
32x4	11.0
33x4½	11.0
35x5	11.0
36x6	12.5
38x7	13.0
44x10	16.0

the greater compression of the rubber under the load and may vary somewhat with the shape of the non-skid markings.

The influence of heavy-wall tubes of the "non-puncture" type has been measured by running alternate tests with thin-wall and thick-wall tubes in the same casing. The results in two instances have shown slightly greater friction for the thick-wall tubes, the difference being about 5 to 10 per cent.

Similar results have been obtained from a tube protector, consisting of a thick endless pad lying between the tube and the casing. The protecting strip was found to increase the rolling resistance by 7 per cent, in a 33 x 4½-in. cord-tire casing.

Large Pneumatic Tires

Test of the smaller sizes, 4 and 5-in. tires, have shown that the rolling resistance is practically identical for the same loads and inflation-pressure. If this holds true, it

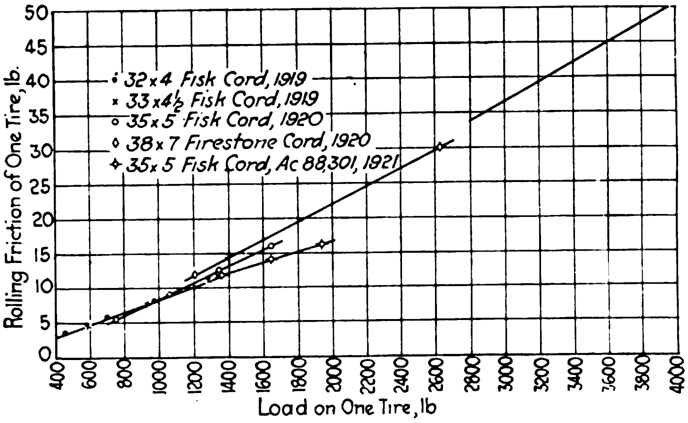


FIG. 14—ROLLING-RESISTANCE OF DIFFERENT SIZES OF CORD TIRES AT AN INFLATION-PRESSURE OF 90 LB.

follows that a single curve or formula for rolling resistance would hold for all sizes of a given type, such as a cord tire of normal inflation-pressure. Unfortunately for this theory, tests of the larger sizes have shown somewhat higher friction, proportionately, than those of the smaller sizes. The values in Table 11 are averages for several sizes of cord tires as measured on the drums.

Fig. 14 affords a comparison of several sizes of cord tire under different loads. It is probably true that the larger sizes were less well inflated than the smaller sizes. In one case, that of the 44 x 10-in. size, the tire was marked with the direction to "Inflate to 140." The test was made at a pressure of 100 lb. per sq. in. because of a limitation of the available air-supply at that time. It can be concluded that the larger-sizes of pneumatic tire will have somewhat higher rolling resistance, proportionately, than the smaller sizes. This statement can be qualified by the further assumption that if all the sizes were kept at the inflation-pressures recommended by the

manufacturers, the rolling resistances would be practically alike.

The evidence of the effect of wear on the rolling resistance of pneumatic tires is rather conflicting, since some old tires have shown excessive friction, while others have agreed well with that of new tires. Close inspection proves that in a majority of cases the rolling frictions of a new and of an old tire are about the same; also that in exceptional instances some reason for the difference usually can be found as when old tires get out of shape and run untrue on the rims. Reliable information on the effect of age and wear on tires is difficult to obtain but fortunately this is not a matter of practical importance. From the evidence at hand it is believed that a worn pneumatic tire has about the same rolling friction as a new tire of the same kind. Solid-rubber-tire resistance is known to decrease as the tire wears out, leaving a thinner layer of rubber.

The Mason Laboratory tests give evidence that there has been a general decrease in the rolling resistance of pneumatic tires for several years. Tests of tires of the same size have been carried on under the same conditions of loading and inflation; hence the results should be comparable. In the 32 x 4-in. size, a considerable decrease was observed from 1917 to 1919 in both fabric and cord tires as shown in Table 12. This has been partly lost between 1919 and 1922. The decrease in rolling friction from 1917 to 1919 is shown in detail in several of the diagrams of comparative results.

The 35 x 5-in. size referred to in Table 13 was first tested in the Mason Laboratory in 1920. Fabric and cord tires of several different makes were compared and showed a satisfactory agreement in rolling friction. Tests of cord tires of this size have shown a constant decrease of friction, amounting to about 10 per cent.

The 33 x 4½-in. tires of 1919 cited in Table 14 were characterized by a low rolling-friction as shown by Fig. 12. This feature seems to have been lost since that time, and recent tests show that a tire of this size now has more rolling-friction than a 32 x 4-in. size.

Comparisons of tire tests made several years apart require proof that the differences really exist in the tires and not in the apparatus itself. It can be said that at least during the last 4 years, while the comparisons have been in progress, cord-tire friction appears to have decreased noticeably. It is noteworthy that tires made by several of the leading manufacturers have shown marked uniform rolling resistances. This indicates, possibly, that the processes of tire manufacture have become standardized.

Deviation from this uniformity has been observed in a few instances. In one case, a pair of tires marked "Cord" was found to have the resistance usually found in fabric tires, that is, 50 per cent in excess of cord tires. In two other cases, cord tires showed 27 and 30-per cent excess over the standard cord-tire figures. All the examples that deviated from the standard values were found in the products of the smaller and less well-known manufacturers.

TABLE 12—ROLLING-RESISTANCE OF 32 X 4-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb. per Sq. In	LOAD, LB. 500	700	1,000
Miller	3	Fabric, non-skid	1917	30	8.8	14.0	21.5
				55	7.2	11.7	18.5
				80	6.5	11.0	17.5
Kelly-Springfield	12	Fabric, non-skid	1917	30	9.1	15.5	25.0
				55	7.7	13.1	21.2
				80	6.3	10.5	17.1
Fisk	27	Fabric, plain	1919	30	7.7	12.3	20.0
		Weight, lb. 18.0		55	6.5	10.5	16.5
		Diameter, in. 32.6		80	4.9	8.0	13.0
Fisk	28	Fabric, non-skid	1919	30	8.8	14.4	22.8
		Weight, lb. 19.5		55	6.1	10.5	17.4
		Diameter, in. 33.0		80	5.0	8.6	14.0
Firestone	16	Cord, non-skid	1917	30	8.4	12.5	18.9
				55	6.1	9.3	14.0
				80	5.5	8.5	13.3
Goodrich	14	Cord, ribbed	1917	30	6.5	10.5	17.2
				55	5.5	8.0	12.5
				80	4.8	6.8	10.2
Goodyear	15	Cord, non-skid	1917	30	7.8	12.3	19.5
				55	6.0	8.9	13.6
				80	5.0	7.4	11.3
Miller	6	Cord, non-skid	1917	30	8.2	12.1	19.5
				55	5.2	8.0	14.0
				80	4.5	7.0	11.2
U. S. Royal	8	Cord, non-skid	1917	30	8.0	12.1	19.8
				55	4.9	7.5	12.7
				80	3.9	5.9	10.8
Fisk	26	Cord, ribbed	1919	30	5.9	8.6	15.0
		Weight, lb. 21.0		55	4.5	6.5	10.6
		Diameter, in. 33.3		88	3.8	5.5	8.5
Fisk	25	Cord, non-skid	1919	30	5.7	9.0	14.0
		Weight, lb. 21.5		55	4.7	7.2	11.4
		Diameter, in. 33.3		80	4.0	6.0	10.3
Fisk	29	Cord, non-skid	1922	30	7.2	9.5	15.0
		Weight, lb. 22.0		55	6.3	8.0	11.7
		Diameter, in. 33.3		80	5.5	7.3	10.8
Fisk	30	Cord, non-skid	1922	30	6.5	9.5	16.3
		Weight, lb. 22.2		55	5.0	7.6	12.5
		Diameter, in. 33.3		80	4.6	6.8	11.3
Yale	31	Cord, non-skid	1922	30	9.0	12.5	22.0
		Weight, lb. 25.0		55	7.0	9.5	15.0
		Diameter, in. 33.75		80	6.5	8.5	13.0

TABLE 13—ROLLING-RESISTANCE OF 35 X 5-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb per Sq. In	LOAD, LB. 700	1,000	1,300	1,600
Goodyear	100	Fabric, non-skid	1920	40	12.8	20.0	27.5	35.0
		Weight, lb. 31.0		55	11.2	17.5	24.1	30.5
		Diameter, in. 35.9		70	10.0	15.8	21.7	27.7
				90	9.5	15.0	20.6	26.2
Fisk	102	Fabric, non-skid	1920	40	11.5	19.6	27.8	36.0
		Weight, lb. 33.5		55	10.5	17.4	24.5	31.5
		Diameter, in. 36.0		70	9.7	15.6	21.5	27.3
				90	9.4	14.4	19.3	24.0
Yale	104	Fabric, non-skid	1920	40	14.0	22.8	31.5	40.5
		Weight, lb. 36.25		55	12.7	19.5	26.2	33.0
		Diameter, in. 36.0		70	11.7	17.5	23.3	29.2
				90	10.7	15.5	20.4	25.2
Goodyear	101	Cord, non-skid	1920	40	9.0	13.5	18.5	23.2
				55	7.3	11.2	15.3	18.3
				70	6.0	9.6	13.5	17.3
				90	5.0	8.5	12.4	16.0
Fisk	103	Cord, non-skid	1920	40	8.5	12.8	18.2	24.0
		Weight, lb. 35.5		55	6.8	11.1	15.5	20.0
		Diameter, in. 37.2		70	5.9	9.6	13.4	17.2
				90	4.8	8.5	12.0	15.5
Fisk	109	Cord, non-skid special	1921	40	......	11.5	15.5	20.0
		Weight, lb. 37.0		55	......	10.3	13.6	17.0
		Diameter, in. 37.2		70	......	9.0	12.0	14.7
				90	......	8.3	11.0	13.5

Temperature and Heat

That rubber tires get warm while running is well known. The true source of the heat is now known to be within the tire structure. It may be due in part to compression of the rubber or in part to the flexure of the carcass or both. Heat generation is not confined to pneumatic tires but is found in the same degree in those of solid-rubber. This proves that compression of the rubber unaided by flexure of the tire walls will result in heat.

TABLE 14—ROLLING-RESISTANCE OF 33 X 4½-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb. per Sq. In	LOAD, LB. 700	1,000	1,300
Fisk	59	Fabric, plain	1919	30	13.0	21.0	28.7
		Weight, lb. 20.0		45	10.5	16.5	22.7
		Diameter, in. 33.2		65	9.5	15.0	20.5
				90	7.7	12.5	17.0
Fisk	60	Fabric, non-skid	1919	30	12.0	20.0	28.0
		Weight, lb. 23.5		45	10.5	17.0	23.5
		Diameter, in. 33.7		65	9.5	15.5	21.5
				90	8.5	13.8	19.0
Firestone	55	Cord, ribbed	1919	30	7.5	12.3	17.0
		Weight, lb. 30½		45	6.0	9.6	13.3
		Diameter, in. 34.8		65	5.3	8.5	11.7
				90	5.0	8.0	11.0
Firestone	54	Cord, non-skid	1919	30	7.5	12.0	16.5
		Weight, lb. 30.5		45	7.0	11.0	15.0
		Diameter, in. 34.9		65	6.1	9.8	13.3
				90	5.5	8.7	11.9
Goodyear	68	Cord, ribbed	1919	30	7.6	12.8	18.0
		Weight, lb. 26.5		45	6.0	10.5	15.0
		Diameter, in. 34.1		65	4.8	8.0	11.5
				90	4.3	7.3	10.5
Goodyear	56	Cord, non-skid	1919	30	9.2	13.7	18.3
				45	7.3	11.0	14.7
				65	6.5	10.0	13.4
				90	5.7	9.2	12.5
Fisk	61	Cord, ribbed	1919	30	6.7	11.3	17.5
		Weight, lb. 24.0		45	6.2	9.5	14.0
		Diameter, in. 33.6		65	5.6	8.5	11.9
				90	5.3	8.0	10.9
Fisk	65	Cord, non-skid	1919	30	7.5	13.0	18.7
				45	6.3	10.3	14.0
				65	5.5	8.7	11.7
				90	5.3	8.5	11.4
Fisk	72	Cord, non-skid	1922	30	10.5	16.0	25.5
		Weight, lb. 27.0		45	8.8	13.0	19.2
		Diameter, in. 34.4		65	7.7	11.5	17.3
				90	7.3	10.7	16.0
Fisk	73	Cord, non-skid	1922	30	11.5	16.5	22.5
		Weight, lb. 29.5		45	9.0	13.4	18.0
		Diameter, in. 34.4		65	7.7	10.8	16.6
				90	6.5	9.5	14.8

The flexure of a tire can be likened to the bending and unbending of annealed wire, where motion in any direction absorbs work and generates heat within the material. An ideal tire structure would be one of perfect elasticity, where the work of flexure or compression would be completely returned after the original positions were resumed. Present tire structures are at least partly elastic as indicated by the fact that cord tires have only two-thirds the rolling resistance of fabric tires; hence they have only two-thirds the heat-generating capacity.

It can be assumed fairly that the generation of heat and the resulting rise of temperature are directly associated with the rolling resistance of the tire. It can be assumed also that the amount of heat generated is exactly equivalent to the work of the rolling resistance and can be computed from this work, as in the case of the friction brake. This suggests a simple but practical way of testing the rolling resistance of two tires, which is to place them on the opposite wheels of a car and observe the surface temperatures after a long run. Measurements of the surface temperature are unsatisfactory as a final measure of tire resistance, although they may have some value as a check.

The surface temperature can be computed, approximately, from the rolling resistance. Useful conclusions can be drawn from such computations. It can be shown, for instance, that a large-size pneumatic-tire will heat more than a small one. The heat equivalent of the rolling resistance in British thermal units per hour can be found by obtaining the product of the resistance times the speed times 2546/375. The heat dissipated can be expressed as the product of the cooling surface or area times the difference of temperature between this surface and the surrounding air times a coefficient. The coefficient of heat transfer is fairly well known from experiments on moving air over surfaces, such as hot-blast heaters, automobile radiators and the like.

Applying this method to a 32 x 4-in. cord tire at a speed of 30 m.p.h., a computed rise of temperature of 29 deg. fahr. was obtained, using a normal full load of 1000 lb. Applying the method to a 44 x 10-in. cord tire at a speed of 30 m.p.h., with a load of 7000 lb., a computed rise of temperature of 90 deg. fahr. was obtained. This increase of temperature is due to the fact that the load capacity and the resultant heat of tire-friction increased faster than the surface for heat dissipation. The computed temperature-rise of small-size tires has been satisfactorily checked on the Mason Laboratory drums, but thus far no observations have been made on the large sizes.

Gearbox Lubricant

The transmission friction may be materially affected by the temperature of the lubricant in the gearbox. This is true especially of trucks, where the transmission friction normally is large. Tests were made by leaving the truck exposed all night to November weather, then running the rear end on the drums and making friction measurements at intervals as the lubricant warmed-up. Runs were continued until the lubricant temperature had risen from 35 deg. fahr. to about 135 deg. fahr.

Fig. 15 shows the observed rise of temperature and the drop in the rear-end resistance of three trucks. In two cases the rear-end friction was increased 50 per cent by the coldest lubricant. The effect of friction is to produce heat that warms the gearbox and its contents, thus automatically reducing the friction. A temperature of 140 deg. fahr. in the gearbox was observed that was produced solely by internal friction. No attention was paid to the gearbox temperatures of passenger cars. All tests were made at the ordinary room-temperature with the engine

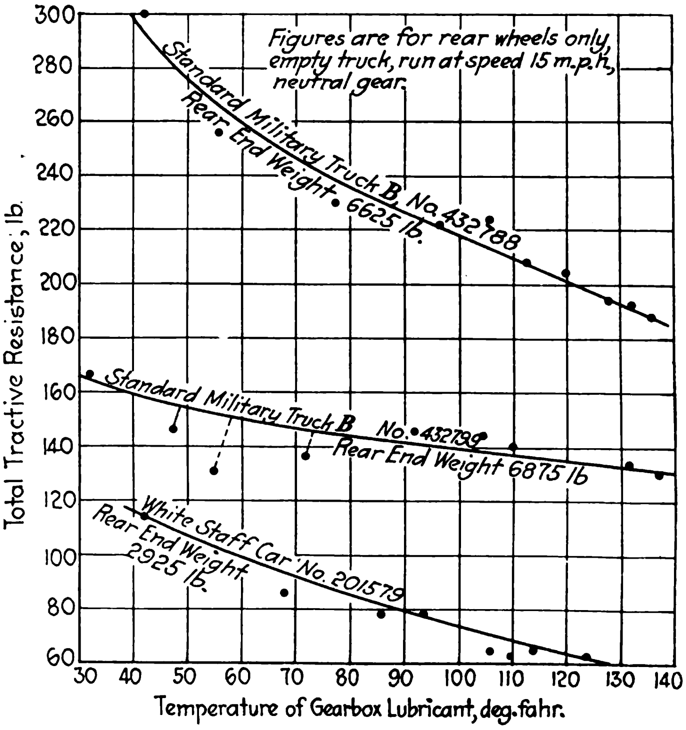


FIG. 15—CURVES SHOWING THE TRACTIVE RESISTANCE OF TRUCKS AT DIFFERENT TEMPERATURES OF THE GEARBOX LUBRICANT

and the transmission well warmed by preliminary running.

THE DISCUSSION

MAJOR M. L. IRELAND:—I would like to express the appreciation of the engineering division of the National Research Council for the very cordial cooperation that we received from Yale University in this work. We doubled the greatest pulling load that they had ever had on this apparatus. The maximum force at the tire surface illustrated by Professor Lockwood was about 600 lb. When we put the 7½-ton Mack truck on it, the pull ran to 1100 lb. The Quartermaster Corps' standardized Class B truck made it 1000 lb. The work done by Professor Lockwood was an essential part of our research because we were endeavoring to tear apart the group of forces that enter into what is known as tractive resistance, and it appears from a very careful survey that we have made of the literature of Western Europe and America on the subject of tractive resistance that there has been no really successful effort to separate that group of forces into its component parts; although Riedler has separated laboratory results, as distinguished from road results, into component parts. It is only by some such method as Professor Lockwood's, particularly when we are dealing with the gasoline-driven vehicle, that we can get at these results. When these results are published, the first thing that engineers will say is, "Why did you not run the speed test out to a greater range?" The answer will be that we went to the limit and beyond the capacity of the equipment. A survey of the equipment installed elsewhere indicates that no one has equipment that will withstand these big heavy trucks under full load and at high speed. One of the developments that I think the next few years will bring is the redesigning and rebuilding of apparatus similar to Professor Lockwood's to handle these heavy loads. Professor Lockwood is leading the way in a creditable manner.

Many of the best researches conducted in the past have wholly overlooked the matter of the degree to which the results are affected by the change of temperature in the differential. In scanning two or three researches that are regarded as practically the last word, we found that the investigators have not taken this into account, and that therefore their results may be affected seriously.

An interesting result in regard to this temperature-effect is this: we made six runs about 4 miles out of New Haven with the Mack 7½-ton truck, one run after the other, a total elapsed time of 82 min., with absolutely no change of any kind in the conditions other than the change in the temperature of the differential and, between the third and fourth runs, we opened the governor so that the speed at the crest of the hill was increased from about 11 m.p.h. to about 17, and the maximum speed from 18 m.p.h. to 21. When we came to take a mean curve for the first three runs, and a mean curve for the second three, we found altogether different results. Fortunately, there was a thermometer in the differential case that showed a change in the temperature of about 30 deg. fahr. According to Professor Lockwood's laboratory work of the day before, this would indicate a change of 15 m.p.h. or 11 tons in tractive resistance. When that correction is made on the chart at 15 m.p.h. it accounts very satisfactorily for the difference between the two final tractive-resistance curves. There is perhaps as much as 9 lb. per ton difference at the extreme ends and about 12 lb. in the middle, so that the laboratory work has borne out in a very satisfactory way many of the results that we are obtaining by actual measurement on the road.

The work of measuring tractive resistance covers a wide range of tires running over a wide range of loads. We have made tests on five different kinds of road thus far.

HERBERT CHASE: — Professor Lockwood's excellent paper represents an enormous amount of work. I found much useful information in the data that he has accumulated and prepared an article entitled Rear Wheel Dynamometer Tests and Their Significance to the Engineer.[3] I had an opportunity to pick out from some 200 tests that Professor Lockwood made, about 20 tests which seem to be fairly representative of the passenger-car field in particular and include also three or four trucks. The data that I secured are tabulated in the article referred to and the curves drawn for comparative purposes are shown therein.

The figures on wind-resistance in the table are computed, as already explained, by determining the projected frontal area of the car and multiplying the number of square feet by 0.003 and the square of the speed in miles per hour. This, of course, is only an approximation since the wind-resistance is affected materially by the streamlines of the body, but it serves well enough for comparative purposes. According to the figures given regarding these 15 to 20 passenger cars, the frontal area varies from 16 to 32 sq. ft. in the passenger cars listed. These areas no doubt could be decreased in some cases by improved body-design. Some means of actually measuring wind-resistance is needed before the actual resistance of certain other types and sizes can be predicted with certainty. Wind-resistance becomes an important factor at speeds as low as 20 m.p.h.

The method of obtaining figures for tractive effort on level roads used by Professor Lockwood is to add to the wind-resistance the resistance of the front wheels of the car. This gives the pull that must be exerted at the periphery of the driving-wheels to propel the vehicle on a hard, level road at the various speeds indicated.

In testing cars on the dynamometer, the brake is set to give loads equivalent to the tractive resistance on the level road at the respective speeds, and the fuel-consumption is measured. The two factors that oppose the forward motion of a car on a level road are the wind-resistance and the rolling-resistance of the front wheels, the rear wheels and the elements connected to them.

The power delivered by the rear tires is the power that remains after overcoming the losses from the engine back to the rear wheels including the power lost in the rear tires, and that is precisely equal to W, the wind-resistance, plus R, the front-wheel resistance; R can be measured accurately by the method Professor Lockwood has described. The value of W is computed from the formula mentioned and is only an approximation. I suggest a more accurate method of obtaining the factor W. It is a method which will take into account the streamline form of the body; in short, measure its actual resistance. Quoting from my article

> The brake horsepower that the engine is required to develop for level-road operation at average driving speeds is small compared to its maximum power at these speeds. The average at 20 m.p.h. for 17 passenger cars is about 5.4 hp. as against a maximum average at the same speed of 23.8 hp. In other words, under what can be considered a normal operating condition, the average engine develops from one-fifth to one-quarter of the power it is capable of developing at that speed. Even the heaviest of the passenger cars tested requires only 7.55 b. hp. to propel it at 20 m.p.h. on a level road,

[3] See *Automotive Industries*, April 20, 1922, p. 859.

while some of the lighter cars require only a little over 4 b. hp. It seems almost incredible that if facts such as these had been more generally appreciated, so little attention would have been given to making engines more efficient at the light loads obtained in normal use.

It is instructive to note from the tabular data the relative importance of the three losses, which together exactly balance the power developed at the flywheel of the engine. These are (*a*) the rolling-resistance of the front wheel, (*b*) the power losses between the engine and the driving surface of the rear tires, including the losses in these tires, and (*c*) the wind-resistance.

The method of measuring the brake horsepower, in the test that Professor Lockwood makes, is to drive the rear wheels from the drums to which power is applied by the electric motor below the floor. The power is transmitted through the tires and the bevel gears to the upper shaft of the transmission. The gears are placed in neutral. There is some loss due to the churning of the lubricant in the gearcase. There is also a loss in the gearing, which may or may not be the same as that which occurs when the engine is driving the car under load. In other words, the gears are running light as compared to running under load when driving the car. Furthermore, there probably is a certain difference between the power absorbed by tires that are merely rolling and tires that are driving. For these reasons the power losses thus secured, when added to the rear-wheel horsepower, probably do not give exactly the brake horsepower of the engine, but for comparative purposes they are doubtless good enough. If Professor Lockwood had the equipment, it would be interesting to know just how much greater this loss would become if the gears and parts that transmit the power were actually loaded.

In regard to wind-resistance, a suggestion is made in reference to a method of measurement, more accurate than using the computation based on the frontal area. To do this the car should be equipped with a vacuum gage, connected to the inlet-manifold, and an air-speed meter. It would then be taken to a stretch of level road and operated at the desired speeds, checked by speedometer and stopwatch. In each case the reading of the air-speed meter and the vacuum gage would be noted, the former indicating any head or tail wind that might affect the results. The car would then be brought back to the laboratory and tested at the same speeds and throttle positions, as indicated by manifold depression, and the rear-wheel power measured. The difference between this rear-wheel power and the power absorbed by the front wheels would be the actual power used in overcoming wind-resistance at the respective speeds.

To secure a more certain determination of the brake horsepower, and consequently a more accurate method of measuring the losses between the engine and the rear wheels, requires equipment that may not now be available in the Mason Laboratory and that may not be as easily connected up and used as the present equipment. I am under the impression that it would be possible in some cases to make a connection to the front end of the engine in the place of the starting-crank, and by a light prony brake, or some other form of brake, measure the brake horsepower of the engine directly. Measuring the difference between the brake horsepower and the rear-wheel horsepower, as it is now measured, would afford a means for precise determination of the losses between the engine and the periphery of the rear tires.

The losses in the rear tires can probably be assumed to be approximately equal to the losses in the front tires, if the tires themselves, the inflation-pressures and the weights on the tires are the same, as they would be for purposes of calibration.

E. FAVARY:—Will Professor Lockwood give particulars regarding the difficulties he experienced with the drawbar-pull tests of his earlier dynamometer?

As to the losses in tires, some of the results I found a few years ago probably hold good at present. Professor Lockwood showed that a partly worn solid tire had less rolling-resistance than a new solid-rubber tire. This is easy of explanation when we consider that the tests are made on a smooth drum. If the road surface were perfectly smooth, a solid-steel tire would, I think, show the least resistance. However, the rougher the road is, the softer must be the pneumatic tire to show the least rolling-resistance, because the most perfect tire is that which will absorb the average height of obstructions on the road without raising the axles. The solid tire has an increased loss on account of the low elastic efficiency of rubber.

PROF. E. H. LOCKWOOD:—Mr. Favary has raised an interesting question on the relative resistance of solid-rubber and pneumatic tires on smooth and on rough surfaces. His doubts that solid-rubber tires on rough roads would roll as easily as fabric pneumatic tires seem well justified. Experiments are now in progress in three different States for road resistance of loaded trucks on smooth portland-cement roads. Preliminary figures from all of the tests appear to indicate that solid-rubber tires have actually lower rolling-resistance than fabric tires on passenger cars. Data are not at hand for badly worn roads, where doubtless the solid-rubber tires would offer relatively more resistance.

THOMAS S. KEMBLE:—Has Professor Lockwood been able to determine the average overall efficiency of passenger-cars and trucks?

PROFESSOR LOCKWOOD:—Questions about efficiency are difficult to answer for the simple reason that the efficiency of a vehicle has not been defined. I should therefore begin by defining the term as the ratio of the power delivered at the periphery of the rear tires to the power developed at the clutch. According to this definition, if the friction losses between the clutch and the rear tires could be reduced to zero, the efficiency would be 100 per cent. On the other hand, if the engine is idling with gears in neutral, the efficiency will be zero, because in that case no power is delivered to the rear tires. Average efficiencies under normal operating conditions can be computed for typical vehicles, as shown in the several

TABLE 15—AVERAGE MOTOR-VEHICLE EFFICIENCIES UNDER NORMAL OPERATING CONDITIONS

Vehicle	Kind of Tire	Tire Size, In.	Weight, Empty, Lb.	Speed, M.P.H.	Efficiency, Per Cent	
					Level Road	Steepest Grade
Light Touring Car—Dodge	Cord	32x4	2,655	20	41 ÷ (41 + 25) = 62	295 ÷ (295 + 30) = 91
Heavy Touring Car—Cadillac	Cord	35x5	4,395	20	61 ÷ (61 + 56) = 52	577 ÷ (577 + 53) = 90
Heavy Truck—Standard, Class B	Solid Rubber	....	11,540	15	111 ÷ (111 + 127) = 47	588 ÷ (588 + 135) = 81

examples chosen from tests at the Mason Laboratory that are given in Table 15.

It appears that the efficiency as defined may extend from zero to a maximum of 90 per cent. It is gratifying to see that an efficiency of 90 per cent can be realized, but it must be remembered that under ordinary running conditions the figure is much lower, say 50 to 60 per cent.

MR. KEMBLE:—Did you give the tractive resistance of the front wheels separately from the rear wheels? Is your comparison of the cord, fabric and solid tires for the front or for the rear wheels? It seemed to me that it was for the rear wheels and, unless there is some further explanation, we might be led astray. There are transmission losses which, if included, affect the percentages; whereas, if the comparison were made on the front wheels, the loss in the bearings being comparatively small, the percentages would be more accurate.

PROFESSOR LOCKWOOD:—That is an important point. When we measured the frictional resistance of pneumatic tires, whether on the front or rear wheels, we invariably got the same results. Ordinarily, the front-wheel bearing-resistance is from 4 to 6 lb.; the rear, including transmissions, from 15 to 20 lb. This resistance, subtracted from the total, leaves the difference we call tire resistance. We believe that the tire resistance is the same whether measured on the front or the rear wheels. Our tests have been made on the rear wheels, as a matter of convenience in changing the loads.

MR. KEMBLE:—You take as the tire resistance of the rear end the difference between the resistance with the load on and the resistance when the wheels have been raised so that they just touch the drum; has that been checked to a certain extent with the resistance that you get on the front wheels?

PROFESSOR LOCKWOOD:—Yes.

C. F. SCOTT:—Is the large-diameter drum essential to successful results? Could satisfactory results be expected with, say, 3½-ft. drums instead of 5-ft.; and with steel drums?

PROFESSOR LOCKWOOD:—I do not feel very positive on this point. Whether it could be predicted that the small-size drum would give different results, I am not prepared to say. It would be an interesting experiment to have the same vehicle tested on the steel drums at Worcester, and then on the Mason Laboratory drums, to see whether the results in one place would be the same as those in the other. When Mr. Favary's apparatus at Cooper Union, New York City, is completed, it would be instructive to have it compared with the others.

MR. SCOTT:—To repeat Mr. Favary's question, why did not the drawbar measurements work out?

PROFESSOR LOCKWOOD:—Like most experimenters, we tried to measure this resistance directly. The first thing we discovered was that the front wheels, resting on the floor, varied the drawbar pull by from 15 to 20 lb. To avoid this error it would be necessary to raise the front tires, allowing the front end to roll on anti-friction rollers, as was done by Riedler in Germany.

We tried to suspend the front end with our overhead crane but found difficulty with side-sway and abandoned the trial. Another thing is the difficulty of keeping the contact of the rear wheels directly over the center of the drums. If it gets slightly off-center, there is a gravity component which has some effect. Due to such troubles, we discarded the attempt to measure the drawbar pull directly and used the friction-brake pull instead.

R. E. PLIMPTON:—How much would the frictional resistance increase with the diameter of the rear wheel, if the load were constant; say with tires between 34 and 40 in.?

PROFESSOR LOCKWOOD:—It is my offhand impression that it would not make much difference. I do not know of any experiments to determine the effect of tire diameter on the rolling-resistance on smooth roads. I suspect that the effect produced by any change of the diameter is very slight, perhaps too small to measure.

CORNELIUS T. MYERS:—The old subject of the gasoline engine working at low efficiency in passenger cars and trucks is still with us and probably will be for some time. In 1910 and 1911, in the design of a new line of trucks, it was very thoroughly threshed out in one instance, and an effort was made to impose on the sales department a line of trucks with much smaller engines than had been considered good practice up to that time. We encountered considerable "rolling-resistance" in the sales department, but a tractive-factor formula was developed[4] which took into consideration the various factors that Professor Lockwood has set forth.

In this tractive-factor formula the resistance was measured in fractions of a pound per pound of weight. Professor Lockwood has suggested that instead of using pounds per ton, we use pounds per thousand pounds. I should like to go a little farther and use a plain pound coefficient. It will be a decimal of a pound, of course, per pound of weight. That is directly comparable with the grade coefficient as say 2 per cent, or 0.02 lb. per lb.

PROFESSOR LOCKWOOD:—It seems to me 100 would be more logical. Then 10 lb. per M would be 1 lb. per 100 lb. Why not take 100?

MR. MYERS:—For the reason just stated. In going into the subject I found nothing whatever in this Country, but in France and Italy the road coefficient used is kilograms of resistance per kilogram of weight. It is understandable and is the ordinary method of expressing a coefficient.

The wind-resistance for any car or truck can be determined indirectly. With how great accuracy I cannot say because I have not made enough experiments. In looking at Professor Lockwood's illustration, we see that if the car were allowed to coast, after it reached a certain speed there would be the resistance of the front and rear wheels and of the axle and transmission, if the gear is in neutral and the engine shut-off.

Professor Lockwood has obtained what we have had heretofore only approximately from Riedler's experiments. He has given us the resistance of the tires. One thing is left and that is the wind-resistance. You can get that from the accelerometer readings, from which the above resistances must be deducted. When the car is run in opposite directions, with and against the wind, if the gear is placed in neutral and the other four resistances are deducted, the wind-resistance can be determined. It is a definite and understandable figure. In 1912 and 1913 we checked the rolling-resistance of cars with the accelerometer and found, as Professor Lockwood did, that if the truck is in good condition, the rolling-resistance is about 40 lb. per ton and that of the passenger car is about 30 lb. per ton at 15 m.p.h.

M. C. HORINE:—We have talked about the mechanical efficiency of the vehicle, the wind-resistance and the tire-resistance. There is another factor, the road-resistance. Professor Lockwood's tests were performed under ideal conditions so far as the road is concerned. Still another is the friction generated by the spring-suspension of the vehicle. It takes power to lift a vehicle bodily into the air and let it down again. That is the acceleration and

[4] See S. A. E. HANDBOOK, vol. 2, p. 67.

deceleration of the mass. There is friction in the springs and the spring-shackles. It takes power to move those parts, although the power is not applied directly. The motion of the parts and the wear that results cause the consumption of driving power. Consequently, more than a laboratory test is needed to determine the exact amount of power required to move a car over a given road. Has Professor Lockwood considered the matter of springs in the power losses in the suspension of the vehicle?

PROFESSOR LOCKWOOD:—That is a topic which is rather new to me. I am not sure that it would be deserving of attention. The measurement of it would be a nice problem.

MAJOR IRELAND:—We are getting some interesting results from road tests, but it is early yet to give definite predictions, much less laws and formulas. One interesting test that we happened to make was on some granite blocks between street-car tracks down a certain hill. We did not know at the time that we had crossed the line between the cities of Everett and Malden, Mass. In Everett the blocks have cement grouting between the joints and the road surface is smooth; on the other side of the city line in Malden, the joints are sand-filled. There is a marked difference in the way some of these granite-block sets stand above one another in the two cities, and there is a marked difference in the resistances at the same velocity in two parts of the run. We have photographed several sections of the road and measured the degree of roughness with a 10-ft. straight edge.

If the amount of energy absorbed in the spring-suspension, frame, tires and other vehicle parts is subtracted, there would be included in the remainder that part of the road-test results which is due to the road itself, and this would show in a much clearer way the comparative effects of the sand-filled and grout-filled joints. The separation of the tire-resistance and the resistance of the transmission system is not difficult. How to separate the effect of the increased impact absorbed by the vehicle parts from that which is absorbed in the road material and therefore belongs in what is called road resistance is a problem. About the only means of doing this simply seems to be to deduct very carefully all tire and transmission-system resistance that is due to other causes, and then, from the principle that action and reaction are equal but opposite in direction, divide the remainder, which should be impact effect, into one-half absorbed by the vehicle parts and one-half absorbed by the road material. It will be seen that, in practice, some serious difficulties will be encountered in doing this.

AUSTIN M. WOLF:—The results obtained in these tests would be comparable to an ideal roadway. Besides the energy expended in the inter-leaf friction and the friction in the spring-shackles, much energy is absorbed in the moving of the great mass represented by the weight on the springs. The universal-joints are working at greater and irregular angularities; the tires are flexed to a greater extent than on the dynamometer drum; the air within the tires is compressed whenever a bump is met; and, in addition, there is the side-slipping and the waste of energy caused by it. After striking an obstacle, one wheel that is in the air is speeded up through the differential and when the tire returns to the ground the tread is chafed. All these factors may be small severally, but collectively they represent a considerable amount of energy.

Taking these things into consideration, Mr. Chase's method of measuring air-resistance would give a result too high, as the losses indicated on the dynamometer would be less than those actually encountered on the road.

I suggest that in future tests an attempt be made to use a dynamometer drum having a detachable tread so that various irregularities in the road could be represented. This would be somewhat similar to the method used some time ago in attempting to run a destruction test on aluminum wheels. It would be interesting to have each rear wheel rest on an independent drum and the speeds varied to reproduce the action of a car in turning a curve. The losses due to the differential might thus be determined.

It probably would be difficult to make accurate measurements on a dynamometer with a detachable track on the drums, but I believe that a careful analysis of the problem would result in adequate equipment. To get more accurate data in road tests, it occurred to me long ago that we are not measuring the development and expenditure of energy by a simple and direct means. I suggest that the engine be cradled in the chassis and the torque reaction be measured directly in this way; similarly, with a separately mounted transmission it also can be cradled and the torque reactions noted, when in other than high gear. With a torque-arm attached to the rear axle, the reaction at its point of anchorage on the chassis can be measured. All these parts should work on a recording type of instrument, as it would not be practicable to take visual reading at all these points during a test.

PROFESSOR LOCKWOOD:—The point raised by Mr. Wolf that the losses indicated on the dynamometer would be less than those actually encountered on the road, seems reasonable. It is surprising, however, that for the few vehicles thus far tested on the Mason Laboratory dynamometer in comparison with new cement roads, the road-resistance appears to be somewhat lower. It must be remembered that this comparison involves the deduction of the wind-resistance from the road results, and thus far wind-resistance has been computed by a formula and is rather approximate.

The dynamometer results are less ideal than might be expected. The Mason Laboratory drums are not perfectly round. The variation of the radius at present is about 1/16 in. Due to this and variations of the tires themselves, most cars run on the drums with a slight up-and-down motion, which undoubtedly absorbs power just as on the road.

No mention has been made of the continual rolling of the tire on the curved drum surface. This is a different condition from rolling on a flat surface, and probably the curved surface will offer more resistance.

A. E. F. ARTILLERY GUNFIRE

ABOUT 2500 guns of all calibers, nearly 700 of which were heavy cannon, were with the First Army American Expeditionary Force, during the Meuse-Argonne offensive. During the battle these guns fired about 4,215,000 rounds, or a greater number of rounds than was fired by the Union forces during the entire Civil War.—*Journal of the United States Artillery.*

Electric Wiring for Automobiles

By William S. Haggott[1] Buffalo Section Paper

THE importance of the electrical wiring on automotive apparatus is indicated by the fact that, in effect, a central electric-power station, as well as its distributing lines and customers, is consolidated upon a high-speed vehicle and the system, inclusive of high-tension and low-tension circuits, direct and alternating currents, generators, transformers, motors, storage-batteries, relays, circuit-breakers and fuses, is expected to function properly every day under all conditions.

The three main divisions of electric wiring for automobiles are high-tension ignition, lighting and starting circuits. The present practice with regard to each division is discussed and suggestive comment is made with a view toward improvement, one feature worthy of serious consideration being that of the adoption of a standard color-scheme whereby the character and service of each separate circuit on any car would be indicated by the color of the wires used.

THE Standards Committee of the Society has given some consideration to the possibility of standardizing methods of installing wiring on automobiles, with the purpose of adding to the usefulness of the existing S.A.E. Standard for Insulated Cable,[2] covering the construction of individual cable. Within certain limits it seems feasible to do something along this line, and it is hoped that this paper and its discussion will serve as a basis for further efforts. I am indebted to various members of the Electrical Equipment Division for much of the information I have used in preparing the paper. Their cooperation has been of great assistance.

We are all more or less familiar with industrial electric powerplant and transmission-line design and construction. We also know that at the receiving end of the transmission line we have energy which is available for light, heat and power. However, the average person seldom stops to think that on present-day automobiles we have practically all of the electrical features that are found in every zone served by a central station. We have high-tension and low-tension circuits; direct and alternating currents; generators, transformers and motors; storage-batteries, relays, circuit-breakers and fuses. In fact, we have the central station as well as its distributing lines and customers all consolidated upon a high-speed vehicle, and the system is expected to function properly every day, in all sorts of weather and under all conditions.

With the foregoing thought in mind, even the layman must readily see the importance of the electrical wiring on automotive apparatus. A broken wire, a loose terminal, a short-circuit or any one of a number of things, can cause delay, inconvenience and often danger. However, automobile cable construction and its installation cannot be patterned after regular light-and-power cable construction and installation. The conditions under which automotive cable does its work are entirely different from those imposed upon light and power cables. Automobile wiring falls naturally into the three main divisions of high-tension ignition; lighting, including low-tension ignition; and starting circuits.

High-Tension Ignition

Present practice and opinion are about evenly divided as between plain rubber-covered and braided rubber-covered high-tension cable. However, I believe the tendency is strongly toward the braided construction, provided the braid is thoroughly filled with a durable flexible insulating varnish. The principal objection to braided cable is that the braid retains moisture, thereby creating a possibility of erratic firing, but this objection can be overcome if the braid is properly treated, and considerable progress is being made in this direction. Braided cable, properly constructed, affords protection from oil, heat and ozone, all of which seriously damage a plain rubber-covered cable. Ozone is the worst of the three, as it causes the rubber to crack and results in misfiring or irregular firing.

If the high-tension wires are short and can be kept free from injury, it is preferable to run them open and not close together. This method gives positive protection against inductive disturbances and leakage between the wires, and will prevent "cross-firing." I believe that in many cases the spark-plug wires can be run open to good advantage, but there are instances, especially in connection with 6, 8, and 12-cylinder engines, some of which have two spark-plugs per cylinder, where it becomes almost and perhaps actually imperative to run the wires in tubes.

Metal tubes are usually employed for this purpose on account of cost. A fiber tube is better, as it will prevent leakage to the ground in case the cable insulation fails. To nullify the condenser action when a fiber tube is used, it is only necessary to run a bare wire through the tube and ground this wire to the engine. Where a metal tube is used, it must be well grounded to the engine so as to carry any static or condenser effect to the ground. The openings in a metal tube must be properly rounded or bushed to prevent cutting the cable insulation. If possible, the coil should be located close to the distributor in battery-ignition systems, thereby keeping the wire from the coil to the distributor short. Where a tube is used for the spark-plug wires, it is best to keep the wire from the coil to the distributor out of the tube.

Lighting and Low-Tension Ignition

There is a strong tendency toward the use of armored cable on lighting circuits. This is particularly true where wires are exposed and subject to chafing. Armored cable, if properly constructed and installed, will give no trouble. The most important factor in the use of armored cable is to be sure that the armor is stripped back sufficiently from the terminals and then soldered down so that a ground cannot occur at these points.

Rubber-covered-and-braided or varnished-cambric-and-braided cable protected by flexible metallic conduit is sometimes preferred. There is also some use of non-metallic conduit to carry these types of wire. The use of metallic conduit perhaps gives a more substantial appearance on head and tail-lamp wires, but is more expensive

[1] M.S.A.E.—Cable sales manager, Packard Electric Co., Warren Ohio.

[2] See S.A.E. Handbook, vol. 1, p. B 33.

than armored cable in both first cost and installation.

It is unnecessary, of course, to have metallic protection for wiring around the instrument-board. I have in mind several installations in which rubber-covered-and-braided cables have been used for years on all circuits, including those for head and tail-lamps, without the use of additional protection, but I believe the use of armored cable is good insurance against lighting-circuit failures.

The question came up recently as to the advisability of running separate head-lamp feed-wires from each head-lamp clear back to the lighting switch, instead of making a splice at a point near the front end of the car and having only one wire from the switch to the point of the splice. The latter method has practically become standard. The splice can be properly made and supported and, when so constructed, will give no trouble. It is better not to wire instrument-lamps in series with tail-lamps as is sometimes done. The only advantage in the series arrangement is to indicate when the tail-lamp is burned-out, and this fact can be checked up easily by looking back for the reflection of the tail-lamp. With tail and instrument-lamps in series, it is necessary to use 3-volt bulbs, which are rather special and sometimes hard to obtain. The series arrangement also requires continuous burning of the instrument-lamp, which is fatiguing to the driver's eyes. With separately controlled lamps, the instrument-lamp can be turned out when the car is parked, and can be substituted for the tail-lamp if it should burn out.

Where fuses are used, each circuit should be protected, even though this is slightly more expensive than to have one fuse control several circuits. With each circuit fused separately it is easier to locate trouble and, when circumstances require, it is possible to operate the car with one circuit out of commission. Considerable attention is being given to the use of a single circuit-breaker placed in the main battery-feed circuit. This method of protection has the advantage that, when the cause of trouble is removed, the circuit-breaker can be closed and the car can proceed without resorting to the installation of makeshift fuses, which is always dangerous.

It is good practice to consolidate lighting and low-tension wires wherever possible and to braid them together or run several wires through one piece of conduit. This method is economical and convenient and tends toward rugged and safe installation. If the proper wire is used and the individual wires are assembled in a workmanlike manner, installations of this sort give little or no trouble from the repair-shop standpoint. This use of assemblies, or wiring harnesses, as they are often called, is admirable from a car-assembly standpoint. The wiring harnesses are formed-up on a bench, which permits uniformity and careful workmanship and a more satisfactory wiring job on the car.

I suggest that some effort be made to adopt a standard color-scheme for various circuits on all cars. For example, every car has a wire running from the battery or starting-switch to the ammeter. Is there any reason certain definite colors cannot be assigned to this and other wires that are common to all cars, so as to have as nearly as possible a standard wiring color-scheme among all car builders? It seems that this would be of great value to the car driver and repairman and provide a uniformity that would be helpful to the industry in many ways.

Starting

The most general practice is to use rubber-covered-and-braided cable. On 6-volt systems, No. 1 American wire gage wire is the prevailing size, although considerable No. 0, as well as some No. 2, is used. The gage size must be right in order that proper starting-torque can be delivered by the starting motor in cold weather. Starting-motor cables should in all cases be as short as possible, to keep the voltage-drop to a minimum.

If the starting-motor cables are short and well supported, they will in many cases need no additional protection such as conduit or armor. Non-metallic conduit affords excellent protection against chafing or mechanical injury. If metallic conduit is used, great care must be taken to prevent the conduit from coming into contact with the terminals. This same precaution must be observed if an armored starting-motor cable is installed.

A most important point in connection with starting-motor cables is that the battery be well grounded. I believe a good standard method would be to use a two-hole lug that can be bolted firmly to the frame.

It is impossible to exercise too much care in attaching and soldering terminals. Wherever spade-type terminals are used, they should have two sets of wings, one pair to grip the insulation and the other pair to grip the copper core. I am much in favor of soldering terminals whereever possible, thus insuring proper electrical connection. There are some types of terminal and lamp connector that cannot be soldered, but I would like to see the solderless types used only in cases where soldered terminals are not at all feasible.

General

I believe it is possible to arrange the electrical wiring and the various parts of the gasoline-supply equipment so that no gasoline can leak onto any part of the electrical equipment that is liable to emit sparks. If this is done, one of the greatest fire-risks is eliminated.

The automobile as we know it to-day is largely indebted to the electrical industry for its popularity. The electrical equipment gives little or no trouble and its continuity of service depends entirely on the wiring system. The installation of wires is being improved constantly and it is hoped that the entire subject will receive continuing careful attention, resulting in still better electrical service on automotive apparatus.

THE DISCUSSION

N. J. Farrell:—Why is it that the negative side of the battery is grounded on the frame on the majority of cars but that, on one very high-priced car, the positive side is grounded?

W. S. Haggott:—After investigating, we found that about half of the automobile builders ground the negative and the other half ground the positive side. The S.A.E. Standard[3] is that "the storage battery should be grounded on the positive side, and by only one conductor."

E. T. Mathewson:—Why was the single-wire system adopted in preference to the two-wire system?

Mr. Haggott: — It is a matter of convenience and economy. We have found that it is a very easy matter to ground one side of the parts of our equipment. Grounding one side of the motor and generator, head-lamps and the like, reduces the cost and, in some cases, it has been found that a grounded system is more efficient electrically than a two-wire system. It lessens the amount of wire on the car by about 50 per cent.

Mr. Mathewson:—I do not question the efficiency so much as the value of insurance in the one system as preferred to the other. It strikes me that there is much more opportunity for a short-circuit in the single-wire than in the two-wire system. Further, in the places

[3] See S.A.E. Handbook, vol. 1, p. B 29.

where the wires lead to the ground, a bolt, nut, screw or something of the kind is required to make that contact complete; often, these loosen and cause a spark that is likely to be produced very close to the gasoline supply-tank itself. I have wondered whether that has any effect on the rate of insurance.

MR. HAGGOTT:—Assuming a two-wire system instead of a grounded system, the liability to loose connections is just as great in that case as it is in the grounded system, and there is practically the same sparking danger. Further, the two-wire system leads into serious ignition troubles where battery ignition is used, if short-circuits appear on either side of the lighting or starting circuits.

I would like particularly to have opinions expressed in regard to the feasibility of using a standard color-scheme for the wires of the different circuits.

DR. E. K. STRACHAN:—How far will a standard color-scheme be valuable to all the different makes of car? They will all show some variation.

MR. HAGGOTT: — That is difficult to determine; but every car has head-lamps and every car gets its battery feed in a certain way. No doubt there will be some variations, but the idea in general seems feasible. Practically all cars feed the ignition and lighting circuits from the hot side of the starter switch. That particular wire would, therefore, be easily standardized as to color. We can have a standard color for all lighting circuits, another standard color for all generator circuits and another for all ignition circuits. We do not need many colors; perhaps four or five circuits are common to all cars. Some cars do not have intricate systems of wiring. Of course, there are circuits on the high-priced cars that are not used on other cars, but they are not within the scope of what we are trying to do. I am strongly in favor of a standard color-scheme. It would be a very great convenience to the wire manufacturer, and to the automobile builder. Anything we can do to further its adoption will help the garage-men, the repair-men and the car-owners.

XEN FAGAN:—I can see a number of very good reasons why the color system should be adopted and, as a manufacturer, I am entirely in favor of the adoption of standardized colors for the electric wiring so far as this is possible.

MR. MATHEWSON:—Has Mr. Haggott found any standards as to a color scheme among the car builders with whom he has come into contact?

MR. HAGGOTT:—Some of them send out a separate wiring-diagram and specify on it the colors of wire for the various circuits, but no two companies use the same color on the same circuit.

N. S. DIAMOND:—Is the action of ozone physical or chemical?

MR. HAGGOTT:—Ozone is a result of static discharge or leakage. There is always a static effect around high-tension ignition just as is the case around a high-tension transmission-line. The effect of ozone on rubber-covered cables is chemical. The chemical action of ozone, however, results in a physical manner that shows up in the form of cracking. I have done some testing along that line. The best way to indicate the action of ozone quickly is to take a piece of high-tension cable, 2 or 3 ft. long, wind it tightly around a mandrel of $\frac{7}{8}$-in. diameter and connect one end of the cable to the terminals of a magneto. The other end of the cable is connected to one side of a spark-gap. The other side of the spark-gap is grounded and the mandrel itself is grounded. When the cable fails the spark will cease to jump the gap.

Using some pretty good cable in this sort of test, the samples failed within 20 to 25 min. We found a series of fine cracks opening upon the outside of the wire. We do not find these cracks where the rubber is under compression, but on the outside where the rubber is under tension. There are several ways of protecting cable against ozone. The best way I know of at present is to cover the cable with a braid, and then saturate the braid with varnish. You might just as well leave off the braid if it is not thoroughly impregnated with varnish. There has been some development along the line of taking a plain rubber cable and putting varnish on it, but we seem not to be able to find a varnish that will not crack when the cable is bent. If we can find such a varnish, we would not need the braid.

Automotive engineers have problems that are not common to all of the other fields of engineering industry. For instance, the wire and the cable for use in automotive apparatus should not be compared with those for wiring a building, either in the construction of the wire itself or in its installation. For that reason the S.A.E. Wire and Cable Standards have been worked out along individual lines based upon automotive experience of a number of years, as well as in a general way upon what has been done in wire construction in the electrical field at large.

R. W. A. BREWER:—It has been a custom in Europe for a long time to mold the high-tension cables into a tube. It has been thought that this has caused considerable lag in the system. I would like to know whether this is true. It seems to me that the construction is very good mechanically.

MR. HAGGOTT:—I have seen one or two installations of that kind. It is feasible from the standpoint of insulation, so far as individual conductors are concerned but from the standpoint of proper firing has, I believe, caused some trouble. Imagine a six-cylinder engine with the high-tension wires all molded closely together. There is a possibility of an inductive effect being set up in this group of wires, actuating a spark in a cylinder at the wrong time. This is called cross-firing. The same thing may occur with fiber tubing if some means is not provided for carrying the inductive effect to ground.

Processing Spline Shafts by a New Method

By JAMES A. FORD[1]

DETROIT PRODUCTION MEETING PAPER *Illustrated with* PHOTOGRAPHS

THE process devised by the author was evolved to eliminate the difficulties incident to the finishing of the spline and body portions of a spline shaft, such as is used in transmission gearing, by grinding after the shaft has been hardened, and is the result of a series of experiments.

The accuracy of the finished shaft was the primary consideration and three other groups of important considerations are stated, as well as four specific difficulties that were expected to appear upon departure from former practice.

Illustrations are presented to show the tools used, and the method of using them is commented upon step by step. The shaft can be straightened to within 0.005 in. per ft. of being out of parallel with the true axis of the shaft, after the shaft has been hardened, and it is then re-centered true with the spline portion.

THE general practice among manufacturers of transmissions requiring a spline shaft has been, for many years, to finish the spline and body portions of the shaft by grinding after hardening, but it has been found that this process necessitates extreme care in the obtaining and maintaining of the desired form.

[1] Production department, Studebaker Corporation of America, Detroit.

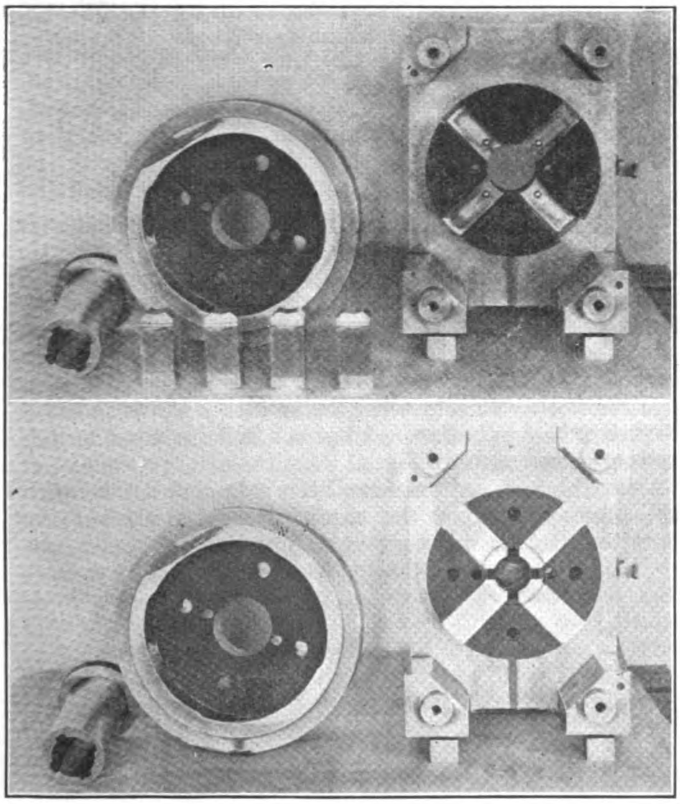


FIG. 1—TWO VIEWS OF THE DIE USED TO PRODUCE THE SPLINE SHAFTS
In the Upper View the Cutters Have Been Removed from the Die. The Cutters Are Shown in Place in the Die in the Lower View. The Cam Ring That Appears at the Left of Both Views Controls the Movement of the Cutters in Practically the Same Way That Chasers Are in a Threading Die

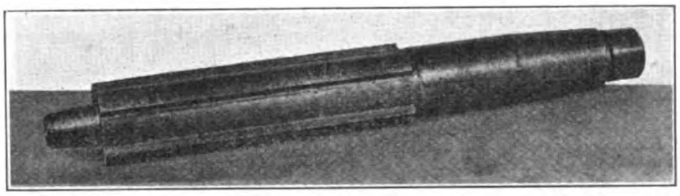


FIG. 2—A TYPICAL SPLINE SHAFT SHOWING HOW THE BODY OF THE SHAFT IS LARGER IN DIAMETER THAN THE PORTION BETWEEN THE SPLINES

Since it was desirable that these difficulties be eliminated, it was decided by the corporation of which I am a representative that the best method would be to omit the grinding operation and substitute a process that would not include these grinding troubles. Experiments along this line consequently were carried through by the methods and standardization department of our corporation, and the results of its investigation are presented briefly as follows.

The accuracy of the finished shaft is the primary thought that was borne in mind during the investigation, and the other important points considered were that the

(1) Splines must be straight, in line with the axis of the shaft, uniform in width, properly spaced and smooth on the wearing portion
(2) Body of the shaft between the splines must be round and parallel with the axis of the shaft, the diameters must be held within prescribed limits at any given portion and the surface must be smooth
(3) Entire shaft must be true in relation to its axis, of the proper degree of hardness and made of the

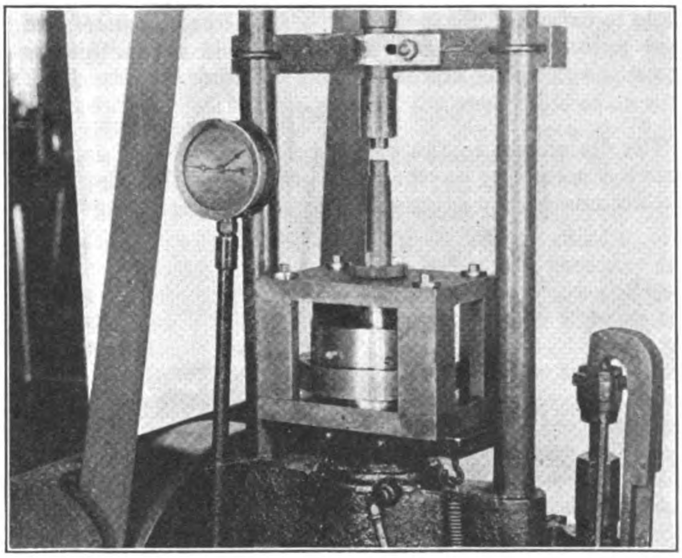


FIG. 3—A SHAFT BEING PRESSED THROUGH THE DIE TO FORM THE SPLINES

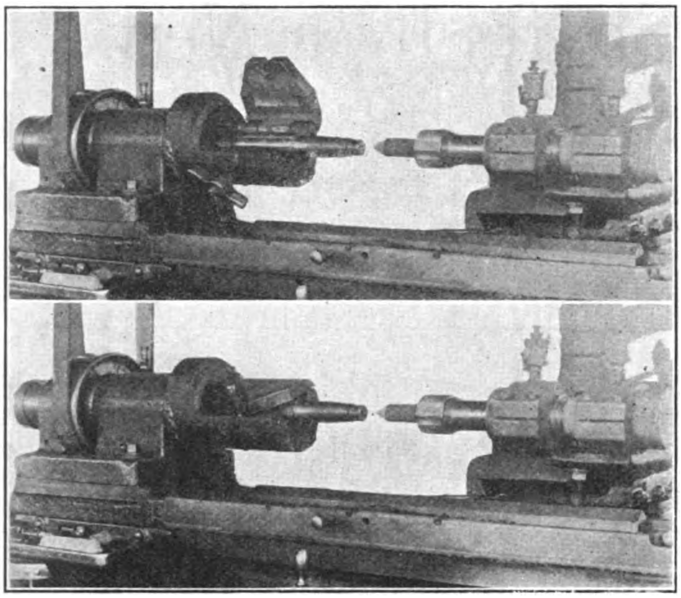


FIG. 4—GRINDING THE CENTERS IN THE SHAFT AFTER THE SPLINES HAVE BEEN FORMED

The Shaft Is Placed in the Fixture Shown in the Upper View and After the Fixture Has Been Closed and Clamped as Illustrated Underneath the Centers Are Ground with a Pencil Wheel. This Method Insures Centers That Are True with the Spline Portion of the Shaft

best obtainable material for the purpose, without regard to any difficulties that this requirement may introduce into machining operations

It was evident that other troubles would appear after departure from the grinding troubles that were experienced in the finishing of the shaft by the old process. The troubles that were anticipated were as follows:

(1) Difficulty in hobbing to exact dimensions, obtaining a smooth, even surface and maintaining a true form
(2) Liability of warpage in shafts during the process of hardening
(3) Variation in diameter due to hardening
(4) Variation in the width of the splines because of hardening

It was believed that a given allowance could be made for shrinkage during the hardening process and that a uniform shape and size could be determined in advance. Therefore, to overcome difficulty (1), the decision was made to draw the shaft through a die after the shaft had been carburized. It was found possible to do this by using the methods and tools that will now be described.

TOOLS AND METHODS

The die shown in the upper portion of Fig. 1 is constructed according to the same principles that apply to the automatic threading-dies now in general usage, the cutters being in the position that the die chasers ordinarily would occupy. In the lower portion of Fig. 1, the cutters are installed in the die. The cam ring is practically of the same construction as that used on a threading die, except that the ring is made stronger. This opening feature is necessary on account of having to pass the shaft back through the die, because the body of the shaft beyond the splines is larger than the body-portion between the splines and the shaft will not pass completely through the die. This is illustrated clearly by the view of the shaft shown in Fig. 2.

The shaft is entered into a bushing that is lined-up with the die. Then the shaft is pressed through the die to a stop that has been set at a sufficient distance to permit the shaft to pass through to the shaft-neck at the end of the splines, as illustrated by Fig. 3. The cutters in the die are then released and the shaft is removed.

During the experiments with this die, it was ascertained that a clearance angle on the cutter of 30 min. was about correct, and it was decided that one pass of the shaft through the die gave the most satisfactory result.

Up to this point we had a shaft that was carburized, and its body and splines were finished to the dimensions desired. After hardening a number of shafts, we were able to determine just what change took place in them, and an allowance was made for this in the adjustment of the die. The change was uniform to a reasonable degree in all shafts, and this enabled us to determine what the standard allowance should be.

WARPAGE

The next problem was that of overcoming the warpage in a shaft after it had been hardened. Fortunately, we found that this warpage was outside of the splined portion of the shaft.

It was very difficult to revolve the shaft on its centers and straighten it to the degree of exactness required, and the operation required too much time. However, we found it possible to straighten the shafts easily to within 0.005 in. per ft. of being out of parallel with the true axis of the shaft. Therefore, we straightened the shafts to within the 0.005-in. per ft. limit.

RE-CENTERING

After having been hardened and straightened to within the 0.005-in. per ft. limit, the shaft is gripped in a fixture mounted on the spindle of an internal-grinding machine as illustrated by the two views shown in Fig. 4. It is then revolved true with the splines while it is being re-centered by grinding against the pencil-shaped grinding-tool shown also in Fig. 4.

After the new centers have been established true with the spline portion of the shaft, the remainder of the operations follow ordinary shop practice.

The Present Status of the Air-Mail Service

By COL. E. H. SHAUGHNESSY[1], U. S. A.

WASHINGTON SECTION PAPER

THE author outlines the history of the Air-Mail Service and states that the recent policy has been to carry out the intent of the Congress, to align the service with the desire of the administration for economy and to discontinue too rapid expansion.

After a description of the routes and divisions and a listing of the present landing-fields and radio stations, the present equipment is outlined and commented upon, tabular and statistical data being presented. The discussion covers the organization and performance of the service, the casualties, the cost of operation and the policy governing future plans.

THE Air-Mail Service operated by the Post Office Department was started in May 1918, the first step being the establishment of a mail route extending from the City of Washington to New York City via Philadelphia, no intermediate stops being made and, at first, Army planes and personnel being used. The planes were of the JN4-H type, equipped with 150-hp. Hispano-Suiza engines, and had a carrying capacity of 150 lb. of letter mail, or approximately 6000 letters. In August 1918 the Post Office Department, having completed a civilian organization, relieved the Army from further duty with the Air-Mail Service, thereupon assuming full charge of all activities and at the same time putting into use the standard E-4 type plane, equipped with the same 150-hp. Hispano-Suiza engine but having a carrying capacity of 200 lb., or approximately 8000 letters.

The Washington-New York route operated steadily and successfully. This warranted further expansion and in May 1919 the first leg of the transcontinental route was established from Cleveland to Chicago, via Bryan, Ohio, using the DH-4 type plane, equipped with the 400-hp. Liberty engine, having a carrying capacity of 350 lb., or approximately 14,000 letters. Further extension followed rapidly. Service was started in July 1919 from Cleveland to New York City via Bellefonte, Pa.; from Chicago to Omaha, via Iowa City, Iowa, in May 1920; and from Omaha to San Francisco, via North Platte, Neb., Cheyenne, Wyo., Salt Lake City, Utah, and Reno, Nev., in the following September. In addition to the transcontinental trunk-route, additional lateral routes were put into operation from St. Louis to Chicago in August 1920 and from Chicago to Minneapolis in December of the same year. When the present administration took office, there were air-mail routes in operation from New York City to San Francisco, St. Louis to Chicago and Minneapolis to Chicago, over which mail planes were flying regularly during the daylight hours.

Great things had been accomplished in the way of extending the Air-Mail Service, but the situation as we found it was not altogether satisfactory. Criticism was being directed at the service by the public and by the Congress, principally due to the fact that there had been a series of unfortunate accidents during 1920 which had resulted in a considerable loss of life. In addition to this feature, members of Congress in committee and on the floor objected to the manner in which the Air-Mail Service had been extended and financed. The whole matter was given very serious consideration. A careful check indicated that most of the operating difficulties came from too rapid expansion without providing necessary facilities with which to operate efficiently; also, there was need for a thorough understanding with the committees in the Congress.

We decided, first, to carry out the intent of the Congress, since it is not unfriendly to an Air-Mail Service but simply, and very properly, unfriendly to what it thinks is maladministration in any service. Second, we determined to align ourselves with the expressed desire of the administration for economy. Third, we planned to put into effect my own thought that we would help aviation much more effectively by stopping the too rapid expansion, which seemed to be making the Air-Mail Service an extra-hazardous undertaking through lack of sufficient facilities, and concentrated our efforts on standardizing and perfecting the operation on a more restricted scale.

For these reasons we decided to discontinue the lateral routes. That from the City of Washington to New York City was abandoned in May 1921 and those from St. Louis to Chicago and from Minneapolis to Chicago were discontinued in June 1921. This reduced our expenditures at the rate of $675,000 per annum. Today, we are operating only the New York City to San Francisco route. It is interesting to note that nevertheless our mail planes will fly at least 1,800,000 miles during the fiscal year beginning July 1, 1921, as compared with 1,760,000 miles flown during the preceding fiscal year, this showing being due to increased efficiency in flying performance.

As operated at present, the Air-Mail Service is used as an auxiliary to the fast mail-train service, because we do not attempt to fly at night. We hope that within a reasonable length of time the proposed Bureau of Aeronautics in the Department of Commerce will come into being and start the work of marking the airways for night flying. When this is done, the real value of an air-mail service will become evident at once, for with night flying mail can be put across the continent in less than 30 hr. As a matter of fact, on a test flight on Feb. 22, 1921, the Air-Mail Service put mail across from San San Francisco to New York City in 25 hr. and 21 min. of actual flying time. This was a notable performance but altogether too hazardous to try again before the way is marked by suitable aids to flying such as lighthouses and beacons.

The transcontinental route is divided into three operating divisions. The Eastern division, from New York City to Chicago, is 770 miles long, with headquarters at the former point. The Central division, from Chicago to Rock Springs, Wyo., is 1125 miles long, with headquarters at Omaha. The Western division, from Rock Springs, Wyo., to San Francisco, is 785 miles long, with headquarters at the Pacific terminal. Air-mail fields are located at Hempstead, N. Y.; Bellefonte, Pa.; Cleveland;

[1] Late second-assistant postmaster-general, Post-Office Department, City of Washington.

Bryan, Ohio; Chicago; at Maywood, Ill. (a checkerboard field); Iowa City, Iowa; Omaha; North Platte, Neb.; Cheyenne, Wyo.; Rawlins, Wyo.; Rock Springs, Wyo.; Salt Lake City; Elko, Nev.; Reno, Nev., and San Francisco. In addition there is an air-mail warehouse at Newark, N. J., and an air-mail repair-depot at Maywood, Ill.

Radio stations are located at the City of Washington headquarters and at all fields except that at Rawlins, Wyo. Navy radio stations are used jointly at Cleveland, Chicago and San Francisco. All the other radio stations are owned and operated by the Post-Office Department.

EQUIPMENT

During the period from May 15, 1918, to Oct. 31, 1921, 3 years and 5½ months, 221 planes have been used in the Air-Mail Service. Table 1 gives their history.

TABLE 1—PLANES USED FROM MAY 15, 1918, TO OCT. 31, 1921

History	Number
Crashed; Parts Salvaged	81
Crashed and Burned; No Salvage	20
Burned on Ground; No Salvage	7
Withdrawn Unsatisfactory Type; Parts Salvaged	19
Transferred to Army	1
Withdrawn on Account of Age; Parts Salvaged	8
Withdrawn and Stored; Small Type	9
Available Oct. 31, 1921, Flying Condition	50
Available Oct. 31, 1921, now Undergoing Repair	26
Total	221

Since June 1921, when the reorganization of the Air-Mail Service took place, great strides have been made in connection with standardizing and improving the flying equipment. From the first a number of different types of plane had been used. These were principally the JN4-H, the Standard E-4, the JL, the twin-engined DH-4 and the single-engined DH-4. We have standardized on the last named, eliminating all other types. This does not mean that the DH-4 is the most suitable plane available, but simply that it is a satisfactory plane for the Air-Mail Service, and that it is good business to standardize on it because the Army has a large surplus of planes of this type, which are transferred to our service, as needed, without expense to us.

In connection with the useful life of the DH-4 planes, based on our experience, Table 2 gives the record of four planes, showing what can be done in this direction. The planes are still in service and in good condition. At present we have 143 planes in the Air-Mail Service; of these 50 are in serviceable condition, 38 are being overhauled and 55 are in storage awaiting overhauling.

TABLE 2—RECORD OF FOUR AIR-MAIL SERVICE PLANES

Plane No.	Date of First Trip	Number of Miles Flown	Number of Hours Flown
71	March 3, 1919	55,416	625
99	Nov. 15, 1919	64,590	699
104	Dec. 18, 1919	58,219	634
175	Oct. 5, 1920	54,079	647

Another interesting item is that during the fiscal year ended June 30, 1921, the Post-Office Department paid $276,109 to companies for rebuilding and converting planes, and purchased eight planes for $200,000. But no planes have been purchased since July 1921 and all conversion work has been done at our own repair-shop at an average cost of $1,200. At the same time our situation has improved materially. We have 14 more planes in serviceable condition today than we had in July 1921.

ORGANIZATION

The Air-Mail Service is directed by a general superintendent located in the City of Washington; he reports to the Second-Assistant Postmaster-General. The field service is divided into three operating divisions under charge of division superintendents located at Hempstead, N. Y., Omaha and San Francisco. There is an assistant superintendent for each division, as well as a local manager at each landing-field. The supply warehouse at Newark, N. J., and the repair depot at Maywood, Ill., are in charge of superintendents who report directly to headquarters. The total number of employes is 479, and the annual payroll is $787,620. Prior to the reorganization on July 1, 1921, there were 521 employes, salaries totaling $864,321.

PERFORMANCE

The operating performance of the Air-Mail Service in recent months has been truly remarkable. During the first quarter of the present fiscal year 98 per cent of all scheduled trips were completed. The record, by divisions, is given in Table 3.

TABLE 3—OPERATING PERFORMANCE OF THE AIR-MAIL SERVICE

Division	Number of Miles Flown	Number of Letters Carried	Scheduled Trips Completed, Per Cent
New York City to Chicago	112,626	3,506,000	98.5
Chicago to Rock Springs	168,805	4,310,760	97.6
Rock Springs to San Francisco	109,587	2,197,520	98.0

In compiling the figures in Table 3, trips were not counted unless completed. In addition to the 391,018 miles flown with mail on regular schedule, our records show 49,662 miles of test flights and ferry trips, making a grand total of 440,680 miles flown in the 3 months. An exceptionally good record was made on that portion of the route between Cleveland and Chicago, an air-distance of 335 miles. No mail trips were defaulted from April 29 to Oct. 31, 1921, a period of 185 days. During that time 98,890 miles was flown and 3,271,360 letters were carried. Our records show that approximately two-thirds of all mail trips are made in clear weather, the remainder being in fog, rain or snow. During the same quarter there were 11 crashes, 8 of which occurred on air-mail fields and 3 at outside points due to forced landings. Two planes were destroyed entirely but during the quarter there were no fatalities or injuries to employes on regular mail trips. There was, however, one fatality on a ferry trip.

CASUALTIES

During the period from May 15, 1918, to Nov. 30, 1921, 3 years and 6½ months, there were 30 fatal accidents in the Air-Mail Service. The record by fiscal years, from July 1 to June 30, is shown in Table 4.

During 1920, using round figures, there was one fatality for each 100,000 miles flown; since July 1, 1921, we have had one fatality for 800,000 miles flown. In my opinion this very much improved showing is due principally to putting into effect a sensible businesslike program, using trustworthy men and material and leaving the experimenting and stunting to those who are not entrusted with the United States mails. I emphasize the

TABLE 4—FATAL ACCIDENTS IN THE AIR-MAIL SERVICE

Year	Number of Months	Number of Pilots	Number of Mechanics	Number of Other Employes
1917	1.5	0	0	0
1918	12.0	2	1	0
1919	12.0	6	1	2
1920	12.0	13	4	0
1921	5.0	1	0	0
Total		22	6	2

fact that since July 1, 1921, during the time this remarkably fine record was made by our pilots, DH-4 planes were used exclusively. I cannot say too much in commendation of our air-mail pilots, several of whom have flown 100,000 miles while carrying mail. Since the beginning of the Service, air-mail pilots have flown 3,400,000 miles and carried more than 2,500,000 lb. of mail, or approximately 100,000,000 letters. No finer class of men or pilots can be found anywhere, individually or collectively, than those in the Air-Mail Service. It is a pleasure to recognize publicly the work they are doing. They are worthily carrying on the best traditions of the Service.

Cost of Operation

Statistics have been published regularly showing the operation and maintenance costs of the Air-Mail Service. They are useful for comparative purposes but have not been truly representative, in my opinion, because the figures have not included such items as general overhead expense, permanent improvements, surplus supplies on hand that were not charged out until used, planes purchased that were not charged to the capital account and the like. Using the statistics as they have been published, the purely operating and maintenance cost per mile during the quarter ended Sept. 30, 1920, was 87 cents; for the same quarter of 1921 it was 71⅓ cents, a reduction of 15⅔ cents per mile. We intend to revise the cost sheet so as to include every dollar properly chargeable to the Air-Mail Service. The new method applied to the periods just mentioned would result in a cost of $1.43 per mile for 1920 and $1.04 per mile for 1921, or a reduction of 39 cents per mile operated.

This material reduction in operating cost is chargeable to (*a*) reduced overhead expense, (*b*) increased efficiency of the personnel and (*c*) economy in purchasing supplies, particularly gasoline. We are operating successfully today on domestic aviation gasoline, whereas formerly the so-called fighting grade of gasoline was used. Cost per mile means nothing unless it is translated into ton-mile figures. Taking the last figures quoted, which are truly representative, we find that in 1921 the DH-4 plane, carrying 350 lb. of mail, was operated at the rate of $8 per ton-mile. This year, the same plane carrying the same load costs at the rate of $6 per ton-mile. We have, however, put into service recently an improved DH-4 plane with the load-carrying capacity increased to 800 lb. This makes possible an increase of over 100 per cent in utility with no additional operating cost; bringing the ton-mile cost down to $2.60. No doubt private enterprise can do much better than this, because, as is well known, the Liberty engine is expensive to operate. Our justification for using it is the surplus stock of the War Department, the first cost of it to us being nothing.

NOTES FROM A PRODUCTION NOTEBOOK

(*Concluded from p.* 412)

were tested in finished cars by inspectors who regularly passed on gear noise. The nature of the test, its object and the condition of the cars were unknown to them. Our belief in the clearance theory was again upheld by their reports. With one or two exceptions they classified the transmissions as unsatisfactory, fair or good, good or excellent, in the order of the three groups, a fact which shows that the small clearance found in some stock-bearings caused noise and that the noise was eliminated when a slightly larger clearance was provided and strictly maintained.

The amount of clearance to be allowed in any case cannot be determined by rule. We have found that it varies in transmissions similar in design but differing in detail dimensions. Our experience leads us to believe that each mounting must be studied individually, by varying the looseness of the bearing until the suitable amount is apparent. In the ball bearings referred to, a total radial clearance of from 0.0006 to 0.0009 in. was found the most satisfactory.

Backlash and Noise

From the two cases cited we judged that the same reasoning might apply in determining the proper allowable backlash in gear-teeth. Here, again, there is a possibility of not providing sufficient room for a film of lubricant, of squeezing all the oil out of the meshing space, and of causing increased noise. Experiments with gears identical in every respect except in the amount of backlash showed that a very evident drop in the noise of the gears was produced when the backlash was increased to a certain point. We now endeavor to hold the backlash of Packard gears within closer limits and work to keep it near to standards, which must be determined for the different designs. Some assemblies will give good results with much more backlash than will others. The high limit is determined invariably by the existence of some rattle; the low limit is determined by the smallest clearance that can be used with oil as thick as that in frequent use in winter weather, without causing growl or bearing noise.

Rear-Axle Bevel-Gears

It has been our experience that whenever you silence one noise in a car, another becomes evident that never caused complaint before. The eventual attainment of silence is the result of a persistent noise-curing campaign which starts with the noise that is most noticeable and works down the line. This method has resulted in finding that the rear-axle noise still persists in most cars though in a much less disagreeable degree.

The spiral bevel-gear, originally introduced by the Packard company, was a big step in the direction of reducing rear-axle gear-noise. Until other units of the automobile had been perfected to their present state of quietness, we were satisfied that rear axles were about as quiet as they could be made commercially. We are now endeavoring to perfect the assembling of the gear so that the noise of operation shall be reduced still more.

If the presentation of these notes results in a valuable discussion of the subjects treated, we shall feel repaid for our efforts in getting the material together.

Stresses and Wear in Automobile Tires[1]

A SURPRISINGLY large proportion of the total energy losses in rubber-tired automobiles has been shown to be due to the rolling-resistance of the wheels and especially of the driving-wheels. How a tire can absorb such relatively large amounts of power has been discussed by Dr. A. Riedler,[2] who concludes that the work done in deforming the tires represents the major part of the tire losses. While Riedler's conception is very plausible, the details of the process by which the work of tire deformation dissipates engine power need further elucidation. It is one of the objects of this paper to present experimental evidence in support of the view that tire losses are due to work expended in tire deformation.

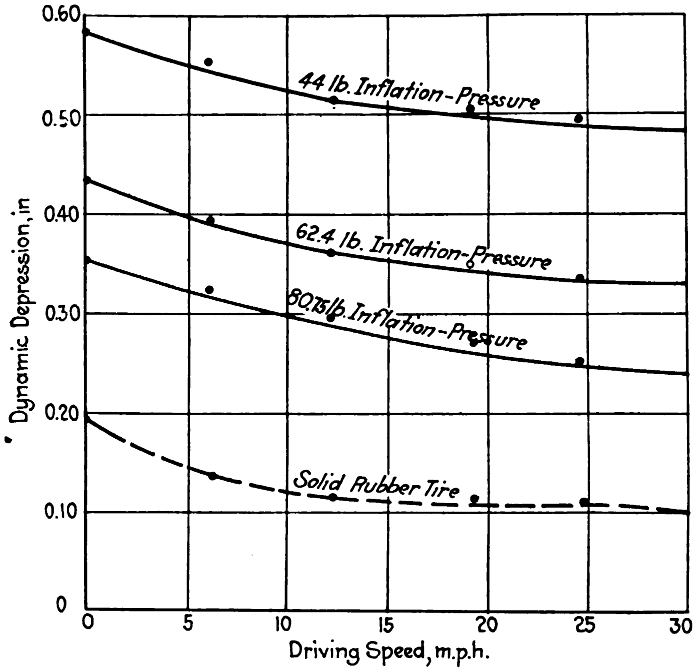


FIG. 1—CHART SHOWING THE VARIATION IN THE DEPRESSION OF TIRES WITH THE DRIVING SPEED AND THE INFLUENCE OF THE INFLATION-PRESSURE ON THE DYNAMIC WHEEL-LIFT

The results of this investigation will be presented in two parts, the first of which relates to conditions on a smooth road, while the second takes into account the effect of obstructions on the road. The experiments discussed in the first part are designed to interpret rolling-resistance in terms of tire deformation as well as to correlate tire stresses and wear. In the second part an attempt is made to extend Riedler's conclusion to the interaction of springs and tires.

STATIC AND DYNAMIC CHARACTERISTICS ON A SMOOTH ROAD

The curve obtained by plotting the deflections or depressions of a rubber-tired wheel at various loads may be called the static characteristic of the tire. A striking fact about such static-characteristic curves is that their trend differs according as the measurements are made at increasing or at decreasing loads. The same peculiarity is shown also by the deflection-load curves of leaf springs where there is friction between the surfaces of the leaves of the spring. In the latter case, the area enclosed between the loop obtained by plotting the deflections corresponding to loads increasing at first up to a maximum and then decreasing back to no load is a direct measure of the work done by the frictional forces between the leaves of the spring. The exactly similar loop shown by the static characteristic of rubber tires suggests at once a similar origin; that is, it may be taken as evidence of frictional sliding between the layers of the rubber tire, and the area of the loop is then a direct measure of the internal frictional forces called into being by tire deformation. In our experiments this internal frictional loss was 5 per cent of the total work of deflection in the case of a solid rubber tire, and from 20 to 30 per cent, according to the inflation, in the case of pneumatic tires.

The deflection or depression determined at varying loads while the wheel is running on a revolving drum was found to be always distinctly smaller than the corresponding static value. Fig. 1 shows how this depression varies with the driving speed in the case of a solid rubber tire and of a pneumatic tire at three inflations, the load remaining constant throughout. These curves show that there is an actual lift of the wheel as the speed increases, this dynamic wheel-lift amounting to 33 per cent of the normal static depression in the case of a pneumatic tire at 80¾ lb. per sq. in. inflation-pressure, and to 53 lb. in the case of the solid rubber tire.

Fig. 1 shows also the influence of the inflation-pressure on the dynamic wheel-lift. A careful comparison of the three upper curves reveals a slight increase of the wheel-lift with the inflation-pressure; for example, the lift for a speed of 19 m.p.h. at 44 lb. per sq. in. pressure is 11 per cent and at 80¾ lb. per sq. in., 14 per cent of the static depression.

This wheel-lift is made more evident by comparing the static characteristic with the dynamic characteristic at some constant speed, as is done in Fig. 2. Here it will be seen that, at a speed of 19 m.p.h., the dynamic wheel-lift is practically independent of the load.

The cause of dynamic wheel-lift may be sought in the action of the inertia forces that resist the sudden accelerations with which the parts of the tire on a revolving wheel are deflected at the moment of contact with the road. The internal frictional forces also are a function of the velocity, for an attempt to slip one layer of tire material suddenly over another will be met by a counter resisting force. Such resisting forces then oppose the sudden deformation of the tire contour, with the result that there is formed an accumulation of tire material immediately in front of the area of contact with the road. This bulge formation, as we shall call it, is compensated for by a decrease in the inertia and friction forces and provides a gradual transition at the point where flattening begins. In effect, this means a smaller deflection of the wheel at a given load.

This view is consistent also with the increase in the wheel-lift with increasing inflation-pressures. The pressure between the layers of tire material increases with the inflation-pressure and, assuming a constant coefficient of friction, the frictional forces will increase as the inflation-pressure increases, and this in turn results in an increased bulge formation and hence a smaller deflection. Load does not affect the lift because the change in the inflation-pressure due to flattening is negligible and, furthermore, the radius of curvature at the point where the tire flattens, a factor that should influence friction, may be regarded as constant in view of the smallness of the depression when compared with the diameter of the wheel.

ROLLING-RESISTANCE

The resultant of all forces acting in a tire represents the rolling-resistance, and can be determined by measuring the rolling losses. This has been done for a solid rubber tire

[1] This paper is a brief adaptation of an article by Otto Enoch published in *Der Motorwagen*, Sept. 10, 20 and 30, 1921. Comment by the Research Department is included, and this is distinguished from the text by enclosing brackets.

[2] See The Scientific Determination of the Merits of Automobiles, p. 314.

and for a pneumatic tire inflated to a pressure of 62.4 lb. per sq. in. The two sets of curves in Fig. 3 show the relation between rolling-resistance, driving speed and wheel load.

An increase in the rolling-resistance with the driving speed might have been predicted as a consequence of the known influence of the velocity on the forces of inertia, but there is another factor contributing to the rate of this increase. This additional loss of power at higher speeds is due to the time lag in the elastic forces that restore the flattened tire to its normal shape; these forces of restitution are damped by friction and by mass-inertia to such an extent that they may be active for some time after the affected part of the tire has left the road, and hence at higher speeds a fraction of the available compression energy is not returned to the tire.

The internal and external frictional forces increase as the flattened area of the tire increases and also as the inflation-pressure increases. The influence of increased surface is shown in Fig. 3 by the greater rolling-resistances at the higher loads, and is confirmed by the two curves taken at a

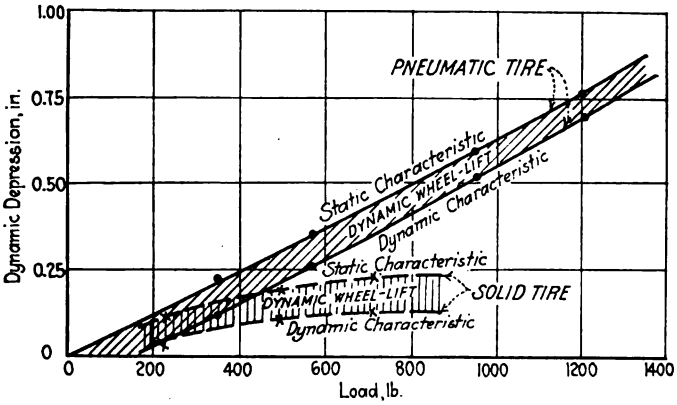


FIG. 2—COMPARISON OF THE STATIC AND DYNAMIC CHARACTERISTICS OF SOLID AND PNEUMATIC TIRES

constant speed of 19 m.p.h., for, since a pneumatic tire flattens much more than a solid tire with increasing loads, the slope of a resistance-load curve should be greater for the pneumatic tire.

A further illustration of the connection between the frictional force and the area of road contact is given in Fig. 4, which shows two sets of curves for the relation between the rolling-resistance and the inflation-pressure. In the set of two curves, the load being constant, there is a smaller area of road contact as the inflation-pressure increases and hence a smaller rolling-resistance, despite the fact that the in-

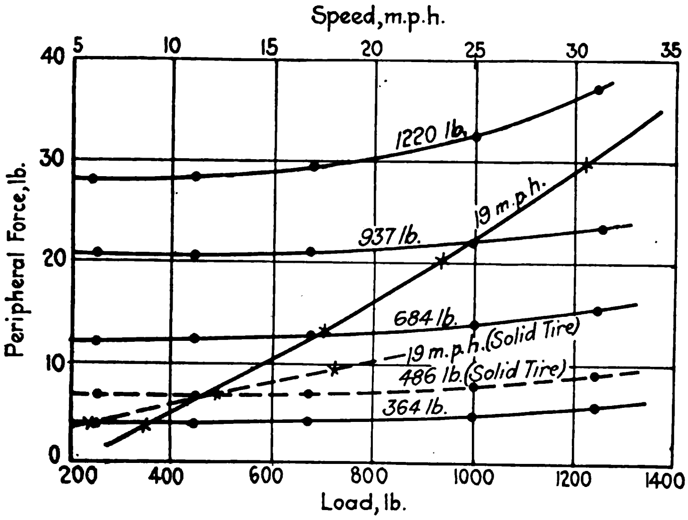


FIG. 3—RELATIONS EXISTING BETWEEN ROLLING-RESISTANCE, DRIVING SPEED AND WHEEL LOAD

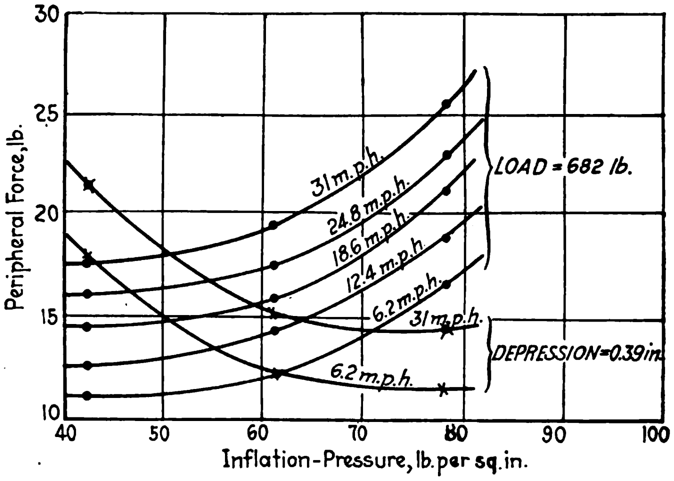


FIG. 4—CHART SHOWING THE RELATION BETWEEN THE ROLLING-RESISTANCE AND THE INFLATION-PRESSURE

creased pressure actually makes the friction somewhat greater. That the internal pressure really has an effect in the sense predicted is proved by the set of five curves in Fig. 4, which were obtained at a constant area of contact; that is, by adjusting the load until the dynamic depression was 0.39 in. for each measurement. In other words, of the two factors determining the magnitude of the frictional work, the area of contact preponderates to such an extent as to mask the pressure effect.

ZONES OF MAXIMUM WEAR

If now the points of the greatest deformation in a tire are the points of the greatest power-absorption and dissipation, one would expect to find the zones of the greatest wear at those points where flattening causes the greatest change in the radius of curvature of a tire section. This would obviously be transversely at the side perpendicular to the driving direction, while longitudinally in the direction of rolling the bending is more gradual.

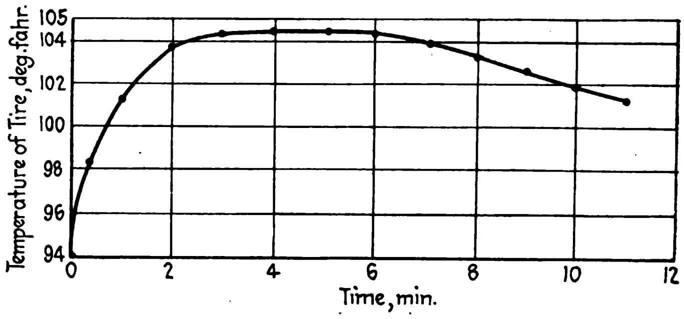


FIG. 5—CURVE SHOWING HOW THE TEMPERATURE OF A TIRE INCREASES AFTER A SUDDEN STOP

Long-run durability tests made on smooth 32½ x 5-in. Calmon tires proved that such is indeed the case. The photographs reproduced in the German original show unmistakably the two lateral zones of maximum wear separated by a zone in which there is a certain amount of accumulated rubber material. Furthermore, the lateral zones are widest in tires of least inflation, as is to be expected according to the view here advanced.

Anti-skid tires, recently appearing on the market in Europe, are very favorably constructed as regards resistance to wear. The ribs on such anti-skid tires present a reinforced surface at the two lateral zones of maximum wear and are, nevertheless, advantageously distributed for the bulge formation. In addition, the ribs should aid materially in the ventilation necessary to keep the temperature of the tires as low as possible.

Wear on the bearing surface of tires is caused also dur-

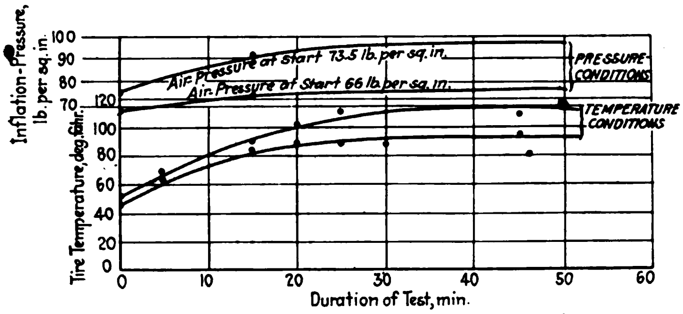


FIG. 6—RELATIONS EXISTING BETWEEN INFLATION-PRESSURE, TIRE TEMPERATURE AND DURATION OF TEST

ing the distortions induced by large variations in torque and by braking.

HEATING OF TIRES

The frictional work in tires is directly converted into heat, and since the quantity of heat thus generated represents from 10 to 20 per cent of the total engine-power, it is not inconsiderable. If the wheel were suddenly stopped, a large part of this heat would go to raise the temperature of the tire, as is shown in Fig. 5.

Under driving conditions, a thermal equilibrium is reached at which the temperature of the tire has risen to such a degree that the gradient toward the surroundings is sufficient to carry off the surplus heat as fast as it is generated. The higher temperature expands the air in the tire, causing an increase in the inflation-pressure which may burst the tire, for instance on sudden stopping after a fast run. Besides this danger of bursting, the life of the tire is very adversely affected at higher temperatures. Fig. 6 reproduces some of our data on the rise in tire temperature and pressure during driving.

[Details of the experimental arrangement by which these figures were obtained are not given in the text, so that it is

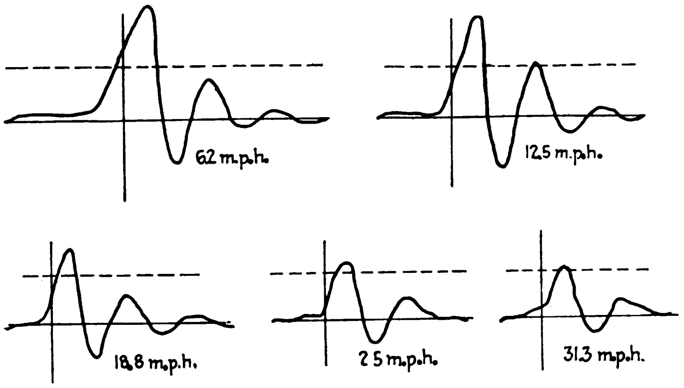


FIG. 7—DIAGRAM SHOWING THE OSCILLATIONS FOLLOWING THE PASSING OVER AN OBSTRUCTION AT VARIOUS SPEEDS

impossible to compare Enoch's results on tire temperatures with those of Lockwood,[3] and of Ellenwood.[4]]

With inferior reclaimed rubber tires we recorded a temperature rise of over 50 deg. fahr.; that is, from 66 to 116 deg. fahr.

The necessity for considering the thermal relations in tire construction is made evident by these results. As already indicated, the ribs of anti-skid tires aid in the reduction of the tire temperature, by the larger radiating surface presented as well as by the additional air-circulation that they induce. Another proposal for a better heat-dissipation is the use of rim material having good thermal conductivity, for example aluminum. Metal buttons sunk into the tire might serve the purpose were it not for possible deleterious effects on the mechanical properties of the tire.

IMPACT STRESSES

In driving over obstructions, tire deformation is of a different nature from that previously described. To get an insight into the effect of a rough road on rolling-resistance, we have recorded the oscillations resulting from vertical impact and calculated approximately the corresponding work of deformation.

Impacts on the wheel were produced by a strip fitted on the revolving drum of the test-stand; the strip was adjusted to strike the wheel once and then drop off. Diagrams of the vertical oscillations of the wheel axle were obtained by using a specially designed device for timing and recording the vibrations. For pneumatic tires, we used a 16-40 hp. Benz chassis and impact strips of 1-in. height, the 32 x 4½-in. tires being loaded to 682 lb. For solid tires, we used a 10-21 hp. Horsch car and a 0.4-in. height of impact strip.

Figs. 7, 8 and 9 and Tables 1 and 2 reproduce some of our data. The oscillation diagrams will be considered as consisting of four phases as follows:

TABLE 1—DIMENSIONS OF OSCILLATION CURVES FOR PNEUMATIC TIRES

Phase 1	Depth of Depression, in.	0.985	1.180	1.230	1.260	1.280
	Length of Period, in.	1.480	0.740	0.490	0.370	0.290
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300
Phase 2	Height of Throw, in.	0.840	0.760	0.550	0.430	0.380
	Length of Period, in.	0.240	0.490	0.550	0.650	0.700
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300
Phase 3	Depth of Depression on Return Throw, in.	0.340	0.350	0.246	0.196	0.145
	Length of Period, in.	0.390	0.630	0.590	0.630	0.590
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300
Phase 4	Height of Second Throw, in.	0.300	0.350	0.220	0.158	0.118
	Length of Period, in.	0.860	0.860	0.860	0.900	0.940
	Car Speed, m.p.h.	6.200	12.500	18.800	25.000	31.300

The first phase starts at the point where the wheel begins to rise from its normal level as determined by the dynamic depression corresponding to the particular driving-speed. In the oscillation diagrams, the horizontal dashed line is the zero from which these dynamic depressions, represented by the horizontal full lines, are measured. The first phase ends at the point where the oscillation curve crosses the vertical line drawn to indicate the middle of the impact strip.

In Tables 1 and 2 the first column is obtained by subtracting the vertical intercepts from the sum of the dynamic depression and height of impact strip; this difference is the amount by which the tire is indented at the moment of rolling over the impact strip. The second column gives the horizontal distance between the initial point of the curve and the impact strip. The second phase includes the oscillation from the middle of the impact strip to the point where the wheel again reaches its normal position for a smooth road. The third and fourth columns of the tables give respectively the vertical throw and the horizontal distance traveled during the second phase, the former measured from the point of contact of the wheel with the middle of the impact strip to the highest point of oscillation curve, and the latter corrected for the width of the impact strip. The third and fourth phases are represented by the two succeeding loops of the curve. The small oscillations after the fourth phase have been found to have no influence on the work of deformation.

TABLE 2—OSCILLATION CURVES FOR SOLID TIRES

Phase 1	Depth of Depression, in.	0.287	0.390	0.430	0.470	0.490
	Length of Period, in.	0.900	0.450	0.290	0.230	0.190
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200
Phase 2	Height of Throw, in.	0.630	0.770	0.820	0.630	0.630
	Length of Period, in.	0.250	0.730	0.860	0.720	0.840
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200
Phase 3	Depth of Depression on Return Throw, in.	0.180	0.320	0.310	0.290	0.290
	Length of Period, in.	0.270	0.350	0.310	0.350	0.350
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200
Phase 4	Height of Second Throw, in.	0.150	0.200	0.310	0.190	0.220
	Length of Period, in.	0.470	0.550	0.630	0.510	0.550
	Car Speed, m.p.h.	6.250	12.500	18.700	25.000	31.200

[3] See TRANSACTIONS, vol. 12, part 1, p. 380.

[4] See THE JOURNAL, August, 1922, p. 129.

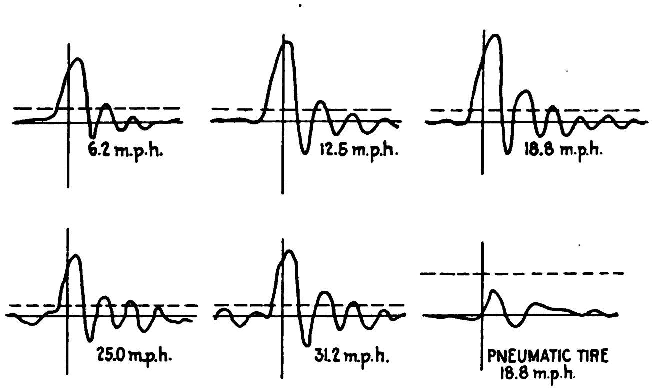


Fig. 8—Another Oscillation Diagram

An examination of Figs. 7 and 8 shows that as the driving speed increases, both the depth of original depression and the amount of throw increase. Furthermore, the horizontal distances traveled during the half-oscillations below the normal position are smaller than the distance traveled during the half-oscillations above the normal position; this is particularly noticeable in the third and fourth phases. This inequality in the vibration periods is caused by the higher equilibrium position of the axle as it is raised by the upward swing of the frame of the car; the frame begins its upward swing at the end of the second phase and, because of the slower oscillation of the heavy frame, continues over phases three and four of the wheel oscillation.

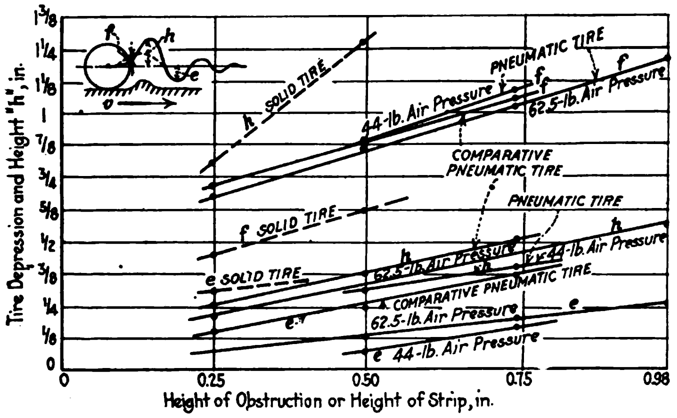


Fig. 9—Chart Showing the Relation between the Height of the Obstruction or the Impact Strip and the Resulting Tire Depression

A comparison of pneumatic and solid rubber tires as regards behavior during impact reveals the much larger amplitude of oscillation in the case of the solid tire. In the diagrams this is shown by a greater height above the horizontal dashed line for the first throw, by a greater depth below the horizontal full line of the return throw, and by the number of half-oscillations that rise above the dotted zero-line. To illustrate these differences, Fig. 8 includes for comparison an oscillation curve obtained after substituting pneumatic tires for the solid tires on the Horsch car. In Fig. 9 this larger throw is shown by the fact that the h-curves for the throw are below the f-curves for the depression in the case of pneumatic tires, but above the f-curves for solid tires. In Fig. 9 the height of the impact strip is plotted against the throw and the wheel depression for two inflation-pressures of pneumatic tires on the Benz chassis and for solid tires together with a comparison set of pneumatic tires on the Horsch car. Lines marked h refer to upward throw, e to return throw, and f to depth of first impact-depression.

Rolling-Resistance During Impact

The work of tire deformation during impact may be calculated approximately from the diagrams of the previous section, in connection with the relation between rolling-resistance at various loads and dynamic depression as measured in connection with the work described in the first part.

We have for any depression of e feet at a speed of v ft. per sec.,

$$d\,W = 550\,L\,dt$$

where

dW = the work done, in foot-pounds and
L = the horsepower per second lost in producing dynamic depression at any instant

If for dt we substitute $\frac{dx}{v}$, where x is the horizontal distance in feet on our oscillation diagram, we have

$$W = \frac{550}{v}\int_{x_1}^{x_1} L\,dx \qquad (1)$$

To evaluate L, we have plotted the results given in the first part of the paper for the power losses L, at various dynamic depressions e. The curves thus obtained approximate very closely to straight lines and therefore we may write

$$L = ce \pm a$$

c and a being constants. Equation (1) then becomes

$$W = \frac{550c}{v}\int_{x_1}^{x_2} e\,dx \pm \frac{550a}{v}\,(x_2 - x_1) \qquad (2)$$

The definite integral $\int_{x_1}^{x_2} e\,dx$ is the area included between the curve and the horizontal axis, and may be evaluated planimetrically for each one of the four phases of the oscillation diagrams. It happens that the straight lines in the L e diagram for pneumatic tires go through the origin and hence the second term of equation (2) has a value for solid rubber tires only. Table 3 gives the values calculated for c and a, where c is the horsepower per inch of dynamic depression and a is the horsepower; and Fig. 10 shows the

TABLE 3—VALUES OF CONSTANTS c AND a

Car Speed, M.P.H.	Type of Tire		
	Pneumatic, c	Solid Rubber	
	Inflation-Pressure, 62.5 Lb. per Sq. In.	c	a
6.25	0.556	0.739	0.00
12.50	1.280	1.817	0.00
18.70	2.145	2.812	0.06
25.00	3.107	4.082	0.15
31.20	4.224	5.384	0.31

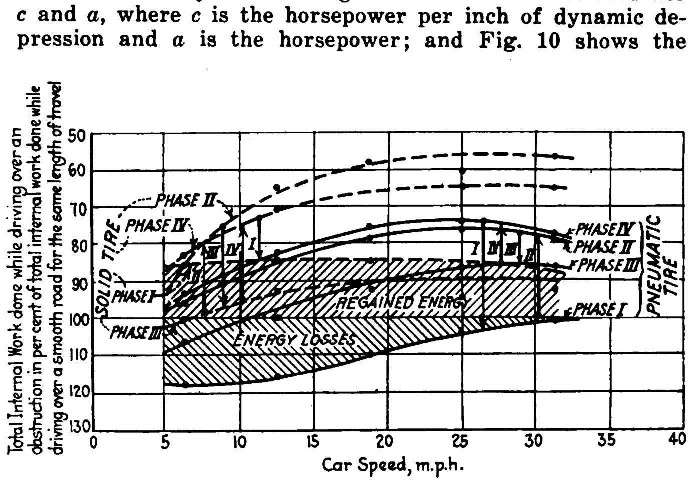


Fig. 10—Chart Showing the Dissipation of Energy in the Tire when Driving Over an Obstruction

values of W in equation (2) for equal horizontal distances of travel.

It will be noticed that the first and second phases practically determine the rolling-resistance, and that the third and fourth phases so nearly counterbalance as to be equivalent to a correction-term only. The second phase may therefore be said to yield a positive gain in elastic work which nearly equals or exceeds the large loss in power occasioned by the high surface-pressure suffered by the tire in the first phase of the impact.

The remarkable conclusion of this calculation is, then, that the work of deformation during impact is not markedly different from the rolling loss on a smooth road, and that indeed the losses on a rough road may actually be less than on a smooth road in the case of stiffer tires, pneumatic with high inflation-pressures or solid rubber, at high driving-speeds.

[We believe that this conclusion cannot be accepted because the principle underlying the calculations on which the conclusion is based is open to grave objections. In the first place, it is not obvious that the power loss at a given wheel deflection during an impact oscillation is the same as the power loss determined at the same depression when running on a smooth road; in fact, this assumption involves the very point that it is desired to test. Furthermore, according to Enoch's contention in the first part of the paper, the friction between the layers of the tire increases with the velocity of depression and thus the power lost on the downward loops of the oscillation curve is more than Enoch calculates, and the power gained on the upward loops, is less.

Even less justifiable is the assumption that when the wheel leaves the road, there is a negative depression of the same type and with the same horsepower factor c as when the wheel is running under load with a consequent positive depression; that is, some of the power-gain obtained planimetrically is certainly fictitious.

Finally, the calculation does not consider several features characteristic of impact, such as the horizontal component of the impact and the spin of the free wheel at the top of its swing. These ultimately result in a power loss, as is shown later in the paper.]

This last conclusion agrees with driving experience according to which no change in the throttle-opening is required to maintain a definite high speed in going from a smooth stretch of road to one of greater unevenness.

[The reference to driving experience is unfortunate at least, because, for the most part, statements such as this are based on very inadequate evidence.]

Interaction of Tires and Springs

Among the factors contributing to tire wear that have not been considered in the above discussion is the frictional work lost in the tire while it is performing the function of a spring. For, during the impact oscillations there is continuous reciprocal interchange of kinetic energy between the springs and the tires; the tire dissipates energy in frictional damping forces just as a spring does.

To estimate roughly the relative losses incurred during the damping of the vertical vibration of spring and tire we took one set of observations on the test-stand, at 18.2 m.p.h. with springs having a constant b, 0.018 in. per lb. and a damping factor k of 0.130 in. per lb. as determined by Bobeth[5] and with a tire having a spring constant, b, of 0.0031 in. per lb. and a damping factor, k, of 0.0350 in. per lb. The damping factor k, like the spring constant b, is defined according to the Continental practice as the deflection for a unit load, and hence the quotient of k/b gives the damping force opposing the force of the spring. Our determinations showed that, for the range of loads used, this damping factor may be taken as a constant for the tire. This force multiplied by the distance over which it acts, the spring deflection y in feet, gives the amount of work in foot-pounds lost by damping; that is,

$$(k/b)\, y \qquad (1)$$

By the same process we obtain the total work done by a spring during a deflection of y feet from an original static depression d as

$$[(d + \tfrac{1}{2}y) \div b]\, y \qquad (2)$$

because, since the force varies from d/b to $(d+y)/b$ during the deflection, the average force is $\tfrac{1}{2}[d/b + (d+y)/b]$, and the work done equals force times deflection. The percentage loss in work is therefore 100 times the work lost by damping, according to equation (1), divided by the total work done by spring deflection, according to equation (2)

$$[k/(d + \tfrac{1}{2}y)] \times 100 \qquad (3)$$

Substitution in equations (1), (2) and (3) of the values for b, d, k and y, as obtained in our experiment shows that, despite the smaller deflections of the wheel, the tire does more work in spring-action than the spring itself, but that the damping loss, $0.0350/0.0031 = 11$ lb. per in. of deflection of the tire, is somewhat less than for the spring, $0.1300/0.0108 = 12$ lb. per in. of deflection.

[This deduction must be accepted with reservation, for data on which it is based are not given, nor is the experiment further described. From the context it seems that, after determining b, d, and k for the wheel and the spring, simultaneous oscillation curves for both the wheel and the car body were obtained at five speeds of the impact drum, and that these curves were used to get the y values for the spring and the wheel. The substitution of these y values, probably the maximum upward throw shown by the oscillation curves, in equation (2) gave columns Nos. 2 and 5 of Table 4; in equation (1), columns Nos. 3 and 6; in equation (3), columns Nos. 4 and 7.]

Table 4 gives the results of our experiment on the relative damping-losses in tire and springs.

TABLE 4—CALCULATED WORK DONE BY SPRING AND TIRE DEFLECTION

	Driving Speed, m.p.h.	6.20	12.40	18.60	24.80	31.00
Springs	Work, ft-lb.	40.60	42.90	32.80	24.40	22.20
	Loss, ft-lb.	2.46	2.46	1.82	1.38	1.23
	Loss, per cent	6.00	6.00	6.00	6.00	6.00
Tires	Work, ft-lb.	41.90	52.50	45.40	24.80	30.10
	Loss, ft-lb.	1.31	2.03	1.60	1.30	0.94
	Loss, per cent	4.30	3.90	3.50	3.30	3.10
Sum of Deflections, in.	Springs	1.06	1.10	0.83	0.63	0.55
	Tires	0.87	0.95	0.75	0.63	0.47

The wear in the tire resulting from the friction represented in the damping vibration is of course an important amount to be added to the wear from the impact losses as given in the previous section of this paper.

Our investigations on the impast stresses were performed on automobiles in which provision was made for vertical spring-action only.

[In the German original there is a consideration of resonance effects in the oscillations. Calculation showed that the range of critical speeds was far removed from the possible velocities of the author's experiments.]

Since the impact stresses were found to be dependent almost wholly on the first two phases of the impact, it is obvious that impact stresses in tires, and consequently tire wear, could be diminished by moderating the first two phases of an impact. Theoretically, this could be accomplished by providing springs for the horizontal components of the impact, and our measurements on a car, fitted with an Arop pendulum spring, confirm this theoretical deduction.

[The author treats spring action at length, but his treatment is omitted here as being only remotely related to tire stresses and wear.]

Figs. 11 and 12 illustrate the favorable action of an Arop horizontal spring.

The import of this improvement in the time and deflection relations of the wheel, as regards the durability of pneumatic tires, may be seen from the following considerations. First, there is the smaller acceleration involved in the inertia forces. The horizontal spring allows the axle to deflect backward, and this is equivalent to a corresponding

[5] See Power Losses and Spring Action in Automobiles, by E. Bobeth, in German.

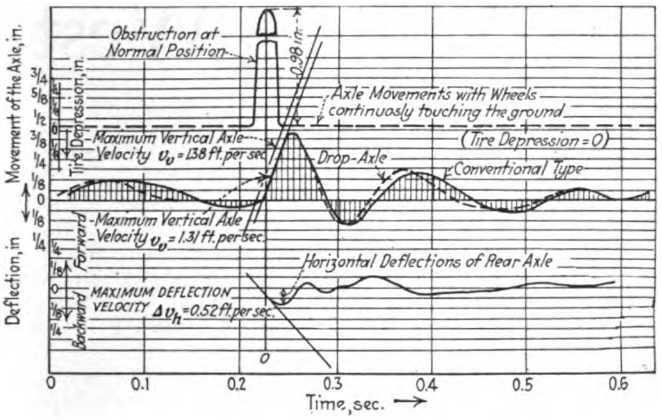


Fig. 11—Comparison of the Oscillation of a Conventional and an Arop Rear-Axle in Passing over an Obstruction 0.98 In. High at a Speed of 18.7 M.P.H.

lengthening of the period of impact. This softening of the impact blow is equivalent to the action of a tire at a low inflation-pressure, without, however, the disadvantages of the low-pressure tire. Incidentally, but by no means unimportant, is the decrease in the tire distortion resulting from this freedom of the wheels in a horizontal direction.

Second, horizontal springs act as as an elastic coupling between the engine and the road, which prevents the distortions in the tire that would otherwise result from changes in torque, sudden throttling or braking. For example, the sudden release of the tractive resistance when a driving-wheel bounces off the road, speeds up the wheel so that when it returns to the road again there is excessive wear for the short time required for the wheel to resume its normal speed. The following calculation will give some idea of the energy involved in tire distortion of this nature: the driving torque of a 30-hp. passenger-car with 31.5-in. tire-diameter at 31 m.p.h. is about 433 ft-lb. This whole torque is used to accelerate the wheel while it is off the ground. If the weight of the wheel is taken as 66 lb., we have for the polar moment of inertia about the axle

$$m\,r^2 = J$$

$$J = (66/32) \times [31.5/(2 \times 12)]^2 = 3.56$$

and the angular acceleraton imparted to the wheels by a torque of 433 ft-lb. would be

$$dw/dt = 433/3.56 = 122 \text{ radians per sec. per sec.}$$

If, now, we assume an average duration of bounce of 0.05 sec. during which acceleration is to be constant, we have as the increment in the angular velocity

$$\Delta\, w = 122 \times 0.05 = 6.1 \text{ radians per sec.}$$

Since a speed of 31 m.p.h. corresponds to an angular velocity of 34.6 radians per sec., we have as the increase in kinetic energy

$$\Delta\, E = J\, w\, \Delta\, w = 3.56 \times 34.6 \times 6.1 = 750 \text{ ft-lb.}$$

This energy that would ordinarily be transformed into tire distortion and thus into wear, is readily absorbed by a horizontal spring.

In Fig. 12 there are plotted the horizontal deflections h, the height of throw w and the maximum vertical velocity V_v. More detailed analysis of the experimental figures obtained with the Arop spring has led us to the design of other spring-systems which are even more efficient, and it may be emphasized that along this line there is great opportunity for improving the spring-suspension of automobiles and for diminishing the wear on tires.

Summary

The wear of an automobile tire is caused by deformations in the inner layers of the tire and by sliding on the road. These two sources of friction develop heat which increases the temperature of the tire. A decrease in the number of layers in building up the tire and a decrease in the coefficient of friction would therefore increase the durability of the tire.

Wear is concentrated in two lateral zones of the tire. Properly shaped ribs are advantageous in reinforcing a tire along these two zones of wear and, in addition, provide increased ventilation for reducing the temperature of the tire. Rims and wheels made of material possessing high conductivity for heat are also advantageous in the latter respect.

Tires are subject also to impact stresses on rough roads. The work of deformation due to these impact stresses is approximately the same as on a smooth road, but decreases with the inflation-pressure. The first two phases of impact determine the deformation work.

Impact stresses can be reduced by decreasing the inertia-mass; that is, by decreasing the weight of wheels, axles and other unsprung parts; also by decreasing the impact accelerations. In this connection, tires with lower inflation-pressure combine the advantage of lessening the impact with the disadvantage of greater deformation work. Horizontal springs possess both an effective impact-damping and a decelerating action of the inertia-masses.

Horizontal spring systems, in addition to other desirable qualities, contribute to an increased durability of the tire.

American and European Practice

[In reference to cord and fabric tires here and abroad, the original series of articles published in *Der Motorwagen*[6] of which the foregoing is a critical digest, contained a statement, which has appeared in other European journals also, to the effect that cord tires used in Germany show a consistently lower mileage and greater power loss than fabric tires.

This statement was so much at variance with both laboratory and service experience in this Country that a letter was written to the author of the articles, Dr. Otto Enoch, asking for further particulars. A translation of our letter appears in a later issue of *Der Motorwagen*,[7] together with a translation of the paper by A. B. Browne and Professor Lockwood entitled Practical Testing of Motor Vehicles[8] and a statement that the letter would be answered later by Professor Enoch.

The very wide discrepancy between American and European experience is so interesting that we should like to hear from any of our members who can throw further light on the subject.]

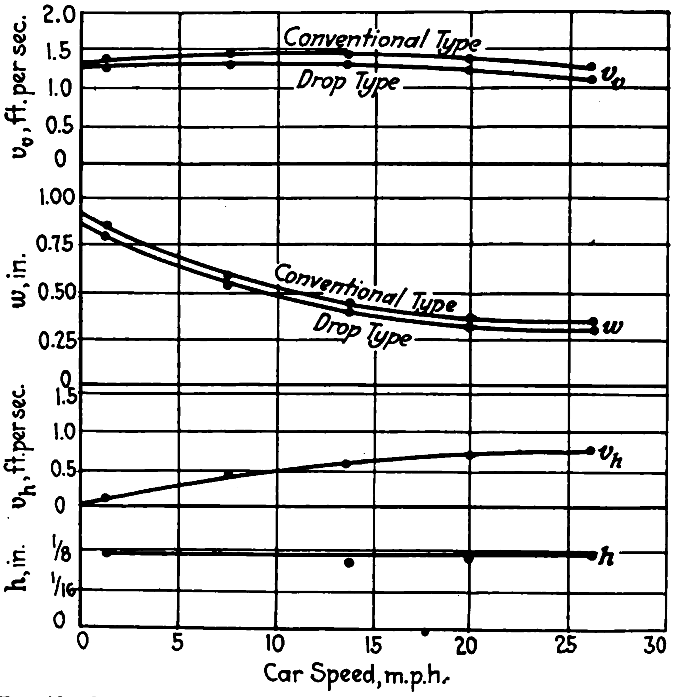


Fig. 12—Characteristics of the Arop Rear-Axle Spring-Suspension in Passing over an Obstruction 0.98 In. High

[6] See *Der Motorwagen*, Sept. 10, 20 and 30, 1921.
[7] See *Der Motorwagen*, Aug. 10, 1922, p. 413.
[8] See Transactions, vol. 10, part 1, p. 68.

Metropolitan Section's Visit to West Point

OF all the summer outings held by the Metropolitan Section, the one of Sept. 16 last will undoubtedly stand high in the memory of those who were fortunate enough to be there. The lazy trip up and down the Hudson River was restful and gave plenty of opportunity to make new friends and talk business or pleasure at the whim of the moment. But the crowning feature of the trip was the visit to the United States Military Academy.

We arrived at the West Point dock at 2:15, almost an hour behind schedule, and were informed that by making a quick trip up the hill we could see the end of the daily inspection. So the crowd immediately boarded the various forms of conveyance, buses, taxicabs, etc., and gained the top in ample time to see the corps of cadets, 1300 strong, drawn up for inspection. In their gray full-dress uniforms, with white crossed belts and high hats, the cadets made a wonderful picture against the green parade-ground, with its background of trees and distant mountains. The Academy Band, in its black and white full-dress, played during the inspection and as the cadets marched off to their barracks.

Immediately afterward we were divided up into sections, and an officer was on hand to act as guide for each section. The sections were headed by Major Charles G. Mettler, professor of ordnance and gunnery, Majors Oscar J. Gatchell and Robert N. Bodine and Capt. Rudolph F. Whitelegg. All these officers belong to the Ordnance Department and are detailed to the Department of Ordnance and Gunnery at the Academy. Under their able and sympathetic guidance the various sections were taken to see where the cadets ate, slept, lived and played.

At the cadets' gymnasium, the fencing, wrestling and boxing rooms, the large gymnasium for floor drill, the swimming pool, the indoor rifle and pistol range were all exhibited and explained. Here, under the direction of the Master of the Sword, Lieut.-Col. H. J. Koehler, the new cadets with bodies partly developed and poorly carried are given an intensive course of physical training. The exercises round out their bodies, develop a correct posture and improve their voices for command, so that within 6 weeks they appear on parade marching erect and steady like the older cadets. The training and athletics are intended not only to develop the individual but also to help him in his future work of training soldiers of the American Army. Each cadet is taught to fence with foil and sabre and to box, wrestle and swim. Outside, he is taught golf, tennis, lacrosse, soccer, baseball, and football.

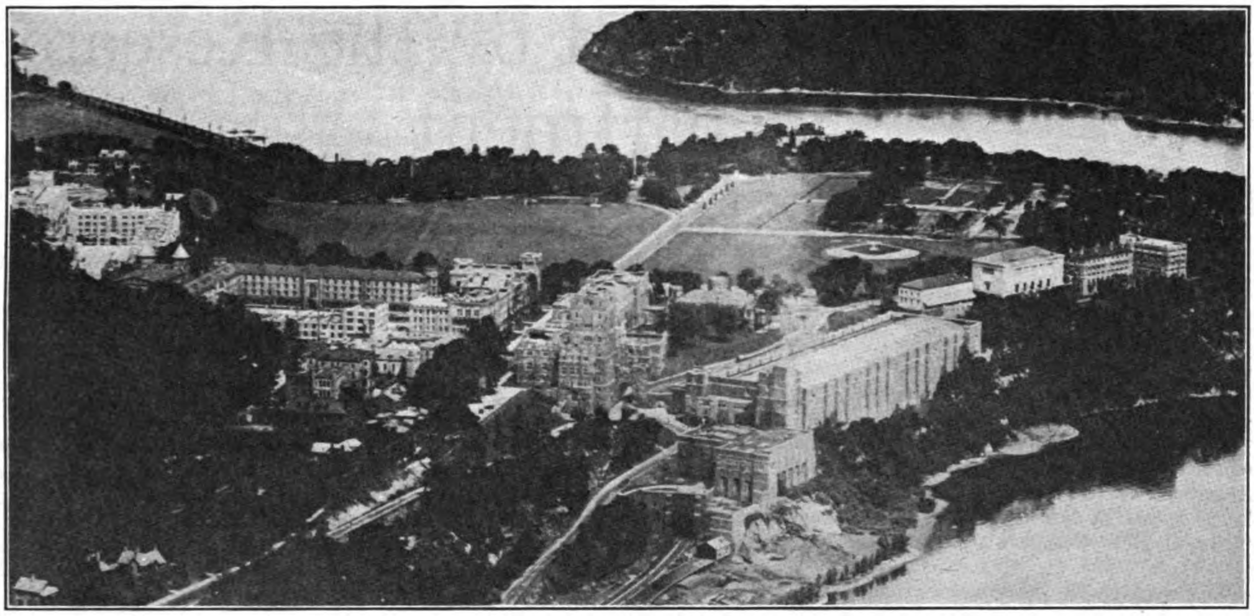
One of the rooms where the cadets sleep and study was next visited. The room that we saw in the North Barracks was used by three cadets, although built only for two. But on account of the congested condition of the barracks, and the failure of Congress to appropriate funds for extension, the room was fitted with a double-deck cot where two men slept, with the third using a single cot. Our attention was called to the arrangement of books and clothing, as this was provided for by the rules and was alike in every cadet's room.

Each of the 12 companies in the corps has a room set aside for the tactical commander, who supervises the discipline and supplies of the organization. In this room, one of which we inspected, the officer in charge is accessible to cadets for advice and suggestion, as well as for the carrying out of discipline.

The next stop was at the beautiful Memorial Hall, the design of Stanford White and the gift of General Cullum. Here are recorded on bronze tablets the deeds of graduates who have been killed on various battle-fields. Set into the walls are many trophies of old wars, such as cannons and arms. The second floor of the Memorial Hall is used for cadet hops, lectures and dramatic performances, although it is entirely inadequate in size. When dances are given, for example, the room is only large enough to accommodate about one-third of the corps.

The balcony of the Memorial Hall, 130 ft. above the river, presents a wonderful view of the great water-gap of the Hudson, which lies between Storm King and Breakneck mountains. On the day of our visit the blue sky was dotted with white clouds so that the mountains were outlined sharply. Underneath was the Hudson, deep blue as the shadows fell across it in the late afternoon. It was easy to

MEMBERS AND GUESTS OF THE METROPOLITAN SECTION AT WEST POINT, SEPT. 16, 1922, ON THE ANNUAL OUTING OF THE SECTION



VIEW OF THE UNITED STATES MILITARY ACADEMY AT WEST POINT FROM AN AIRPLANE

believe the statement of our officer guide, that here is one the most beautiful scenes in America.

The view also indicated why in Revolutionary days West Point became the stronghold of the Hudson. At first the east point of the river on Constitution Island was fortified, but when General Washington, accompanied by Kosciusko, Lafayette, Von Steuben and others of his immortal staff, surveyed the situation, they immediately directed the fortification of the west point of the river, and the name has since been retained. On the hilltops above the academy lies Fort Putnam and other redoubts. These were pointed out and their use during the Revolutionary War explained to us. Also exhibited were a few links of the old chain that was stretched across the river to prevent the passage of the British Fleet during the Revolutionary War.

There is not space here to tell about all the things we saw, but mention should be made of the gilded monument presented to the Academy by a sister school, the Ecole Polytechnique of France, in commemoration of the entrance of the United States into the World War. Beyond Trophy Point, a short promontory used to display cannons and relics of old wars, we visited the old Ordnance Laboratory, built in the early '30's of the last century as a central fort to protect ammunition and supplies.

At the Laboratory were gathered the different types of motor-transport vehicle used on the reservation. These included machine-shop and supply trucks, ordnance tractors, staff cars and motorcycles. Motor-transport officers at the Point were on hand to exhibit the vehicles and explain their operation to the members of our party.

A few brief speeches were made immediately after the dinner. The Outing Committee, A. C. Bergmann, introduced Chairman Kemp of the Metropolitan Section, who in turn called upon President B. B. Bachman of the Society, who was a guest of the Section for the trip. Mr. Bachman complimented the officers on the way in which the visit had been handled and for the pleasure experienced by all the members of the party. In a few graceful words of acknowledgment, Major Mettler expressed the hope that we would all come again, and also spoke of the need for larger facilities at the Point, in view of the reduction of our Military forces that made more than ever necessary the continuous training of young men who could be competent to turn citizens into soldiers in case of a National emergency.

Those who made the trip with the Metropolitan Section now appreciate, more than ever before, the important part that West Point has played in our National life. Since its establishment, about 6900 cadets have been graduated. These men have fought in all the wars of the United States, have been successful in many prominent positions in civil and political life, and have occupied the Presidential Chair with credit.

The present corps of nearly 1300 cadets comes from all parts of the Country, since each State and every Congressional District have a fixed number to appoint. There are also 180 men from the enlisted force of the regular army and national guard. Upon graduation a cadet is commissioned as second lieutenant in the regular army, where he is expected to serve for at least 4 years.

These cadets are paid $1,174.20 a year, and from this amount provide themselves with clothing and food and the small necessaries of life. Their funds are kept by a treasurer and are not given to the cadets except in such quantities as are required for any journeys away from the Academy that they are permitted to make.—R. E. Plimpton, secretary of Metropolitan Section.

CENTRIFUGAL CASTINGS

COMMENTING editorially upon a brief paper by L. Cammen on Chromium-Alloy Steel Cast Centrifugally,[1] *The Iron Age* says that this probably is the first account of the casting in a centrifugal machine of a high-carbon chromium steel and the successful production of an alloy-steel casting by the centrifugal process. The chief feature is the attainment directly of a structure that only the most careful and time-consuming forging or rolling, with subsequent heat-treatment, can develop in the same kind of alloy-steel cast as an ingot. The centrifugally cast chromium-steel, unworked and heat-treated, is at least equal in structure and properties to similar metal treated by present methods. To this is added the possibility of producing an alloy with higher carbon and chromium-contents than can be secured in ordinary working by the methods employed at present in foundries producing steel castings.

[1] See *The Iron Age*, Sept. 14, 1922, p. 655.

Questions Answered by the Research Department

MANY of the inquiries answered by the Research Department are of much general interest and involve a considerable amount of thought and research. For the information of our members a few of them will be published each month. It is hoped that in this way those members who have not yet availed themselves of the facilities offered by the Research Department may be encouraged to take advantage of them in the future.

Question:—Could you give me the approximate figure for the Brinell hardness of cast iron?

Answer:—Brinell hardness numbers for cast iron of the composition range used for automobile engine cylinders can be found in the *Foundry Trade Journal*, Vol. 21, pp. 811 and 812, where there is given an abstract of results published by the prominent French metallurgist, Leon Guillet; also in an article by J. E. Hurst on The Heat Treatment of Gray Cast Iron at Low Temperatures, in *Engineering*, Vol. 108, p. 1.

The values given in these two articles were obtained with a 10-mm. ball at 3000-kg. pressure, and vary from 179 to 228 for castings cooled in sand. Both authors are interested in the effect on hardness and wear by the use of the castings under conditions where comparatively high temperatures prevail. In fact, the latter paper gives micrographs of an automobile engine piston that failed because of such a temperature influence, while Guillet's curves show that the hardness begins to diminish at about 600 deg. cent. (1470 deg. fahr.).

There are a few other published figures, but as the two independent sets presented above not only agree well, but also relate to conditions obtaining in automobile engine cylinders, they will probably serve the purpose. References:

L. Guillet. Hardness of Cast Iron. *Foundry Trade Journal*, Vol. 21, pp. 811-812.

J. E. Hurst. Heat Treatment of Cast Iron (Gray) at Low Temperatures. *Engineering*, Vol. 108, p. 1.

H. Hatfield. Cast Iron in the Light of Recent Research. 1918.

Question:—Can you supply me with a list of important recent articles and books on (a) bearing friction and (b) determination of viscosities?

Answer:—The following are a few of the recent articles on bearing friction and determination of viscosities:

The Ball Bearing: In the Making, Under Test, and in Service. By Henry Heathcote. Pamphlet can be secured from the Institution of Automobile Engineers, 28 Victoria Street, London, England. Also published in the *Proceedings* for 1920-1921, p. 569.

Friction and Carrying Capacity of Ball and Roller Bearings. By H. L. Whittemore and S. N. Petrenko. Bureau of Standards Technologic Paper No. 201. Published by the Government Printing Office, City of Washington. Gives an account of experiments undertaken to determine maximum safe load and static friction under load of ball and roller bearings. Results agree roughly with Hertz's theory, differences being ascribable to inhomogeneity of material.

Viscosity and Friction. By Winslow H. Herschel. THE JOURNAL, January, 1922, p. 34.

Macmichael Torsional Viscosimeter. By W. H. Herschel and E. W. Dean. Bureau of Mines Serial No. 2201, January, 1921.

Saybolt Furol Viscosimeter. By E. W. Dean. Bureau of Mines Serial No. 2215, February, 1921.

Determination of Absolute Viscosity by Short-Tube Viscosimeters. By Winslow H. Herschel. Bureau of Standards Technologic Paper No. 100. Published by the Government Printing Office, City of Washington.

Question:—The variation in the performance of trucks introduces some difficulties in interpreting results. We understand that the best passenger cars are designed with a view to securing an acceleration of about 3 ft. per sec. per sec. Is a similar basis being adopted for the design of motor trucks in terms of acceleration or power per ton of weight? Can you give us an idea of the range of tractive effort of various sizes of truck?

Answer:—Using the best estimates we could make for engine torque and mechanical efficiency, and 35 lb. per ton for tractive effort on average roads, the maximum grade and accelerations in direct or high gear work out as shown in the accompanying table. The trucks were selected very much at random, but represent for the most part models that, we believe, are put out in considerable numbers. The weights used in the following table are the chassis and body weights as given by the makers, plus the *rated* load. The figures can be readily recomputed for any desired load. A brief statement of the methods used in computing these results is given below. For obvious reasons, the names of the trucks are not given.

Truck	Rated Capacity, tons	Bore, in.	Stroke, in.	Gear-Reduction	Diameter of Rear Wheel, in.	Total Weight, lb.	Grade, per cent	Mean of Groups[1]
1	½	3⅞	4½	4.16	32	3,850	6.04	6.16 per cent grade. 1.98 ft. per sec. per sec.
2	¾	3 11/16	4	5.50	34	4,119	5.54	
3	¾	3½	5	6.30	34	4,500	6.90	
4	1	3¾	5	6.28	34	5,300	6.65	7.11 per cent grade. 2.29 ft. per sec. per sec.
5	1	3¾	...	6.20	34	5,300	6.53	
6	1	3¾	4	7.25	32	4,400	8.15	
7	1½	4	5¾	7.00	36	7,000	6.93	6.05 per cent grade. 1.95 ft. per sec. per sec.
8	1½	3¾	5	7.60	36	7,000	5.52	
9	1½	3¾	5	7.80	34	7,400	5.69	
10	2	4 3/16	5 1/12	8.00	40	8,800	5.00	4.85 per cent grade. 1.56 ft. per sec. per sec.
11	2	4	5	7.60	36	8,800	4.83	
12	2	4	5½	6.43	34	8,800	4.73	
13	2½	4⅛	5¼	9.00	38	10,650	5.00	4.12 per cent grade. 1.33 ft. per sec. per sec.
14	2½	4⅛	5¼	7.75	36	11,000	4.25	
15	2½	3¾	5¼	8.66	36	11,000	3.79	
16	2½	4	5	7.99	36	11,700	3.43	
17	3	4¼	5	9.00	36	11,700	4.84	4.57 per cent grade. 1.47 ft. per sec. per sec.
18	3	4⅛	5¼	9.45	36	13,200	4.39	
19	3	4¼	5½	9.66	36	14,500	4.52	
20	3½	4½	5½	10.26	36	15,400	5.30	4.97 per cent grade. 1.60 ft. per sec. per sec.
21	3½	4½	6	8.75	40	15,400	4.17	
22	3½	4½	5½	10.33	36	15,400	5.36	
23	3½	4½	6½	10.00	40	16,500	5.06	
24	4	4½	5½	10.25	36	17,600	4.42	3.79 per cent grade. 1.22 ft. per sec. per sec.
25	4	4½	6	8.75	40	18,500	3.15	
26	5	5	6¼	8.80	40	22,000	3.60	3.99 per cent grade. 1.29 ft. per sec. per sec.
27	5	4½	6¾	10.00	40	21,800	3.60	
28	5	5	5½	10.66	40	23,150	3.65	
29	5	4¼	5½	8.70	36	22,000	1.99	
30	5	5	6½	11.66	40	23,500	5.11	4.58 per cent grade. 1.48 ft. per sec. per sec.
31	7½	5	6	11.58	36	26,050	4.58	

[1] Second decimal has no significance.

The method used in the calculation of the tractive effort is given by Cornelius T. Myers in an article entitled Power and Performance of Gasoline Motor Trucks[2] and also in one entitled Chassis Design of Class B Motor Trucks[3]. Accord-

[2] See TRANSACTIONS, vol. 9, part 2, p. 122.

[3] See THE JOURNAL, January, 1922, p. 29.

ing to the former the tractive factor is the tractive effort in pounds divided by the total weight in pounds on the tires or the product of the engine torque and the gear reduction divided by the product of the total weight in pounds on the tires and one-half of the diameter of the driving wheel. The torque is taken as 1.2 of the value corresponding to the National Automobile Chamber of Commerce rating.

$$T = [\{ (33{,}000D^2n \times 12) \div 2.5 \} \div \{ (2\pi \times 1000 \times 12) \div 2L \}] \times 1.2$$
$$= 4.2D^2nL \times 1.2$$
$$\therefore TF = [(8.4D^2nLe) \div Wd] \times 1.2e_m$$

where

d = the diameter of the rear wheel
D = the cylinder bore
e = gear reduction
e_m = the mechanical efficiency
L = the length of stroke
n = the number of cylinders
T = the torque
TF = the tractive factor
W = the total weight on the tires

Assuming a mechanical efficiency of 85 per cent, the formula for four-cylinder engines becomes

$$TF = [(8.4 \times 1.2 \times 0.85 \times 4)D^2Le] \div Wd$$
$$= 34.27D^2Le \div Wd$$

This gives the total tractive effort for each pound of weight of the truck. By subtracting the tractive resistance per pound on good dirt roads, which is taken as 35 lb. per ton or 0.0175 lb. per lb., the tractive effort that is available to take a grade is obtained. This value multiplied by 100 gives the values for the "Grade, per cent" column in the accompanying table. This method, using however a mechanical efficiency of 90 per cent gives an acceleration of from 2.8 to 3.2 ft. per sec. per sec. for passenger cars.

Question:—I wish to know where I could obtain the following information: Initial boiling-point, end-point and range of distillation of some of the substitute gasoline products now offered on the market. Can you describe any method whereby I can derive the British thermal units and the horsepower of any of these products from their gravity and distillation?

Answer:—The best source of information for distillation data for various gasoline substitutes is the Bureau of Mines in the City of Washington. It may be said that the volatility of most of these fuels does not differ radically from that of commercial gasoline. Some of them containing benzol will show larger percentages, boiling around 200 deg. fahr., but this may have little effect on the end-point. There is no direct relation between the British thermal units or horsepower and gravity or distillation range, except for known classes of hydrocarbons. In a series of articles published in the *Automobile Engineer* from February through August, 1921, Ricardo has shown that the horsepower obtainable from a given engine is very nearly the same for a wide range of fuels. There is thus no definite relation between the British thermal units per pound and the horsepower of an engine, though there is of course a relation between the British thermal units and the horsepower hours per pound of fuel.

As for the relation between British thermal units or the horsepower and the gravity or volatility, if one confines his attention to a single hydrocarbon series such as the paraffin series C_nH_{2n-2}, there is a definite relation between the number of British thermal units and either the volatility or the gravity which can be found in the various tables, such as those of Landolt and Börnstein. The same is true of the other series, but not of their indiscriminate mixtures such as occur in some of the fuels that are now to be found on the market.

OBITUARIES

Arthur Benjamin Browne, of Cambridge, Mass., died at the age of 55 years in his native city on Sept. 16, 1922, from cancer of the larynx, after an illness of 22 months. Major Browne was born, Nov. 29, 1866. He was graduated from the Cambridge High School in 1883, and then studied chemical and mining engineering for 5 years under W. French Smith, of the Massachusetts Institute of Technology. From 1889 to 1897 he pursued a general practice as a mining and metallurgical engineer at Boston. During this time he discovered and patented the original electrolytic process for the manufacture of white lead, which received international recognition as a noteworthy contribution to the art. From 1897 to 1908, Major Browne followed a general engineering practice in Montana and Oregon, which led to specialization in internal-combustion engineering as applied to mining and railroading. For 3 years of this time he was engaged as chief engineer in building the Central Railway of Oregon, which he afterward operated as its superintendent.

Following a prolonged illness in 1911, he spent 2 years in study and development work relating to gasoline and oil engines, giving particular attention to the constant-pressure or Brayton type. The years 1913 to 1917 inclusive were spent in research and development in the field of carburetion. During this period he lived in Branford, Conn., and acted as consulting engineer for the Malleable Iron Fittings Co. of that place. In 1915 he wrote an engineering treatise bearing the title, Handbook of Carburetion, which is one of few authoritative books on the subject treated.

Shortly following the entry of this Country into the World War, Mr. Browne was awarded a captain's commission in the United States Army and assigned to engineering duty in the Sanitary Corps. For several months he was in charge of the production, testing and inspection of ambulance and other truck chassis at the General Motors Truck Co.'s plant in Pontiac, Mich. He was later transferred to the City of Washington, where he was promoted to the rank of Major and served as a representative of the Sanitary Corps on the board of engineers charged with the test and final selection of vehicles for use by the Army. Some months later he was made chief of the motor branch, motor and vehicle division of the Department of Purchase, Storage and Traffic.

Upon his discharge from the service in July, 1919, Major Browne accepted the position of chief engineer and director of sales of the A. J. Detlaff Co., Detroit, from which he resigned about a year later. For several months in 1920 he served as consulting engineer of the Trent Process Co., in the City of Washington. Until a few months prior to his death he was engaged in consulting and development work relating to automotive clutches, carbureters and engines.

Major Browne was elected to Member grade in the Society, May 26, 1913. He served as a member of its Standards Committee and contributed several valuable papers.

William T. Murphy, president and general manager of the Standard Machinery Co., Auburn, R. I., died, July 9, 1922, aged 40 years. He was born, June 1, 1882, at Providence, R. I. In June, 1903, he was graduated from Brown University with the degree of mechanical engineer.

Mr. Murphy was employed as a machinist by the Taft-Peirce Mfg. Co., Woonsocket, R. I., from 1903 to 1904, and then by the Gorham Mfg. Co. in Providence as a designer of special shop equipment until 1905, when he became a tool designer in the automobile department of the American Locomotive Co. in Providence and continued this work until 1907. From 1904 to 1907 Mr. Murphy was also connected with the Rhode Island School of Design as a teacher of evening classes in machine design.

His connection with the Standard Machinery Co. dates from 1907. His specialties were anti-friction bearings and sheet-metal machinery, and he designed numerous ball and roller bearings for automobile, truck and aircraft usage, as well as having also designed and built many machines for producing special automobile parts. Mr. Murphy was elected to Member grade in the Society, April 22, 1918.

Current Standardization Work

DURING the months of September and October 15 meetings of Divisions and Subdivisions of the Standards Committee were held at which progress was made on many important subjects assigned to the Divisions and several definite recommendations were approved for consideration at the Standards Committee Meeting on Jan. 9. The complete recommendations of the various Divisions will be published in the December issue of THE JOURNAL under the usual title of Reports of Divisions to the Standards Committee. Several Division meetings will be held early in November, the dates of these being as follows:

Agricultural Power Equipment Division	Nov. 7	Chicago
Engine Division	Nov. 6	Chicago
Electric Vehicle Division	Nov. 3	Detroit
Passenger-Car Division	Nov. 6	Chicago
Passenger-Car Body Division	Nov. 2	Detroit
Stationary-Engine Division	Nov. 7	Chicago

At the meeting of the Storage Battery Division held on Sept. 15 in New York City final action was taken on recommendations for monobloc containers and isolated electric lighting-plant battery ratings. The Lighting Division Subdivision on Bases, Sockets and Connectors held a meeting on Sept. 19 in Cleveland, reporting at the meeting of the Lighting Division in the same city on the following day. The Truck Division met on Sept. 22 in Detroit, discussion centering on the subjects of three-joint propeller-shafts, power take-off shaft-ends, motor-truck cabs and motor-truck ratings.

The Electrical Equipment Subdivision on Magnet Wire met in New York City on Sept. 22 at which a tentative recommendation was adopted which was considered at the Electrical Equipment Division meeting on Oct. 2 in Rochester. At this meeting reports were also submitted on flexible conduit, magneto mountings, generator and starting-motor mountings and spark-plugs. Detailed accounts of these reports will be found under the various subject-headings in this article. The Frames Division held a meeting on Sept. 25 in Detroit at which definite recommendations for frame and running-board brackets were approved. The Lubricants Division members met on Sept. 26 in Cleveland, the tentative crankcase lubricating oil specifications that were discussed at the Summer Meeting of the Society being reviewed and revised in part. The Motorboat Division held a meeting in New York City on Sept. 28 at which Subdivision reports were submitted covering motorboat couplings, exhaust-manifold connections, tachometer drives and stuffing-boxes. Many subjects were suggested for standardization in connection with boat hardware which will be taken up by the Division in the near future.

The Axle and Wheels and the Ball and Roller Bearings Divisions held a joint meeting in Detroit on Oct. 4, this meeting being followed by separate Division meetings at which subjects of special interest to each Division were discussed. The principal subject for joint discussion was the work of the Subdivision on Passenger-Car Front-Axle Hubs and Differentials. The Iron and Steel Subdivision on Sheet Steel held a meeting in Detroit on Oct. 7. The Springs Division members met in Detroit on Oct. 24, several reports being submitted by the Subdivisions appointed at the previous meeting.

The Iron and Steel Division members met on Oct. 30 in New York City. The Parts and Fittings Division members met in Detroit on Oct. 30, a Subdivision report on Felt Specifications being submitted in the afternoon as a result of a Subdivision meeting held during the morning of the same day. The Transmission Division also held a meeting in Detroit on Oct. 31.

BASES, SOCKETS AND CONNECTORS

The Subdivision on Bases, Sockets and Connectors was appointed to review the present standards for bases, sockets and connectors, pp. B4 and B5 of the S.A.E. HANDBOOK, and to recommend such revisions as are necessary to make them satisfactory for truck practice. At the meeting of the Subdivision on Sept. 19 in Cleveland, it was considered, however, that the present standards are so firmly established in practice that it would be unwise to recommend any changes. A proposal to change the limits on the lamp base, which are now 0.592 and 0.600 in., so as to reduce the allowable tolerance, was considered impossible as the wall is so thin that distortion occurs when the hot glass bulb is inserted. Manufacturers hold the lamp base to within 0.003 in., but the additional 0.005-in. tolerance is necessary in order that the finished lamp may come within production requirements.

It was thought desirable to add an additional dimension to facilitate checking, this dimension to be the distance from the end of the pin to the end of the base, including the solder; to eliminate the notes stating that the pins and slots shall not vary from the center-lines by more than 0.003 in.; and to specify the dimensions of a ring and plug gage, for use in checking the location of the pins for both the base and the plug and in checking the slots in the connector, in accordance with the dimensions specified in Fig. 1.

To provide for manufacturing tolerances, it was also recommended that the slots in the connector tube should be changed from a fixed dimension of 3/32 in. to limits of 0.094 and 0.104 in., and that the following notes referring to the focusing of sockets should be inserted:

> All sockets in which focusing-type electric incandescent lamps are used shall be provided with some means for holding the bulb base firmly throughout its entire length in the socket-shell, because of the difficulty in maintaining the tolerances specified for the base and socket and to eliminate variations in the alignment of the light source.

As the sockets specified in the present standards are of the old spring-plunger type that has given trouble because of poor electrical contact, it was recommended that the design of the socket be changed to one with the spring on the outside of the plunger, inserting the following note:

> It is recommended that all sliding contacts and current-carrying springs be eliminated and that plungers with springs on the outside be used so that the current will pass through the plungers and not through the springs.

As the present standards do not cover the metal type of plug and cap, it was decided that the Subdivision would extend the present standards to cover this type so as to permit interchangeability. Such standardization was considered important as several automobile companies are using armored lighting-cable which cannot be used with the composition cap specified in the present standards.

The Subdivision reported progress at the meeting of the Lighting Division on Sept. 20 and in view of the importance of the work the Subdivision personnel was extended to in-

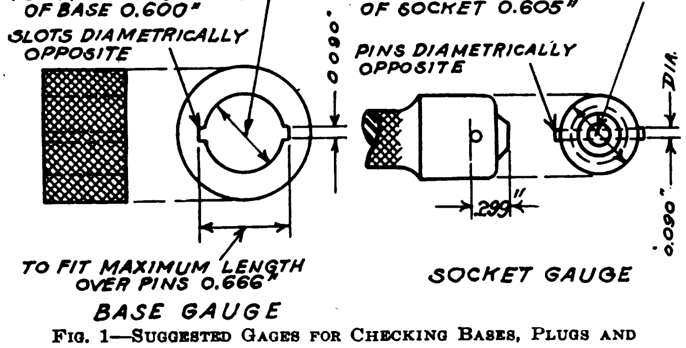


FIG. 1—SUGGESTED GAGES FOR CHECKING BASES, PLUGS AND CONNECTORS

clude representatives of automobile companies, the complete personnel now being as follows:

C. E. Godley, chairman	Edmunds & James Corporation
Ernest Wooler	Cleveland Automobile Co.
T. I. Walker	Providence Base Works
J. C. Stearns	Culver-Stearns Mfg. Co.
A. K. Brumbaugh	Autocar Co.
G. A. Walters	Chicago Electric Mfg. Co.
J. T. Caldwell	National Lamp Works

The elimination of the insulated-return system was discussed at the Division meeting, but as this system is used to some extent on motor trucks, it was retained in the standard. It was thought that most of the trouble experienced in motor-truck practice is due to the unsubstantial construction of lighting fittings, the heavy vibration causing them to fail. It was suggested that a series of large and more substantial fittings should be standardized.

Body Nomenclature

An editorial recently appeared in one of the automobile magazines criticising the action of the Society in recommending the term "phaeton" instead of "touring car." The following, quoted from a letter written by Geo. J. Mercer as an answer to the criticism, will be of value to anyone interested in considering the justification of the term "phaeton."

> The writer agrees heartily with that portion of your editorial in which it is stated, "It is usually a mistake to overturn standards, once the public has established them." It was precisely along these lines that the committee worked in recommending a name for each type of body. Due consideration was given to established practice and to the fact that the committee was setting a standard for the future as well as for the present. Consideration was given also to the origin or derivation of the names.
>
> As proof that the committee bore the above considerations in mind, take two of the names selected, "brougham" and "cabriolet." Neither has a derivative sense as used. There are several outstanding features that were characteristic of the type of body known as the brougham in the carriage days and in the early days of the automobile, that are lacking in the automobile body which the committee has selected to bear this name. At the Salon one or two genuine broughams are invariably exhibited. Yet the committee has given this name to the type shown on p. K23 of the S.A.E. Handbook because the general public had seen fit to use the name selected. In addition, it was felt that the older or original type was not suitable, in general, for use as an automobile body type. We agreed, therefore, to pass nomenclature on to successors and in naming both the brougham and the cabriolet, we were actuated solely by the fact *that the public commonly recognized these bodies by the terms mentioned.* The writer wishes to add that he has, until within the past two years, combatted the use of these names for these now established types and has been won over to their use only because he has been guided by the principle of usage.
>
> In naming the bodies at this late date we felt that we would meet with considerable opposition, because in nearly every case we had to choose one of several names, but uptodate your editorial is the only objection voiced on behalf of the automobile public. We have had the commendation and the support of the Automobile Body Builders' Association which has unanimously accepted the nomenclature.
>
> One reason the committee chose the name "phaeton" is that mentioned in your editorial; the term "touring" applies to any type of body because all types are used for touring, but that is not the only reason. This type of body is the logical successor of the family horse-drawn open carriage known as the phaeton. It was so called in the earliest handbooks issued by the Association of Licensed Automobile Manufacturers. In the 1906 Handbook, Packard, Walters and Stevens-Duryea used the name "phaeton," Pierce-Arrow designated the vehicle in question as a "tonneau" and others used simply the series letters or numbers of the respective models without names. Later, as the design was changed and became a flush-side body such as we know to-day, "torpedo" was the common name in Great Britain and was used to some extent in this Country; while in France the use of the name "phaeton" was general. Packard, Franklin, Cadillac, Stevens-Duryea, Cunningham, Pope-Hartford and Thomas used the name "phaeton" in the 1913 Association of Licensed Automobile Manufacturers Handbook. Hudson has been a consistent user of this term.
>
> This certainly warranted the committee in concluding that the name "phaeton" has a logical right from a priority point of view, based solely on automobile custom, to contend with the name "touring car." Furthermore, as a name it has a wonderful background of history. It is associated with the type of carriage that people have owned and used beyond our memory. It is also applicable to the more expensive carriages that were exhibited at the horse shows.
>
> As it was the task of the committee to select one name from the several in use for each of the various types of body, due consideration was given to the claims of all and the decisions were based on the facts outlined herein. We feel that our work has met with general approval.

Engine Rotation

At the meeting of the Motorboat Division members on Sept. 28, the possibility of standardizing the direction of engine rotation was discussed in conjunction with replies received from motorboat engine manufacturers. It was the consensus of opinion that it would be impossible to recommend definitely either right or left-hand rotation, but that definite nomenclature should be approved as to what constitutes right and left-hand engine rotation.

Lamp Glasses

At the last meeting of the Lighting Division it was suggested that the present standard for lamp glasses, p. B6 of the S.A.E. Handbook, should be revised so as to make it more satisfactory for various devices used by lens manufacturers. A semi-circular notch was suggested, but not approved by the lamp manufacturers as it would require redesigning the lamp doors and conflict with servicing the large number of lamps now in use. It was decided, however, to modify the form of the notch and a tentative recommendation was approved which will be tried out by the lamp manufacturers before it is finally approved by the Division.

Crankcase Lubricating Oils

The Lubricants Division submitted a report at the Summer Meeting of the Society covering tentative specifications for crankcase lubricating oils, the report being printed in full on p. 533 of the June issue of The Journal.

Subsequent to tentative approval of this report the Division has carried out a large number of tests on five different kinds of oil. The tests were made by various members of the Division on samples of the oil, identified only by number, which were distributed by the chairman. The results of the tests were discussed at the meeting of the Lubricants Division on Sept. 26 and indicated that the error in testing oils in accordance with the proposed specifications would vary considerably even among those actually connected with the petroleum or the automotive industries. It was therefore felt that the limits should not be too close, but on the other hand should be close enough to result in fairly representative test results. It was generally agreed that the principal purpose of the proposed standard is to provide information to the automotive engineers that will be satisfactory as a purchase specification. The specifications as printed in the June issue of The Journal were further amplified by the

inclusion of general descriptive grade names in addition to the specification numbers. This action was taken in view of the general opinion that the specifications would be more widely used if both methods of identifying the different oils were recommended. Other changes were made in the various values as published in June, the revised tentative specifications being given in the accompanying table.

In view of the work of the Governmental Interdepartmental Committee and the American Petroleum Institute, it was considered advisable at the meeting to arrange for a joint session of these organizations and the Lubricants Division in order that a common specification might be formulated which would meet with general approval.

CRANKCASE LUBRICATING OILS

General. — These specifications cover grades of petroleum oil for the lubrication of internal-combustion engines, except aircraft, and are not recommended for the lubrication of turbines.

Compounded lubricating oils containing products other than those derived from petroleum are not dealt with in these specifications.

ments which had been submitted. The specification was approved for adoption as S.A.E. Recommended Practice in the following form:

MONOBLOC CONTAINERS FOR STARTING AND LIGHTING BATTERIES

Height. — Containers shall be made in two heights only; namely, the B height for plates approximately 4¾ in. high, and the C height for plates approximately 5¼ in. high.

Overall Height.—The height from the outside of the bottom of the case to the top of the handle shall not exceed 9¼ in. for B containers, or 9⅞ in. for C containers.

Height from Top of Ribs to Top of Container.—These heights shall be:

B containers 6¼ in. } Plus 0
C containers 7 in. } Minus 1/16 in.

Maximum variation between different compartments, 3/32 in.

Inside Width of Compartments.—5 31/32 in., plus or minus 1/32 in.

Spec. No.[1]	Supplementary Grade Names	Flash-Point, Deg. Fahr., Min.	Fire-Point, Deg. Fahr., Min.	Viscosity, Saybolt Sec. 100 Deg. Fahr. Min.	100 Deg. Fahr. Max.	210 Deg. Fahr. Min.	210 Deg. Fahr. Max.	Color[2] (NPA)	Pour Test, Deg. Fahr., Max.	Conradson Carbon Residue, per cent Max.	Corrosion Test
20	Light	325	365	180	220	42	....	5	35	0.20	
020	Light	325	365	180	220	42	....	5	10	0.20	
30	Medium	335	380	270	330	44	....	5	40	0.30	
030	Medium	335	380	270	330	44	....	5	10	0.30	Required for all grades
40	Heavy	345	390	360	440	46	....	5	45	0.40	
50	Extra Heavy	355	400	450	575	50	....	6	50	0.60	
60	xx Heavy	360	....	....	....	55	65	....	55	0.80	
80	xxx Heavy	380	....	....	....	75	85	....	55	1.50	
95	xxxx Heavy	390	....	....	....	90	100	....	55	1.75	
115	xxxxx Heavy	400	....	....	....	110	120	....	60	2.00	

[1] For Specifications Nos. 20 to 50, inclusive, the numbers indicate the first two figures of the average Saybolt viscosity in seconds at 100 deg. fahr. for the grades indicated. The first cipher in Specifications Nos. 020 and 030 indicates that the pour-test value of these two grades is 10 deg. fahr. The numbers for Specifications Nos. 60 to 115, inclusive, indicate the average Saybolt viscosity in seconds at 210 deg. fahr.

[2] Darkest color allowed for a mixture of 50 per cent oil and 50 per cent kerosene.

Corrosion Test.—The following corrosion test shall not cause discoloration of copper strip. Place a clean piece of mechanically polished pure strip-copper about ½ in. wide and 3 in. long, and 10 cc. of the oil to be tested, in a clean test-tube. Close the tube with a vented stopper and hold for 3 hr. at 212 deg. fahr. Rinse the copper strip with sulphur-free acetone and compare it with a similar strip of freshly polished copper.

MOTORBOAT NOMENCLATURE

It was brought out at the last meeting of the Motorboat Division that the establishment of nomenclature for engine and motorboat parts is of vital importance to the motorboat industry and that the present automobile engine nomenclature now published in the S.A.E. HANDBOOK could be used to advantage as a basis for such nomenclature. A Subdivision was appointed, consisting of Leonard Octman, Jr., W. J. Deed and Irwin Chase, to initiate the work, it being understood that the personnel will be added to as the work shall require. The Society was asked to obtain catalogs and parts-lists from engine and motorboat builders for use in the Subdivision work.

MONOBLOC CONTAINERS

At the Storage Battery Division meeting on Sept. 15 the tentative specifications for monobloc containers were reviewed and several changes made therein as a result of com-

Inside Lengths of Compartments.—(a) 6-Compartment Containers

B Containers		C Containers	
S-3-B	1 5/16 in.	S-4-C	1 1/2 in.
S-4-B	1 1/2 in.	S-5-C	1 11/16 in.
S-5-B	1 11/16 in.		

(b) 3-Compartment Containers

B Containers		C Containers	
S- 7-B	2 1/16 in.	S- 8-C	2 3/8 in.
S- 8-B	2 3/8 in.	S-10-C	2 13/16 in.
S- 9-B	2 7/16 in.	S-13-C	3 1/4 in.
S-10-B	2 13/16 in.	S-14-C	3 5/16 in.
S-16-B	3 11/16 in.	S-16-C	3 11/16 in.
S-18-B	3 15/16 in.		
S-19-B	4 1/8 in.		

(c) Tolerances of plus 0 and minus 1/32 in. shall apply to container lengths of 3 5/16 in. or less and tolerances of plus 1/64 in. and minus 1/32 in. to lengths of over 3 5/16 in.

Partitions Between Compartments.—The thickness of the partitions between compartments shall be 3/16 in. minimum and ¼ in. maximum.

Overall Width.—The overall width shall not exceed 7½ in.

GENERATOR AND STARTING-MOTOR MOUNTINGS

At the meeting of the Electrical Equipment Division on Oct. 2, T. L. Lee, as chairman of the Subdivision on Gen-

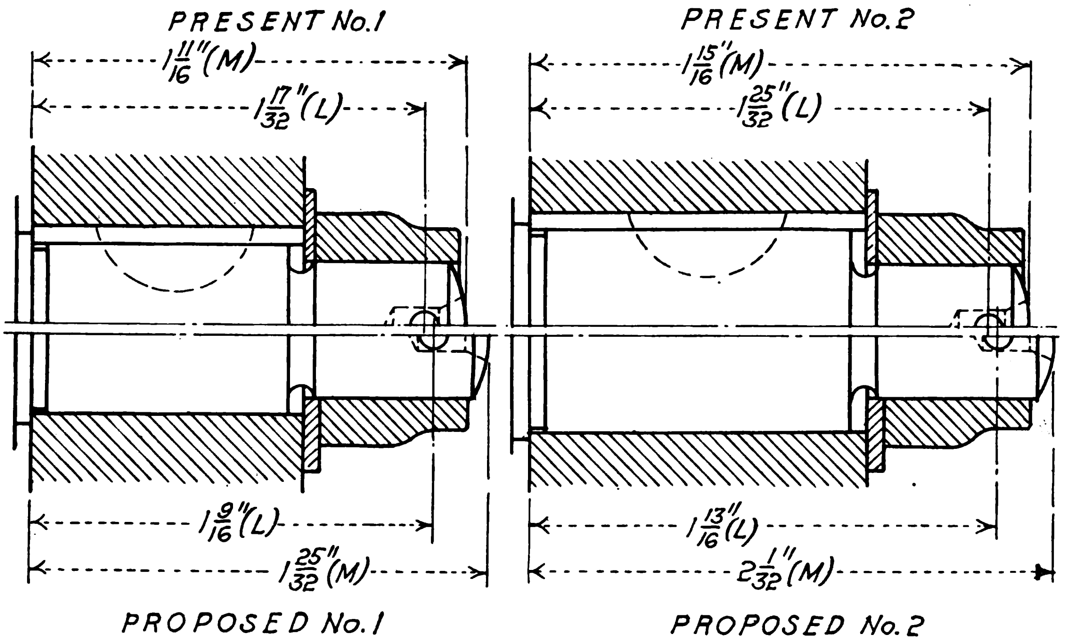


FIG. 2—PROPOSED CHANGES IN GENERATOR FLANGE MOUNTINGS

erator and Starting-Motor Mountings, submitted several recommendations relating to the present standards. As these recommendations had been submitted to generator and starting-motor manufacturers and users and had met with general approval, the Electrical Equipment Division approved them as given herewith.

GENERATOR FLANGE MOUNTINGS

As considerable trouble has been experienced with the present S.A.E. Standard Nos. 1 and 2 generator flange shaft extensions because there is insufficient room for a washer between the nut and the pinion and because of the location of the cotter-pin, the Subdivision recommends that dimension L, Fig. 2, be increased by 1/32 in. and dimension M by 3/32 in. The revised dimensions allow sufficient room for a washer to be used and locate the cotter-pin hole so that it will not break into the shaft center.

To provide engine builders with all information necessary for making provision for the mounting of S.A.E. Standard flanges, the Subdivision also recommends that

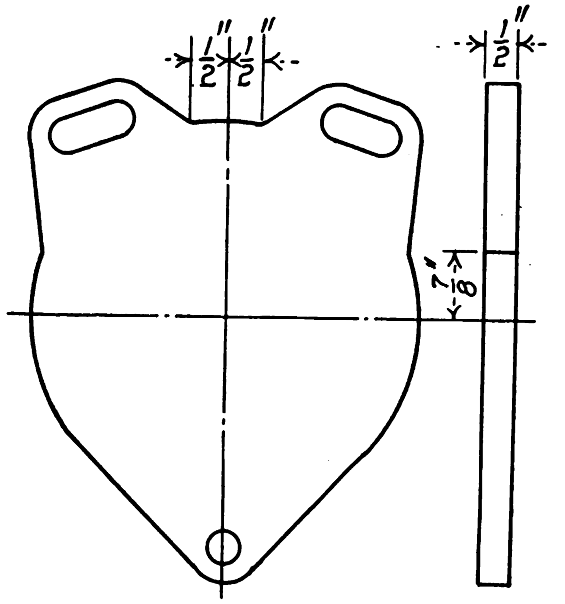


FIG. 3—PROPOSED ADDITIONAL DIMENSIONS FOR GENERATOR FLANGE MOUNTINGS

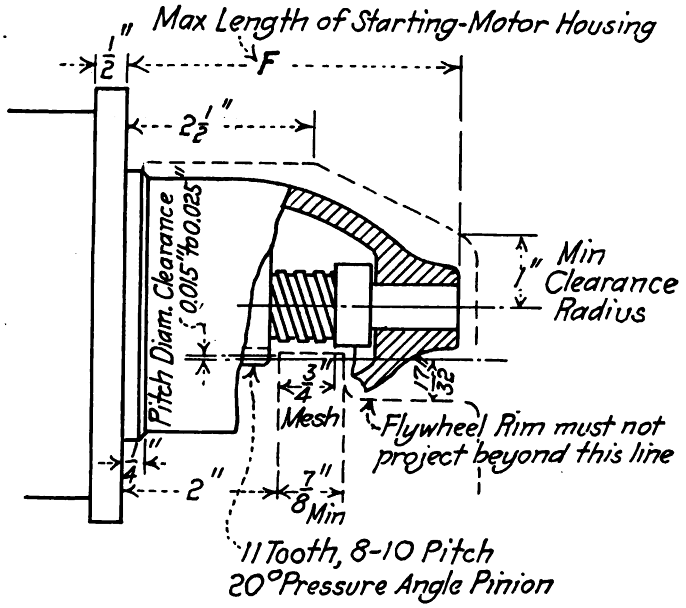


FIG. 4—PROPOSED MOUNTING DIMENSIONS FOR STARTING-MOTOR FLANGE

Dimension F Is 4 7/16 In. Maximum for the 10, 11, 12 and 13-Tooth Small Hollow Shift and 4¾ In. for the 13-Tooth Large Hollow Shift

the additional contour dimensions shown in Fig. 3 be included in the present standard and that the thickness of the flange be specified as ½ in.

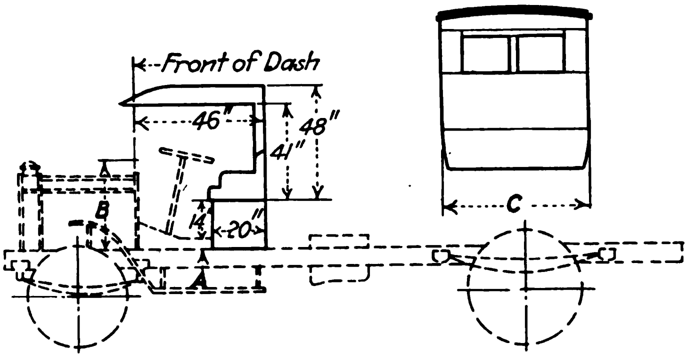
STARTING-MOTOR FLANGE MOUNTINGS

In view of the demand for an S.A.E. Standard for Starting-Motor Flange Mountings for outboard installations only, the Subdivision recommends the adoption of the mounting dimensions given in Fig. 4.

The Subdivision also recommends the adoption of a flange thickness of ½ in. for the present S.A.E. Standard for Starting-Motor Flange Mountings which is printed on p. B19 of the S.A.E. HANDBOOK.

MOTOR-TRUCK CABS

Early last spring it was decided to undertake the standardization of motor-truck cab mounting-dimensions as it was considered important to make possible the production of stock



Truck Capacity, Tons	1, 1½, 2, 2½	3, 3½, 4, 5
A	3½	4
B[1]	35	39
C	48	56

[1] *Minimum for windshield lower edge*

Where cab sides, doors or side curtains are used, care should be taken that the driver's vision is interfered with as little as possible

FIG. 5—MOUNTING DIMENSIONS OF MOTOR-TRUCK CABS

cabs in order that users might not be obliged to have cabs specially designed with the resulting loss of time and added cost. Information was obtained from truck builders as to the mounting dimensions required for cabs used on their trucks, this information being tabulated and a tentative recommendation based thereon. The recommendation was reviewed at the last meeting of the Truck Division together with comments which had been obtained from truck and cab builders and as a result the recommendation was revised in some details.

The table of dimensions shown in Fig. 5 was approved by the Division members subsequent to this meeting and will be submitted at the Standards Committee meeting for adoption as S.A.E. Recommended Practice. Copies of the report will be sent to all Society members and others interested in order that comments may be submitted prior to the Standards Committee Meeting.

POWER TAKE-OFF SHAFT-ENDS

J. R. Coleman has been appointed a Subdivision of one to review information obtained by the Society on power take-off shaft-ends and to submit a report to the Truck Division for consideration.

RATINGS OF STORAGE BATTERIES FOR ISOLATED ELECTRIC LIGHTING-PLANTS

The history of the Standards Committee work in endeavoring to establish ratings for storage batteries for isolated electric lighting-plants was reviewed at the Sept. 15 meeting of the Storage Battery Division, with particular reference to the action of the Isolated Electric Lighting-Plant Division in recommending at a Division meeting held on July 28, 1921, a standard rating of capacity expressed in terms of watt-hours based on a continuous 8-hr. discharge test, which recommendation did not receive general approval as determined by a letter ballot of storage battery and lighting-plant manufacturers.

In view of the fact that at the present time several methods of rating storage batteries are used, it was considered advisable to recommend that either the intermittent or the continuous ratings should be used in accordance with the recommendation of the Storage Battery Subdivision of the Electric Equipment Division made at a meeting on May 28, 1919, in Indianapolis, the present Storage Battery Division being the successor of the Storage Battery Subdivision in existence at that time.

The following recommendation was therefore formulated and will be submitted to the Standards Committee for adoption as an S.A.E. Recommended Practice:

RATINGS OF STORAGE BATTERIES FOR ISOLATED ELECTRIC LIGHTING-PLANTS

Lead-acid storage batteries for isolated electric lighting-plant service shall have two ratings, a continuous rating and an intermittent rating. The ratings shall be determined at an initial temperature of 80 deg. fahr. and shall be based on a final voltage of not less than 1.75 volts per cell.

Continuous Rating.—The continuous rating shall be the capacity in ampere-hours of the battery when it is discharged continuously at the 8-hr. rate.

Intermittent Rating.—The intermittent rating shall be the ampere-hour capacity of the battery at a current rate equal to 1/24 of the said ampere-hour capacity when the discharge is spread over 72 hr.

To avoid night work, the following discharge and rest periods are suggested:

> An initial discharge period of 4 hr., followed by a 16-hr. rest; then two 8-hr. discharge periods, each followed by a 16-hr. rest; the final discharge period being 4 hr.

The short periods at the beginning and at the end permit the test to begin at noon of the first day and end at noon of the fourth day.

Except for the final voltage per cell the ratings apply to nickel-iron batteries as well as lead-acid batteries.

Battery manufacturers should specify both ratings in their catalogs using the following form:

S.A.E. Ampere-Hour Capacity Ratings: 100-140

The first number represents the capacity based on the continuous rating and the second number the capacity based on the intermittent rating.

In making this recommendation it was understood that if one of the two ratings proposed is eliminated by subsequent Standards Committee or Society action, the Division recommendation shall be referred back as it is considered that the situation can be covered adequately only by specifying both the continuous and intermittent ratings.

RUNNING-BOARD BRACKETS

The present S.A.E. Recommended Practice for Running-Board Brackets printed on p. H23 of the S.A.E. HANDBOOK was reviewed at the meeting of the Frames Division members on Sept. 25 with special reference to the thickness of stock, which is not specified in the present standard. After consideration of comments which had been received in this connection it was proposed that the recommended practice be amplified to specify a stock thickness of 5/32 in. (0.156 in. or No. 9 U. S. gage).

SPARK-PLUGS

At the meeting of the Spark-Plug Subdivision on Aug. 17 the present standard for spark-plug shells, p. A10 of the S.A.E. HANDBOOK, was reviewed in detail. In reference to the thread size, it was stated that two-thirds of the spark-plugs manufactured at the present time have a ½-in. pipe thread, but that the general objection to pipe threads for spark-plugs seems to be that it is extremely difficult to maintain accurately the distance that the spark-plug screws into the cylinder, owing to the fact that the tap and dies are not run down to a constant depth; that metric spark-plugs are used only by the Wills Sainte Claire Co., the Hudson Motor Car Co., Essex Motors, Rolls-Royce of America, Inc. and White Motor Co.; that spark-plugs with ¾-in. pipe threads are being gradually discontinued, as are also the few 5/16 and ⅜-in. spark-plugs with pipe threads; and that the majority of spark-plugs *exclusive of the sizes mentioned* are made with the S.A.E. Standard thread of ⅞ in. —18. It was therefore considered advisable by the Subdivision to retain

the present standard thread as well as all dimensions affecting that portion of the spark-plug shell below the gasket.

In discussing the width across the flats of the hexagon or the "hex diameter" as it is generally called, it was brought out that 20 per cent of the spark-plugs with a ⅞-in. —18 thread have a ⅞-in. hex diameter, 20 per cent have a 1⅛-in. hex diameter and 60 per cent have a 15/16-in. hex diameter. Although there are two or three large users of the ⅞-in. hex spark-plugs, it was considered inadvisable to continue to recommend the use of this size because it does not leave sufficient wall to stand up under wrench strain, especially in two-piece spark-plug construction. It was recognized that it is possible to design two-piece spark-plugs for a ⅞-in. hex but as it would necessitate special inside parts that would not be used on the standard one-piece spark-plug, expensive tool-changes would be required. It would be, however, possible for machinery now used in making ⅞-in.-hex spark-plugs to be reset so as to make the 15/16-in.-hex without additional tooling expense.

The Subdivision considered it advisable to recommend dimensions that would permit interchangeability of different makes of spark-plug having the same type of terminal, and therefore proposed for adoption the following terminal dimensions:

Threaded Type.—The terminal thread shall be No. 8-32 (0.164-in. diameter) S.A.E. Coarse Thread.

Ball Type.—For the ball type of spark-plug terminal the ball diameter shall be 13/32 in.

Slip Type.—For the slip type of spark plug terminal the ball diameter shall be 13/32 in. and shall have a groove width of 0.043 in. plus 0.006 minus 0.000 and a groove diameter of 0.203 in. plus 0.015 minus 0.000.

Post Type.—For the post type of spark-plug terminal the large diameter of the taper post shall be 0.250 in. plus 0.000 and minus 0.003 and the neck diameter 0.220 in. plus 0.000 and minus 0.008.

At the meeting of the Electrical Equipment Division on Oct. 2, the Division approved the report of the Subdivision after careful consideration of the problems involved. It was recognized that the proposed revisions could not be adopted in practice by certain automobile companies, but it was thought that the revised standards would be adopted at some future time by such companies simultaneously with other changes in engine design.

Starting-Motor Pinions

The Electrical Equipment Division has recommended that the present S.A.E. Standard for Starting-Motor Pinions be revised to read:

Starting motors shall be equipped with an 8-10-pitch, 11-tooth, 20-deg-pressure-angle pinion. The starting motor shall be installed so that the pitch-circle about which the teeth on the pinion are generated will be separated from 0.015 to 0.025 in. from the pitch-circle about which the teeth on the flywheel are generated.

Tail-Lamp Illumination

The establishing of definite specifications for tail-lamp illumination was discussed at the Sept. 20 meeting of the Lighting Division. Tests which were carried out by Division members Godley, Kenyon and Porter were described, it being stated that the results indicate that the following are the principal factors that should be included in a definite specification:

(1) The tail-lamp should be located over the center of the license-plate
(2) The light opening in the tail-lamp should be large enough to illuminate directly the entire length of longest plates used
(3) The candlepower of the incandescent lamp should be specified
(4) The license-plate should be set at an angle with the direction of the beam of a light so as to secure more direct illumination of the plate
(5) The specification should cover the worst combination of letters, figures and colors

Three-Joint Propeller-Shafts

At the time the present S.A.E. Recommended Practice for three-joint propeller-shafts was voted upon by the Society members, one or two criticisms were submitted in reference to the use of the square type of shaft-end. This matter was reviewed at the last meeting of the Truck Division and, as it was felt that it is preferable to use either the taper or spline type of shaft-end, the Division recommended that the present square type be omitted. It was thought that in case a square type of shaft-end is desired for use at this point, the present S.A.E. Standard square-shaft fitting should be used as a basis for the shaft-end design.

Transmission Lubricating Oils

At the meeting of the Lubricants Division on Sept. 26, the Subdivision on Transmission Lubricants submitted a report covering transmission lubricating-oil specifications that was approved with slight revisions for adoption as S.A.E. Recommended Practice. The revised report is given in the accompanying table.

TRANSMISSION LUBRICATING OILS

These specifications cover grades of petroleum oil suitable for the lubrication of transmission gears, differential gears, worm drives and roller and ball bearings used in connection with such motor-vehicle equipment.

Compounded lubricating oils containing products other than those derived from petroleum are not covered by these specifications.

Specification No.	Grade	Flash-Point, Deg. Fahr.,[4] Min.	Viscosity at 210 Deg. Fahr., Saybolt Sec.		Pour Test, Deg. Fahr., Max. (American Society for Testing Materials Method)
			Min.	Max.	
110	Winter	350	100	120	10
160	Summer	450	150	170	45

[4]American Society for Testing Materials, Cleveland Open Cup.

Activities of the Sections

Secretaries of the Sections

BUFFALO SECTION
A. J. Fitzgibbons, 168 Claremont Avenue, Buffalo
CLEVELAND SECTION
E. W. Weaver, 5103 Euclid Avenue, Cleveland
DAYTON SECTION
R. B. May, Dayton Engineering Laboratories, Dayton
DETROIT SECTION
Thomas J. Lítle, Jr., 733 Seyburn Avenue, Detroit
Mrs. B. Brede, Assistant Secretary, 416 Capitol Theater Building, Detroit
INDIANA SECTION
B. F. Kelly, Weidely Motors Co., Indianapolis
METROPOLITAN SECTION
R. E. Plimpton, 129 East 45th Street, New York City
MID-WEST SECTION
H. O. K. Meister, Hyatt Roller Bearing Co., 2715 South Michigan Avenue, Chicago
MINNEAPOLIS SECTION
Phil N. Overman, 10 South 10th Street, Minneapolis
NEW ENGLAND SECTION
V. A. Nielsen, 701 Beacon Street, Boston
PENNSYLVANIA SECTION
Edward L. Clark, Hunting Park and Rising Sun Avenues, Philadelphia
WASHINGTON SECTION
Benjamin R. Newcomb, 211 Victor Building, City of Washington

A GLANCE through the following reports of Sections meetings activity will convince any member that the local meetings of the Society are constantly rising to a higher plane. The papers and speakers selected reflect the careful planning of the Sections officers in their effort to make Sections meetings attractive in themselves without requiring the misdirected appeal for loyalty and support to build audiences. If you are taking a passive interest in Sections meetings, particularly those in your own district, you are overlooking an opportunity to secure advanced engineering thought that has a tangible value in your work.

There is a mistaken idea in some quarters that Sections meetings are inferior in value and interest to the national Society meetings. Nothing could be farther from the truth. It is not unusual to find Section papers that surpass those given at the Annual and Summer meetings. Discussion is very thorough at meetings of the Sections because of the longer time available for this phase of the program. The author has time to go into his subject in a very comprehensive way. There is an atmosphere of informality in Sections meetings which approaches that of the factory conference. Regular attendance soon enables one to maintain a personal contact with the engineers and production men in the industry that leads to the forming of intimate friendships which cannot easily be acquired in any other way. Study the following paragraphs and attend the meetings that concern you and your branch of the industry.

Schedule of Sections Meetings

NOVEMBER

1—MINNEAPOLIS SECTION — Manufacture of Gray Iron Pistons—George G. Bouthinon

9—INDIANA SECTION—The Relation of the Engineering and Service Problems — Fred E. Moskovics

16—METROPOLITAN SECTION — Regulations Governing the Use of Highways—A. L. McMurtry

17—MID-WEST SECTION—BUSES—G. A. Green

17—CLEVELAND SECTION—Gliders—Georg Madelung

17—BUFFALO SECTION — Air-Cooled Engines — C. P. Grimes

23—PENNSYLVANIA SECTION — Leaf Springs — H. B. Winchell

24—NEW ENGLAND SECTION — Automotive Electric Service—M. B. Speer

24—DETROIT SECTION

PHOTOGRAPHS OF SECTIONS TREASURERS

Photographs of the Treasurers of the Sections are printed on the facing page. This completes the series of photographs of the officers of the Sections as far as it has been possible to secure likenesses from the individuals concerned. The following list gives the names of the Treasurers and their Sections.

Name	*Section*
Otto Burkhardt	Buffalo
J. S. Clapper	Minneapolis
H. E. Figgie	Cleveland
G. Walker Gilmer, Jr.	Pennsylvania
W. P. Kennedy	Metropolitan
Robert F. McCann	Dayton
Nelson B. Nelson	Mid-West
L. B. F. Raycroft	New England
Mark Smith	Indiana
Charles S. Whitney	Detroit
Conrad H. Young	Washington

DAYTON SECTION

C. F. Kettering's address before the meeting of the Dayton Section, Oct. 3, was received with enthusiasm and a large audience gathered for the occasion. Mr. Kettering urged a more thorough appreciation of economics on the part of engineers. He discussed the automobile of the future and declared that it would be lighter, more economical and better sprung. Lubrication is one of the most important phases of

G. WALKER GILMER, JR.

ROBERT F. McCANN

J. S. CLAPPER

OTTO BURKHARDT

CHARLES S. WHITNEY

L. B. F. RAYCROFT

W. P. KENNEDY

NELSON B. NELSON

CONRAD H. YOUNG

MARK SMITH

H. E. FIGGIE

automotive engineering requiring study and engines must be designed to handle low-grade fuels.

The Dayton Section will not meet during November. P. S. Tice will address the Section on the Utilization of Low-Grade Fuels in Automotive Engines at its next meeting which is scheduled for Dec. 19.

New England Section

O. J. Rohde's paper on Wheels excited great interest at the October meeting of the New England Section because it revealed the conclusions that have led an established wire wheel company to enter the steel-disc wheel field. Mr. Rohde still considers the wire wheel the best type for passenger-car use, but he finds that the steel-disc wheel possesses the strength and flexibility qualities of the wire wheel at a greatly reduced production cost. Important improvements in wire wheel construction were described by the author, particularly the advancements made in spoke material and design. Methods of testing wire and steel wheels were shown with motion pictures. Mr. Rohde concluded that the wire wheel still possesses greater strength in combination with lighter weight than all other types of passenger-car wheel.

The next meeting of the New England Section will be held in Springfield, Mass., at the Hotel Kimball on the evening of Nov. 24. The meeting will be addressed by M. B. Speer, superintendent of the automotive service department of the Westinghouse Electric & Mfg. Co. Mr. Speer has been engaged in the automotive electrical field for many years and has specialized in the service end of the business. He has had a broad experience instructing repairmen and service-stations on the proper care and repair of automotive electrical apparatus. Electrical service men from all sections of New England are invited to attend this meeting.

Cleveland Section

The Cleveland Section had a very successful meeting, Oct. 20, when George M. Graham gave an excellent talk on the Relation of the Engineer to the Sales Department.

The great public interest displayed in the recent accomplishments of the German sail-planes has led the Cleveland Section to select this interesting means of flight as the topic for its meeting on Nov. 17. Dr. Georg H. Madelung will present the paper of the evening. Dr. Madelung has been in this Country about a year and was previously connected with the Technische Hochschule, Hannover, Germany, where he worked on the design of motorless gliders for the Rhoen contests. His acquaintance with the pilots and engineers who were active in the establishing of the various motorless flight records assures the presentation of an interesting paper. Dr. Madelung was at one time associated with Dr. Junkers and the Albatros Werke but is now a member of the Glenn L. Martin Co.'s engineering staff. The meeting will be held in the rooms of the Cleveland Engineering Societies, Hotel Winton, starting at 8 o'clock.

Detroit Section

The Detroit Section did not meet in October since all of its energy and interest was concentrated on the Production Meeting of the Society held in Detroit, Oct. 26 and 27 and described elsewhere in this issue.

The Section plans to continue the holding of meetings devoted to production matters and there is a possibility of its scheduling two meetings a month in the future, one on design and engineering, the other on production and shop problems. The next meeting of the Detroit Section is scheduled for Nov. 24, but the topic and the speaker have not been announced. All meetings are held at the Detroit Board of Commerce Building and start at 8 o'clock.

The Detroit Section office was moved Nov. 1 to 416 Capitol Theater Building. The new quarters are larger in size and permit the holding of larger committee meetings in the office. The location is central and readily accessible from all sections of the city. A complete library of Society publications is maintained in this office for the use of the members. The Assistant Secretary, Mrs. B. Brede, keeps in close touch with the New York City office and she is able to serve the members in their relations with the Society. Those members seeking employment will find the Detroit Section office of particular value in locating a new position.

Minneapolis Section

Four-wheel brakes were discussed at the October meeting of the Minneapolis Section. H. E. Morton described the features of the brake system bearing his name and outlined the general advantages of four-wheel brakes. He stated that front-wheel brakes are superior to rear-wheel brakes if either type is used singly. The front-wheel brakes effect a momentary transfer of 25 per cent of the weight of the car to the front axle when braking action takes place. Reasons underlying the lessened tendency to skid with brakes on all wheels were given by the author. Although four-wheel brakes stop a car in a much shorter distance than the conventional type, this particular advantage cannot be utilized in heavy traffic since vehicles in the rear of a car so equipped would collide with it if quick stops were made. The big advantage in this instance is the decreased wear and tear on tires and brakes with the four-wheel system which stops the car effectively without putting heavy loads on the brake bands, locking the wheels or sliding the tires.

N. S. Kingsley gave an illustrated talk on the origin of gas and crude oil, the treatment of the crude oil in stills and the refinery methods used to produce motor-fuels and lubricating oils.

The November meeting of the Section will be held at the Manufacturers Club, Nov 1 at 8 o'clock. George G. Bouthinon will present a paper on gray iron pistons.

Pennsylvania Section

The Pennsylvania Section met, Oct. 26 at the Engineers Club, Philadelphia, and discussed the production and design of steel passenger-car bodies. Motion pictures were shown depicting the manufacturing processes employed in the building of all-metal bodies.

Automobile Leaf Springs will be the topic at the meeting of the Pennsylvania Section on Nov. 23 The paper of the evening will be presented by Harold B. Winchell who is associated with William & Harvey Rowland, Inc., Philadelphia, manufacturer of automobile springs. Mr. Winchell has done considerable work on the specifications of springs for passenger-car use and has assisted in the development of a progress system of spring production. He will discuss spring action, the mechanics of spring design and stress the importance of careful selection of spring proportions in the early stages of design rather than experimenting to get a result on a finished car.

Washington Section

R. E. Carlson of the Bureau of Standards addressed the October meeting of the Washington Section. Mr. Carlson explained the operation of the apparatus developed by the Bureau for recording the performance data of an automobile while it is undergoing road test. This apparatus makes it possible to study the effect of fuel volatility changes on car performance under actual operating conditions. The particular apparatus described by Mr. Carlson is the one being used in the Society's Cooperative Fuel Research which is being carried on at the Bureau of Standards at the present time.

The date and program of the November meeting of the Section were not decided at the time The Journal went to press.

The Washington Section decided at its last meeting to secure the active interest of local automobile dealers and service-men in its meetings. Those who heard R. E. Carlson's talk agreed that it was intensely interesting to service-men and a committee was appointed to get more of them to attend the future meetings.

Indiana Section

One of the largest audiences ever assembled at an Indiana Section meeting gathered on the evening of Oct. 12 to hear an interesting talk by Thomas Midgley, Jr., of the General Motors Research Corporation. The attendance

reached nearly 300 and the Section officers appreciated the enthusiastic send-off for the coming winter program.

The Indiana Section meets, Nov. 9 to hear an address by Fred E. Moskovics, vice-president of the Nordyke & Marmon Co. Few men have studied automobile service with greater care than Mr. Moskovics and his experience and recommendations will deserve the attention of all engineers. The topic selected for this meeting is the Relation of Engineering to the Service Problem. Indiana Section meetings are held at the Athenæum, Michigan and New Jersey Streets, Indianapolis. They are preceded by a dinner at 6:30, the meetings starting at 8 o'clock.

Mid-West Section

C. F. Kettering was the guest of the Mid-West Section at a dinner held at the Engineers Club on Oct. 20. Over 100 members and guests were present and shared the enjoyment of Mr. Kettering's customary mixture of good humor and sound logic. He presented a very thorough analysis of the Fundamentals of Engineering.

The Section has been extremely fortunate in securing G. A. Green, formerly vice-president and general manager of the Fifth Avenue Coach Co., to address its November meeting. The excellent paper given by Mr. Green at the Summer Meeting of the Society aroused great interest in Chicago, where there is an unusual opportunity for the expansion of motorbus transportation. The municipal authorities concerned with Chicago's transportation media will be present at the meeting; many electric railway men are also being invited. This meeting will be held Nov. 17 in the rooms of the Western Society of Engineers, Monadnock Building, starting at 7:30 p. m.

Buffalo Section

L. H. Pomeroy gave a very comprehensive paper on the fundamentals underlying light-weight design at the meeting of the Buffalo Section, Oct. 19. Mr. Pomeroy discussed the conditions determining chassis weight in general as they are related to bearings sizes and their supports. The extent to which bearing loads in engines are a function of inertia rather than gas pressure was studied in the paper. Constructional materials were considered in terms of their specific strength and the effect of forged aluminum connecting-rods on engine performance and efficiency was shown. The paper contained many valuable data for the use of engineers who may be considering the more extensive use of aluminum alloys in automotive engines and chassis.

The next meeting of the Buffalo Section is announced for the evening of Nov. 17. C. P. Grimes, research engineer, H. H. Franklin Mfg. Co., will present a paper on the problems encountered in the design and development of air-cooled automobile engines.

Metropolitan Section

William B. Stout read a paper on the Modern Airplane at the October meeting of the Metropolitan Section and told the members that he believed the commercial airplane of the future will be of the all-metal monoplane type; the inability of wood veneer and linen to withstand severe service in inclement weather will drive them both out of commercial plane structure. The monoplane with internally-trussed wings of thick section lends itself admirably to all-metal construction. Thick wings can be selected for high-lift characteristics and the elimination of struts and cables reduces the head-resistance to a minimum. Mr. Stout exhibited samples of some very stiff rolled duralumin sections that he had developed and described a large all-metal torpedo plane in which these were employed. He believed that relatively low-power engines would be used in the commercial airplanes of the future because of their lower first cost.

The Metropolitan Section will be addressed by A. L. McMurtry at its meeting on Nov. 16. Mr. McMurtry was formerly engineer of the Connecticut State Motor Vehicle Commission and has been largely responsible for the constructive highway legislation program of that State. His experience in highway traffic regulation is very broad and motor-truck operators and fleet owners will find his talk particularly valuable to them. Highway engineers and members of city planning commissions will be invited to attend the meeting and discuss Mr. McMurtry's paper. The meeting will be held at the Automobile Club of America at 8 o'clock, being preceded by the customary dinner at 6:30.

IMPACT TEST

THE impact test reveals the amount of work that must be done on a given specimen to produce rupture or a certain deformation. In the usual tension-test, if the elastic-limit is determined the engineer can safely apply this result in figuring the dimensions of any member of a structure. In the case of the impact test, the results cannot be so applied. The work required to break any given section cannot be calculated from the work required to fracture a test-bar of different dimensions. This does not, however, destroy the value of the impact test. Its comparative value is still vital in passing upon the properties of a given material for certain specific purposes.

The above statements are based upon results obtained from notched test-bars broken as beams, supported, as the case may be, at one or both ends. It may not represent conditions for notched tension-tests, but sufficient data are not yet available to make a definite statement regarding this point.

How then can the results of the impact test be interpreted in engineering practice? This can probably best be answered by the following illustration. No component of a gun, such as the jacket or tube, which has ruptured in service from a cause that cannot definitely be assigned to defective ammunition, excessive power pressure, etc., has shown an impact value exceeding 6 ft-lb. on specimens tested in a small Charpy impact-machine. It has therefore become the practice of Watertown Arsenal to take impact tests from all components of a gun and, regardless of their ordinary tensile properties and the acceptability of the forging as judged from this test, no piece is accepted unless the results on the small Charpy impact-machine show a value of at least 6 ft-lb. To have a factor of safety, no major forging would be accepted unless this result was at least 10 ft-lb. It is believed that for any class of service a careful study of failures, together with material that is satisfactory, will enable the determination of a critical impact-value for a specific service.

Although not necessarily pertinent to the subject under discussion, it is undoubtedly a fact that many metallurgists and engineers question the value of the impact test on either notched or unnotched specimens. It has been the experience of the laboratories at the Watertown Arsenal that the Charpy impact-test has materially assisted in the diagnosis of many failures of parts in service and has made possible the prevention of their recurrence when applied to the testing of replacement parts. Although no definite relationships exist between the static tensile-properties as conventionally used and the results of the Charpy tests, this in itself cannot condemn the Charpy test or any other form of impact test on either notched or unnotched bars. Many specific cases could be cited where the impact test has been of the utmost practical value, and in view of this condition it is not believed that condemnation of the test is justified.—From a paper by F. C. Langenberg and N. Richardson, presented at meeting of the American Society for Testing Materials.

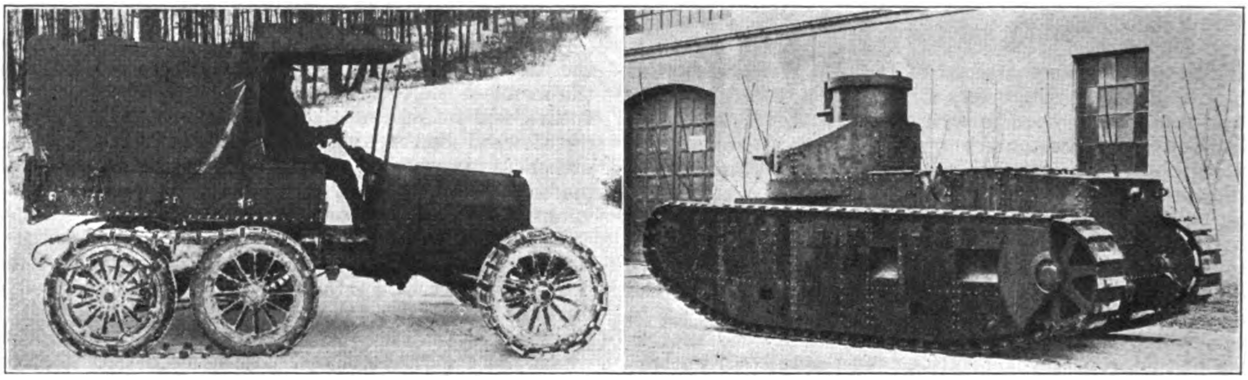
Aberdeen Proving Ground Visit

THE members of the Society, through the courtesy of Gen. C. C. Williams, chief of ordnance, again enjoyed the privilege of visiting Aberdeen Proving Ground, Md., where a program of test firings and demonstrations of post-war ordnance material was held on Oct. 6. The arrangements for this year's meeting were similar to those made for 1921, and this, without detracting from the interest of the exhibit, served to emphasize the progress made during the year. Thus, firing the 16-in. gun was again the first event, but the new feature, the recently perfected solenoid chronograph used for determining the velocity of the projectile, excited as much comment as the more spectacular shooting of the gun. Similarly, the railroad mount for 14-in. guns and 12-in. howitzers was shown in a design allowing of greater mobility during emplacement than was possible last year. The distinctly new developments shown for the first time were the supersensitive fuzes and the nitrotite containers; the former serve to explode anti-aircraft projectiles by impact against so slight an obstacle as two thicknesses of airplane silk, while the latter protect explosive charges, even when completely immersed in water, against the effects of moisture.

The automotive apparatus circus, which so pleased the Society members last year, was staged once more. Among its novelties this year were the medium A tank described below, a radio-directed whippet tank and the Mark VIII tank with a revolving observation tower operating on the stroboscopic principle. The "amphibious" tank, however, omitted its amusing performance of last year.

In the general run of automotive development in the Ordnance Department there are but few features which, taken singly, are not of interest to the commercial designer and builder of trucks and tractors. For example, the medium A tank, illustrated on this page, has a new type of track suspension, and the construction of the track shoes proper employs novel means to secure more efficient lubrication of the track pins than has been the practice in previous types of track-laying vehicles. Another instance, also illustrated herewith, is the Dodge truck to which has been added a fabric track. The purpose of the track application to a commercial vehicle is to provide increased mobility in military service. It is hoped that all general cargo work for supplying an army in the field will be accomplished in the future by the use of this or a similar type of vehicle. In case this development is ultimately successful, there will be a field for its general use on the farm and for road transport in those sections of the country where there are few or practically no improved roads.

Other exhibits of interest to the automotive engineer were much mobile artillery material, including the Christie mount for 4.7-in. guns, the dirigible D-4, the Owl bombing airplane and the foreign army trucks, tractors, mounts and tanks on display in the Museum. This array of automotive devices makes an impressive demonstration of the important problems for which the Ordnance Department sees an eventual solution in the simplicity and compactness of the gasoline engine as a powerplant.



TWO PIECES OF AUTOMOTIVE ORDNANCE MATERIAL THAT WERE SHOWN TO THE MEMBERS ON THEIR VISIT TO THE ABERDEEN PROVING GROUND ON OCT. 6

At the Left Is a Dodge Truck Equipped with a Fabric Track That Is Expected To Provide Increased Mobility in Military Service. The Tank at the Right Is Characterized by a New Type of Track Suspension and the Use of Novel Means in the Construction of the Track Shoes To Secure More Efficient Lubrication of the Track Pins

WORK OF AMERICAN ENGINEERING STANDARDS COMMITTEE

THE growing interest in standardization on the part of almost every American industry is emphasized by the quarterly report of the activities of the American Engineering Standards Committee. Of the projects that have official status before the committee 20 are concerned with mechanical engineering; 17 are civil engineering projects; 15 are electrical; 3 are automotive; 10 are concerned with transportation; 10 with ferrous metals; 11 with chemical; 5 with non-ferrous metals; 4 with mining; 2 with textiles; 1 with shipbuilding; and 8 projects are of general interest.

Twenty-four standards or safety codes have been approved and 36 are up for approval. The remaining 46 projects represent codes and standards that are in the process of formulation or are now being considered by committees of representatives, designated by the various bodies, industrial, technical and governmental, interested in each particular subject. In this way more than 200 such bodies are officially participating in the work of the American Engineering Standards Committee through their accredited representatives.

A regular interchange of information as to the status of work under way is maintained by the American Engineering Standards Committee with the national standardizing bodies of Austria, Belgium, Canada, Czecho-Slovakia, France, Germany, Great Britain, Holland, Italy, Japan, Norway, Sweden and Switzerland.

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Sept. 9 and Oct. 10, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ANDERSON, IVAN L. (E S) student, University of Utah, *Salt Lake City, Utah,* (mail) 1859 Lake Street.

BARROW, ARTHUR THOMAS (A) general manager, City Motor Works, Ltd., Kent Terrace, *Wellington, New Zealand.*

BASCH, JACOB JUSTIN (J) sales engineer, G & O Mfg. Co., New Haven, Conn., (mail) 828 North Broad Street, *Philadelphia.*

BIDDLE, CHARLES JONATHAN (E S) student, California Institute of Technology, Pasadena, Cal., (mail) 2815 Stuart Street, *Berkeley, Cal.*

BITTERMAN, SIMON (A) president and manager, American Auto Products Co., 240 South Broadway, *Denver, Col.*

BOSCHAN, ALEXANDER S. (E S) student, University of Michigan, Ann Arbor, Mich., (mail) 213 First Street, *Keyport, N. J.*

BURN, WALTER P. (A) sales promotion and advertising manager, Transcontinental Oil Co., 510 Benedum-Trees Building, *Pittsburgh.*

BURT, LEO O. (M) designer, Chevrolet Motor Co., *Detroit,* (mail) 1732 West Bethune Avenue.

CHASE, RUSSELL D. (A) P. O. Box 163, *Summit, Cook County, Ill.*

CHESEBRO, C. M. (M) chief engineer, Detroit Motor Co., *Washington, Pa.*

COBB, CARROL J. (E S) student, Ohio State University, *Columbus, Ohio,* (mail) 45 Crestview Road.

COFFIN, O. W. L. (A) branch manager, White Co., 1715 Preston Avenue, *Houston, Tex.*

COX, CAPT. M. R. (S M) U. S. Army, *Fort Sill, Okla.*

CROSS, CHARLES H. (E S) student, Ohio State University, Columbus, Ohio, (mail) 3210 Arthington Street, *Chicago.*

DAVISON, WALTER W. (M) factory manager, Wire Wheel Corporation of America, Inc., 1700 Elmwood Avenue, *Buffalo, N. Y.*

FOGELSON, EMIL (M) mechanical engineer, Victor Pagé Motors Corporation, *New York City,* (mail) 1086 Kelly Street.

GOMMEL, D. E. (J) chief engineer, Monroe Automobile Co., *Indianapolis, Ind.,* (mail) 2905 Meredith Street.

GORDON, JOHN RUTHERFORD (F M) body builder, Holdens Motor Body Builders, *Adelaide, Australia.*

GROVES, FRANK A. (A) president and general manager, Stemco Engineering Co., Second and Webb Streets, *Dayton, Ohio.*

HERMANN, JOHN F. (A) assistant engineer, Phelps Light & Power Co., Rock Island, Ill., (mail) 614 Seventh Avenue, *Clinton, Iowa.*

HIRSH, DAVID H. (J) instructor, Greer College of Automotive Engineering, 2024 South Wabash Avenue, *Chicago.*

HORTHY, WILLIAM A. (M) mechanical engineer, tractor works, International Harvester Co., *Chicago,* (mail) 4828 Dorchester Avenue.

HOWER, HARRY S. (M) director of research, Macbeth-Evans Glass Co., *Pittsburgh.*

HUGHES, F. J. (A) technical representative, Tilling-Stevens Motors, Ltd., *Maidstone, England* (mail) 60 Holland Road.

KARLSON, K. EDWIN M. (M) factory manager, Cleveland Automobile Co., *Cleveland.*

KENNEDY, JAMES T. (A) sales department, Goodyear Tire & Rubber Co., 2817 East Grand Boulevard, *Detroit.*

KNOWLTON, H. B. (M) instructor of metallography heat-treating, Central Continuation School, Seventh and Prairie Streets, *Milwaukee.*

KOHR, ROLAND MEREDITH (E S) student, Ohio State University, Columbus, Ohio, (mail) *New Philadelphia, Ohio.*

MENEWISCH, WILLIAM T. (J) draftsman, Fox Motor Car Co., *Philadelphia,* (mail) 1332 South Divinity Place.

MILLER, HENRY J. (J) draftsman, Zeder-Skelton-Breer Engineering Co., Newark, N. J., (mail) 178 Montgomery Street, *Bloomfield, N. J.*

MOHORI, TERUO (A) 13816 Argus Avenue, *Cleveland.*

MORTIMER, BRUCE G. (A) assistant superintendent, Wilson Motor Sales Co., *Toronto, Ont., Can.,* (mail) 1504 Dufferin Street.

PARISH & BINGHAM CORPORATION (Aff) *Cleveland.*
Representatives:
Conrad, F. K., secretary and treasurer.
Fitch, R. G., sales manager.
Harmon, O. B., assistant chief engineer.
McMunn, William N., vice-president.
Maloney, J. E., assistant sales manager.
Morse, Agnes D., president.

PLUMRIDGE, TOM G. (A) consulting engineer, American Technical Society, *Chicago,* (mail) 4455 Ellis Avenue.

ROEMMELE, HOWARD CARL (E S) student, Stevens Institute of Technology, Hoboken, N. J., (mail) 31 Astor Street, *Newark, N. J.*

SALISBURY, C. E. (M) manager, service department, Hupp Motor Car Corporation, *Detroit.*

SCHENCK, R. B. (M) metallurgical engineer, Buick Motor Co., *Flint, Mich.,* (mail) 708 Clifford Street.

SCHNEIDER, HAROLD P. (E S) student, Ohio State University, Columbus, Ohio, (mail) 123 14th Street, *Toledo.*

SCULLY, JAMES N. (J) president, Houdaille Co., *Buffalo,* (mail) 401 Delaware Avenue.

SEILER, PAUL W. (A) president and general manager, Ternstedt Mfg. Co., *Detroit,* (mail) Box 6, West Fort Station.

SKINNER CO., LTD. (Aff) *Gananoque, Canada.*
Representative: Skinner, Frederick J., president and general manager.

SOWER, GEORGE W. (E S) student, Ohio State University, Columbus, Ohio, (mail) 11721 Oakview Avenue, *Cleveland.*

STUTTER, CLINTON L. (J) assistant foreman, J. Linek, 6 Washington Street, *Maspeth, N. Y.*

SUN CO. (Aff) Finance Building, 1428 South Penn. Square, *Philadelphia.*
Representatives:
Buderus, W. H.
Cox, C. P., manager motor oil department
Drysdale, R. L., cutting oil engineer
Eckert, S. B., district sales manager
Pew, J. N., Jr., vice-president

TALL, G. W. (A) sales manager, electric furnace division, Leeds & Northrup Co., 4901 Stenton Avenue, *Philadelphia.*

TEETOR, LOTHAIR (A) sales manager, Indiana Piston Ring Co., *Hagerstown, Ind.*

TETENS, RAYMOND E. (M) assistant chief engineer, Earl Motors, Inc., *Jackson, Mich.*

VOORHEES, R. C. (A) R. F. D. No. 3, *Ypsilanti, Mich.*

WILSON, EDWARD ARTHUR (E S) student, California Institute of Technology, *Pasadena, Cal.,* (mail) 192 South Cypress Street, *Orange, Cal.*

Applicants *for* Membership

The applications for membership received between Sept. 15 and Oct. 16, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

BACON, DAY H., designer, Ansted Engineering Co., *Connersville, Ind.*

BISHOP, CHARLES ALBERT, designer, Glenn L. Martin Co., *Cleveland.*

BAYLESS, RAY T., chemical engineer and metallurgist, American Society for Steel Treating, *Cleveland.*

BROOKE, HAROLD LEE, in charge of dynamometer experimental department, Maxwell Corporation, *Highland Park, Mich.*

CALLUM, JOHN R., designer and builder of racing cars, J. R. Callum & Co., Inc., *Norfolk, Va.*

CARNEY, ROY C., service superintendent, Federal Motor Truck Co., *Chicago.*

CAUTLEY, RANDOLPH, investigator of costs and special manufacturing problems, Wright Aeronautical Corporation, Paterson, N. J.

CERVENY, FRANK, instructor, Janesville High School, *Janesville, Wis.*

CHERNIACK, NATHAN, assistant transportation engineer, Ward Motor Vehicle Co., *Mount Vernon, N. Y.*

CHRISTENSEN, E. G., sales manager, Carl Pick Co., *West Bend, Ind.*

COLWELL, DONALD L., chief chemist and metallurgist, Stewart Mfg. Corporation, *Chicago.*

COOKE, B. W., president, Cooke Auto School and Coyne Trade and Engineering School, *Chicago.*

D'ANNUNZIO, UGO V., president, Isotta Motors, Inc., *New York City.*

DEAN, KENNETH G., engineer, Lavine Gear Co., *Milwaukee.*

DELANEY, GEORGE A., engineer, Paige-Detroit Motor Car Co., *Detroit.*

DOLL, GUS. P., treasurer and director of sales, Thomas J. Corcoran Lamp Co., Cincinnati.

EHRKE, MALCOLM N., draftsman, Western Automatic Machine Screw Co., *Elyria, Ohio.*

ELLIS, CAPT. CARROLL L., *Aberdeen Proving Ground, Md.*

FERRIS, WALTER, chief engineer, Oilgear Co., *Milwaukee.*

GREENWALD, H. A., engineer, Cadillac Motor Car Co., *Detroit.*

GUBITZ, WERNER, body designer, Locomobile Co., *New York City.*

HAMPSHIRE, GEORGE W., district manager, Colonial Steel Co., *Pittsburgh.*

HELLINGS, S. A., vice-president and sales manager, Stewart Mfg. Co., *Chicago.*

HOPKINS, FRED. J., designer, International Harvester Co. of America, *Chicago.*

HOPKINS, LUTHER H., maintenance foreman, tractor works, International Harvester Co., *Chicago.*

HUNT, SAMUEL J., manager and lubrication engineer, J. D. Streett & Co., *St. Louis.*

IRVIN, W. A., assistant to vice-president, American Sheet & Tin Plate Co., *Pittsburgh.*

JANSON, OLOF, draftsman, Advance-Rumely Co., *Laporte, Ind.*

KERSTEN, A. EDGAR, service manager, Garford Motor Truck Co., *Boston.*

LEISTER, FAYETTE, sales engineer, Fafnir Bearing Co., *New Britain, Conn.*

LILLIBRIDGE, BYRON J., JR., B. F. Sturtevant Co., *Hyde Park, Mass.*

LIU, CALVIN Y., student, University of Michigan, *Ann Arbor, Mich.*

MACGREGOR, B. N., cable sales, Packard Electric Co., *Warren, Ohio.*

MALCHOW, HENRY J., division engineer, Standard Oil Co. (Indiana), *Milwaukee.*

MONK, MARVIN E., assistant sales manager, U. S. Ball Bearing Mfg. Co., *Chicago.*

PICKARD, THOMAS, inspector, Moon Motor Co., *St. Louis.*

ROBERTS, LOUIS L., experimental engineer, C. H. Wills & Co., *Marysville, Mich.*

ROOKS, ALFRED W., maintenance engineer, Chevrolet Motor Co., *Detroit.*

SHIRASAWA, GEN., engineer, Japan Automobile Co., Akasaka, *Tokio, Japan.*

SMECKEL, ERIC F., assistant chief engineer, Walker Axle Co., *Chicago.*

SMITH, HAROLD E., model maker, Remy Electric Co., *Anderson, Ind.*

TAMPLIN, WILLIAM G., special representative, American Sheet & Tin Plate Co., *Pittsburgh.*

THOMSON, ROBERT, chief draftsman, Ainsworth Mfg. Co., *Detroit.*

THORSON, WILBUR R., assistant manager, Central Iowa Electric Co., *McCallsburg, Iowa.*

TIMMES, JOHN A., office manager, Studebaker Corporation of America, *New York City.*

WANG, CHENG FU, draftsman, Glenn L. Martin Co., *Cleveland.*

WARFORD, HENRY W., student, Tri-State College of Engineering, *Angola, Ind.*

WELCHANS, EDWARD, chief engineer, J & B Mfg. Co., *Pittsfield, Mass.*

WHITE, KARL H., aeronautical engineer, Aeromarine Plane & Motor Co., *Keyport, N. J.*

WOOD, GEORGE A., factory manager, Mutual Wheel Co., *Moline, Ill.*

ZOBEL, CARL G. F., laboratory assistant, Bureau of Standards, *City of Washington.*

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XI December, 1922 No. 6

Chronicle and Comment

The Chicago Service Meeting

THE annual Chicago Meeting of the Society will be held Jan. 31, 1923, during Show Week. Automotive service will be the dominant topic at the technical sessions and the dinner. This meeting will be especially valuable to factory and dealer service-managers and operators of large fleets of vehicles. Further details will be found on p. 552.

The Production Men

THERE are many members of the Society who should be "past masters" in production. There are wonderful brains behind some of the factories that are in production in a large way. These men have very superior knowledge. I hope that the best production men in the United States will tell the Society what should be done along production lines.—Charles W. Nash.

Standardization

STANDARDIZATION by one firm to suit its particular purposes and convenience is a simple affair and a quick process. Industrial standardization requires that the needs of industry be unified with the least possible disturbance. Recommendations and conclusions must be sound and economical and conform generally to the practice of the manufacturer, as well as meet the requirements of the buyer. Otherwise nothing but failure can result.—D. R. MacDonald.

The Annual Meeting

THE Annual Meeting of the Society will be held in New York City, Jan. 9 to 12, 1923, during the week of the National Automobile Show. A number of technical sessions, the Annual Business Meeting and the Standards Committee Meeting will be conducted during this period in the Engineering Societies Building. The Annual Dinner has been set for the evening of Jan. 11 at the Hotel Pennsylvania. Tentative details of the program and a dinner reservation blank are given on p. 551. A *Meetings Bulletin* will reach the members about Dec. 15 with the complete and final announcement.

Future Standardization Work

ON p. 555 of this issue will be found a resume of the work of the Standards Committee for 1922 and a list of subjects that will be considered during the coming year. In the majority of instances Subdivisions are appointed to formulate tentative recommendations covering subjects assigned to the various Divisions of the Standards Committee. Non-members of the Standards Committee, as well as non-members of the Society, are serving on these Subdivisions as it is the purpose of the Society to have all interests represented in the standardization work carried on by it for the automotive industries. Anyone interested in particular subjects is at all times invited to communicate with the Subdivisions through the Society in order that assistance may be rendered while the work is in the formative period.

Oil-Pumping

THERE are many contributory causes of the service trouble familiarly known as oil-pumping. Any one, several or all of them may result in excessive passage of oil to the combustion-chamber. Piston design and clearance, ring width and form, crankshaft oil-hole location, oil-pressure intensity and many other factors affecting this bugbear of the service-man are discussed in this issue of THE JOURNAL. The volume of experience related by the several engineers is evidence of a general and diligent study of the problem. Numerous remedies are suggested but it is noteworthy that no single solution is agreed upon. Apparently oil-pumping can be forestalled only by careful elimination of all the contributory causes, by a combination of refined design, selected clearances and good workmanship. The effect of wear can be anticipated in the original design so that conditions causing oil-pumping will not develop or can be easily controlled in service. Read the matter on p. 491.

Potential Profits

ON p. 529 of this issue will be found the various recommendations of the Divisions of the Standards Committee that will be acted upon at the Annual Meeting in January. Each of these recommendations represents a potential saving or profit that can be realized only through the actual adoption of it in future automotive practice.

There is no question that a standard which meets the approval of the qualified engineers of the industries affected will be adopted as changes in design and production permit.

Each Society member should therefore see that the recommendations of the Divisions are referred to the engi-

neering and production departments of his company in order that any desirable changes in the recommendations may be considered at the Annual Meeting. The responsibility of approving these recommendations rests with every qualified member of the Society who should, if possible, attend the meeting of the Standards Committee on Jan. 9, and participate in the discussion, or else submit comments in writing.

Only a small amount of time and expense is thus required of an individual member to make possible a potential saving of hundreds of thousands of dollars.

1923 Roster

A LARGE volume of detail work is involved in the preparation annually of the Membership Roster. The first step is the recording of changes of position and address of members. The members are urged strongly to return promptly, properly filled-in, the blanks that have been sent them this month, whether or not their address or position has changed recently. To produce a satisfactory printed roster, adequate cooperation of the members is essential.

The classification printed on the back of the blank sent to the members is an important feature, and every member is earnestly asked to supply thereon the information called for. About one-half of the members have returned similar blanks sent out in September. No additional classification forms will be mailed to the members prior to the issuance of the 1923 Roster.

It has been directed by the Council that the Roster be sent to only those members who request copies of it in advance of printing. A very small surplus stock is carried at the Society offices, and a charge of 50 cents per copy will be made for Rosters ordered by members after Feb. 15. Such orders will be filled until the supply is exhausted.

It is expected that the 1923 Roster will be issued early next spring.

S. A. E. Employment Service

THE scope of the Employment Service mailing list has been extended to include approximately 1000 employers engaged in different fields of the automotive industry, including those of the passenger-car, motor-truck, bus, tractor, motorcycle, cab, aircraft, motorboat, engine, body-building, and miscellaneous parts. Better results are being obtained in the service at less expense.

By the use of a selector addressing-machine it is possible to address quickly Men Available notices to prospective employers in any one or more of the fields mentioned. When a member registers for employment he is requested to designate the types of employer to whom he wishes notice of his availability sent. The bulletin in which his notice is printed is sent only to those in the classes specified by him. Also, the list of men available has been classified so that, when there is an opening in any given line of work, only those members who are apparently qualified to fill it are notified.

During the last half-year the Society has, it is definitely known, been instrumental in securing for 86 members positions as draftsmen, engineers, chief engineers, body designers, factory managers, sales managers, salesmen, production managers and general managers. It is believed that many others have secured positions either directly or indirectly through the S.A.E. Employment Service.

Aluminum Connecting-Rods

THE aluminum-alloy piston is rapidly gaining ground as standard equipment in passenger-car engines. It is apparent that the serious troubles encountered with the first aluminum pistons have been overcome. Radical changes have been effected in piston design; the alloys have reached a higher stage of development; special heat-treatment is producing a hardness approaching that of cast iron. This persistent development, in the face of the condemnatory experience of the pioneer users, has been stimulated by one fundamental consideration, namely, the inherent advantage of decreasing reciprocating weight in high-speed engines. The success achieved by the aluminum piston and the resulting improvement in the performance of engines equipped with it have led logically to the production of lighter connecting-rods.

One or two car companies are now using aluminum-alloy connecting-rods in their production models. The commercial practicability of forged aluminum connecting-rods is being studied to-day by many engine designers. The current interest in this latest progression of automotive design lends significance to the publication of the paper by L. H. Pomeroy on page 508 of this issue of THE JOURNAL. In it will be found a careful treatment of the fundamental reasons underlying the advantages of the forged-aluminum connecting-rod. Valuable recommendations and data relating to design are presented.

S. A. E. Handbooks

MEMBERS have probably found that since the insertion of the August 1922 issue of data sheets, their S.A.E. HANDBOOKS are rather larger than is convenient for use with 1-in.-ring leather binders, for the August issue has brought the total number of data sheets up to 188 which is really more than can be conveniently used in the standard binder.

If members desire to break their S.A.E. HANDBOOKS into two parts, this can be done readily by transferring Sections D to K inclusive to another binder, the first binder then being marked, "Vol. I, Sections A to C," and the second, "Vol. I, Sections D to K." Additional binders stamped in this way can be obtained from the Society at $4 each. If the first binder used is returned when the second is ordered, the letters indicating the Sections it is to contain will be stamped on it.

It should be borne in mind also that the Contents are arranged so as to aid members in limiting their S.A.E. HANDBOOK to only those pages that are of particular interest to them respectively, the standards being classified according to the automotive industries to which they apply by a system of key letters. Members interested, for instance, in only tractor engineering may want to retain in their books only those standards which are followed by the letter *T* in the Contents. The standards in the S.A.E. HANDBOOK may therefore be considered as being classified in the Contents in two ways; by parts and materials, such as powerplant and electrical equipment, the classifications being denoted by letters prefixed to the page numbers; and by automotive industries, the industries being denoted by letters appearing in the right-hand column in the Contents.

Vol. XI December, 1922 No. 6

Ford Engine-Cylinder Production

By P. E. Haglund[1] and I. B. Scofield[1]

Detroit Production Meeting Paper *Illustrated with* Photographs and Diagrams

The authors state the principles governing intensive quantity-production and describe the sources and methods of handling the basic materials that compose the Ford engine-cylinder. The fundamental plan of the River Rouge plant is outlined, illustrations being used to supplement the text that explains the reasons governing the location of the various units of the plant. Details are given of the use made of conveyors with the idea of keeping everything moving.

The relation of the blast furnace and coke ovens to the engine cylinder are commented upon, the power-house and foundry are described, and the production of the cylinder is set forth step by step.

The Ford Motor Co. began making its own cylinder-block in 1907, accepting the men and methods of the day. Its foundry at that time was located at Romeo, Mich., 60 to 70 miles north of Detroit. The design of the cylinder-block has changed somewhat from the one then produced, but it will suffice, for a rough comparison, in contrasting the results of the old methods with those resulting from a greatly increased production.

In 1908, we cast 50 cylinders per day; now we are producing 8000 cylinders in 16 hr. This greatly increased production is possible only with modern methods in which the conveyor plays the most important part. If you are in the least acquainted with foundry work, try to imagine what it would mean to cast 8000 cylinders on the floor by the old methods. This would seem nearly impossible, especially from the standpoint of the iron-handling problem. It would require acres of ground, the labor cost would be multiplied four or five times and working conditions would be unbearable because of the intense smoke and gas escaping from the molds. This immense foundry production represents the ultimate result of applying Mr. Ford's ideal of producing at the minimum cost.

To market cars in such unusual quantities, it was evident that automobiles had to be produced at a lower figure. This could not be done by lowering the men's wages and driving them, but had to be done by improving manufacturing methods. With this idea in mind, every economy was effected. First, the work and the material were brought directly to the men, thus minimizing wasteful transportation of material from one part of the shop to another. This was a severe blow at high labor cost, which is the principal item in manufacturing costs.

The next step, and the most difficult one, was to decrease the cost of the materials used. It demanded that the manufacturer have control over the raw materials from nature's source of supply to the finished product. Although this could not be accomplished at once, progress has been made in the right direction.

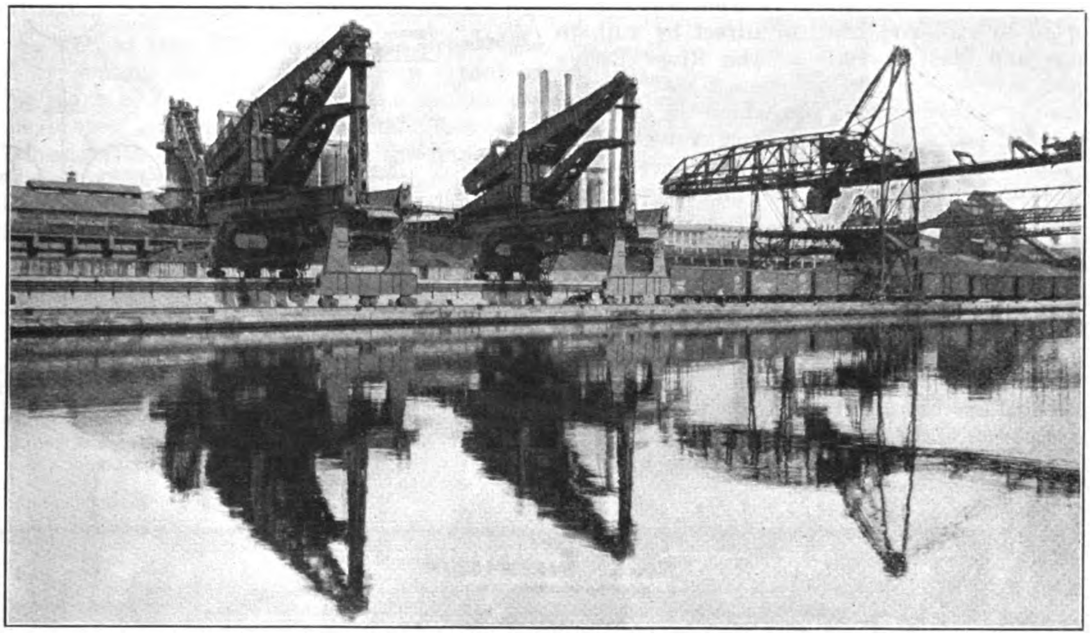


Fig. 1—Dock at Which the Ore, Coal and Limestone Required in the Production of the Ford Engine Cylinders Are Unloaded

Basic Materials

The natural sources from which the automobile is derived are principally iron ore and coal. The engine cylinder is roughly 95.0 per cent iron and 3.5 per cent carbon from the coal. The remaining elements of its composition are also derived from the ore, with the exception of possibly some sulphur from the coke used.

As yet, the Ford company does not mine all its iron ore, but a good portion of it comes from the Ford-Imperial mine, in the upper peninsula of Michigan. It is loaded into railroad cars, transported to a Lake port and

[1]Production department, Ford Motor Co., Detroit.

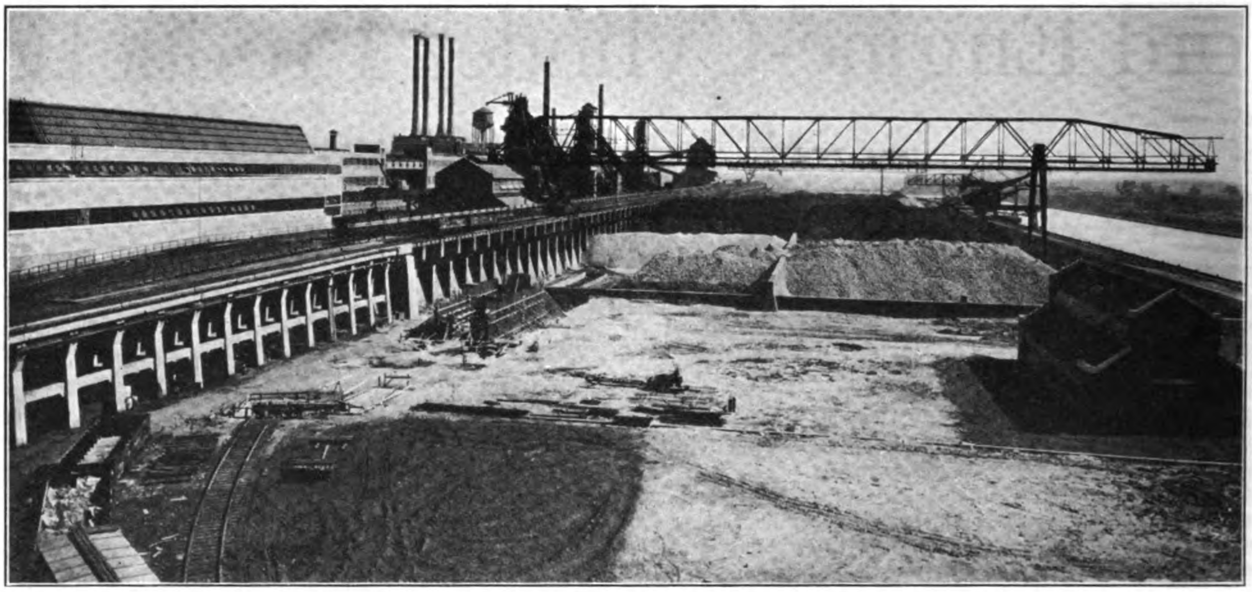


FIG. 2—THE RAW-MATERIAL STORAGE BINS HAVE A CAPACITY OF 2,000,000 TONS AND ARE SERVED BY TWO 550-FT. TRANSFER BRIDGES

conveyed directly to the blast furnace by a water route, the most economical means of transportation at present. The coal is mined in the Ford mines in Kentucky. It is then transported by rail and boat, or direct by rail, to the coke ovens and blast furnace at the River Rouge plant.

Fig. 1 shows the unloading docks to which the ore, coal and limestone-laden boats are moored upon arrival at the River Rouge plant. Two Hulett ore-unloaders are shown at the left, and a Mead-Morrison coal-unloader at the right. These huge machines transfer the raw materials direct to the storage bins behind them. The bins, illustrated in Fig. 2, have a storage capacity of 2,000,000 tons, and are traversed by two transfer bridges each 550 ft. long. These transfer bridges, one of which is shown clearly in Fig. 2, transport to the various plant units coal, ore and limestone as they are needed.

Both ore and coal are brought to their destination with the minimum amount of loading, reloading or handling. This system also effects a considerable economy by elim-

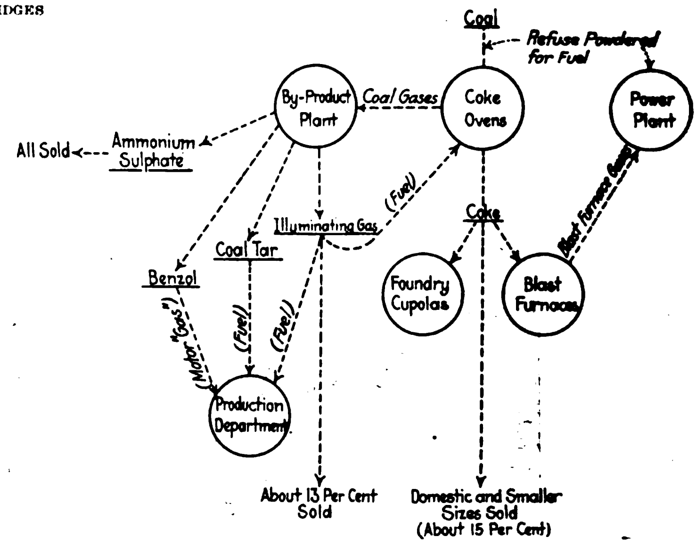


FIG. 4—DIAGRAM SHOWING HOW COMPLETELY COAL AND ITS BY-PRODUCTS ARE UTILIZED

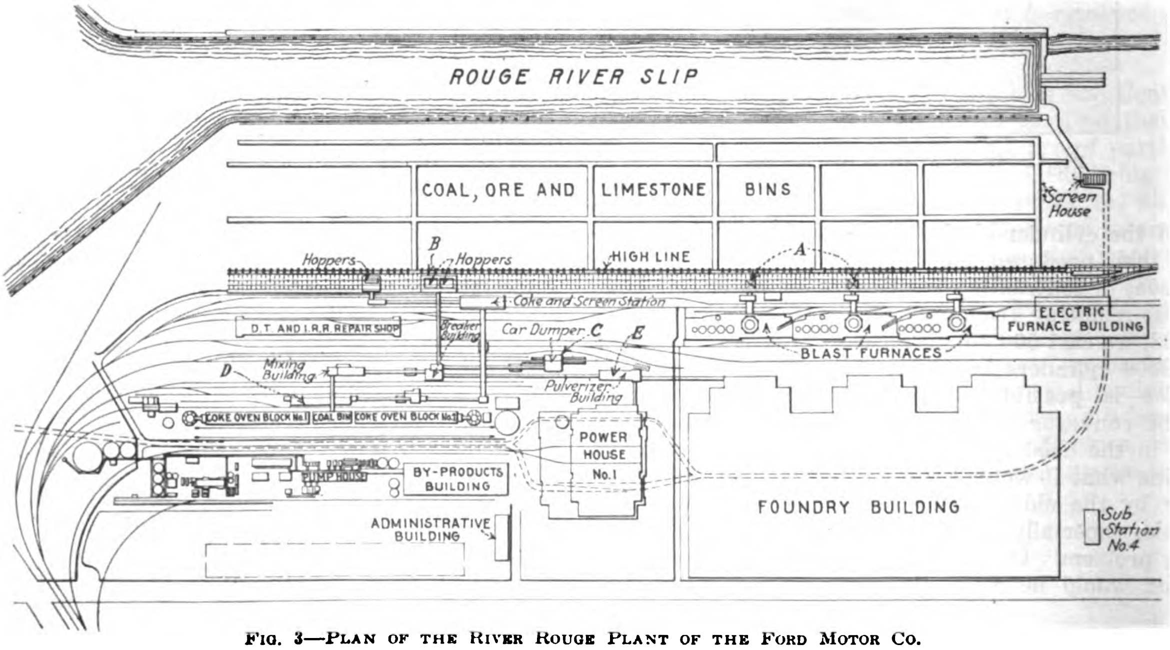


FIG. 3—PLAN OF THE RIVER ROUGE PLANT OF THE FORD MOTOR CO.

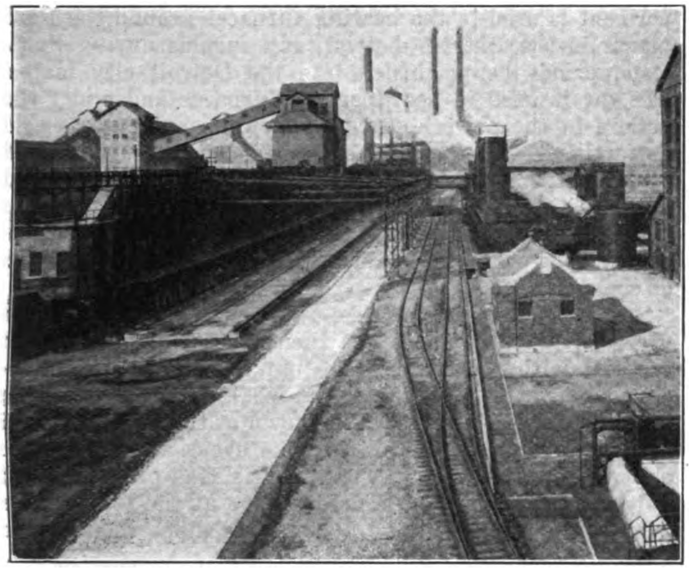


FIG. 5—A GENERAL VIEW OF THE COKE OVENS AND THE CHEMICAL PLANT

inating the profits of dealers, brokers and middlemen, one of our great present-day economic wastes.

BLAST FURNACE AND COKE OVENS

It might be well at this point to describe briefly the fundamental or general plan upon which the layout of this immense plant is based, indicating the results attained by the proper grouping of the units. The basic idea is to connect, with conveyors, each of the important units of the manufacturing plant, thus eliminating rail or truck transportation from one unit to another. To do this the units had to be built as close together as possible.

Fig. 3 shows clearly the relation of the units described in this paper. Iron ore and limestone are transferred to the blast-furnace skip-cars at *A*, where they are weighed and loaded directly into the furnace. Coal is transferred from the storage bins to the hoppers *B* and thence by mechanical conveyor through the breaker and mixer buildings to the coke-oven coal-bin. Coal arriving by rail is unloaded by the car dumper at *C* and conveyed through the same breaker and mixer buildings to the coke-oven coal-bins. The coke is mechanically conveyed from the coke bin *D* to the coke and screen station *E*, there it is graded and distributed by mechanical conveyor to the blast furnace, to the foundry and to storage. The finer coal is pulverized in building *E* and piped direct to the powerhouse.

The powerhouse is located in the center of all of the important units; namely, the coke ovens, blast furnaces and foundry. This is done for the double purpose of placing the boilers close to the sources of by-product heat-energy, and locating the electric generators near the units consuming electric current. The arrangement of

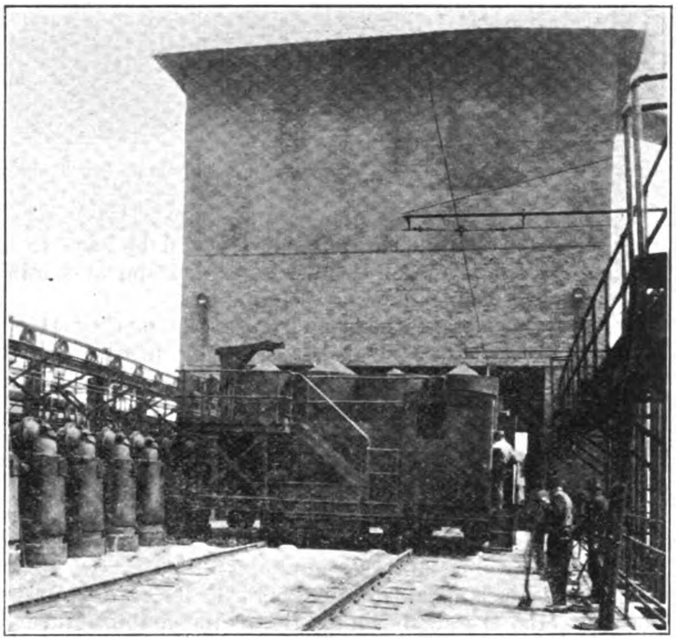


FIG. 6—VIEW TAKEN ON TOP OF THE COKE OVENS SHOWING THE ELECTRIC CHARGING-CAR

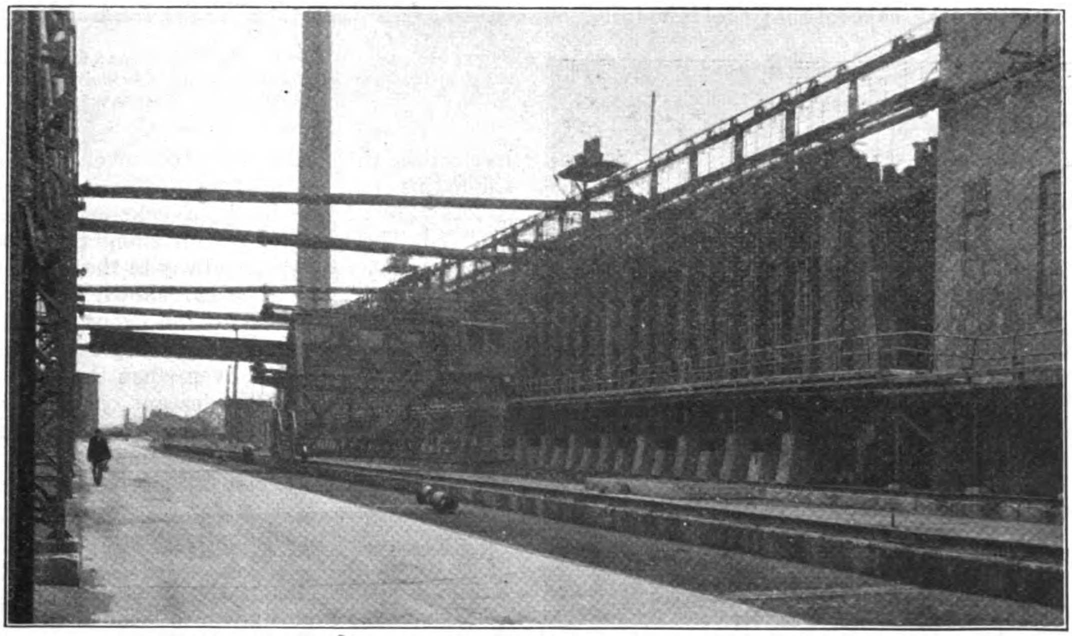


FIG. 7—THE SIDE OF A BATTERY OF COKE OVENS
The Gas Mains That Convey the Gas from the Ovens to the By-Products Plant Can Be Seen Running along the Top of the Ovens and across the Roadway. The Car in the Center Carries a Plunger That Is Used To Push the Coke out of the Oven

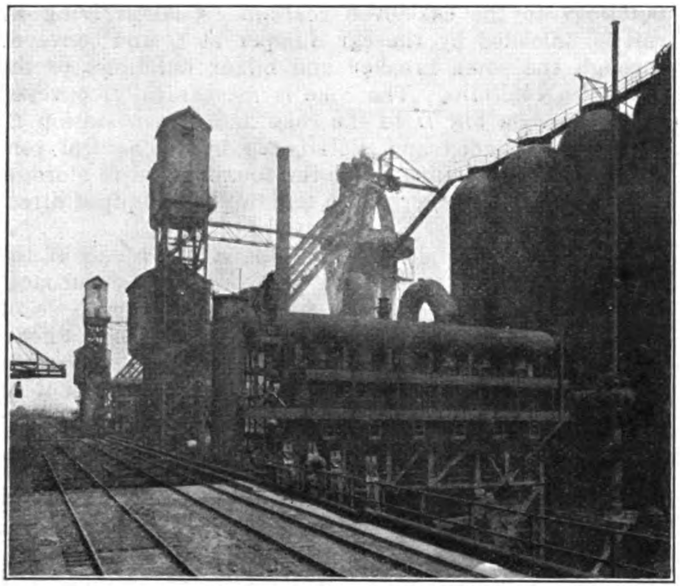


FIG. 8—ONE OF THE BLAST FURNACES THAT PRODUCES 500 TONS OF IRON DAILY

the buildings around the powerhouse will be seen to be justified by the eventual production of casting at a minimum price.

To obtain coke of the proper size and quality at the lowest cost, transportation expense had first to be decreased and the by-products of the coal and blast furnace, which are wasted ordinarily, had to be utilized. Coke is necessary to run the blast furnace and also the foundry cupolas. Since it usually costs the same or slightly more than coal, we buy the coal instead and reap the benefit of the by-products. Fig. 4 shows in diagrammatic form the disposition of the by-products. One ton of coal will produce approximately 0.75 ton of coke, 10,500 cu. ft. of gas, 8.5 gal. of tar, 25 to 30 lb. of ammonium sulphate and 2 gal. of light oil for the manufacture of motor fuel. From this the relative value of 1 ton of coal and 1 ton of coke is readily seen.

Of the gas lowest in by-products and heat value, 44 per cent is burned under the coke ovens; the remaining 56

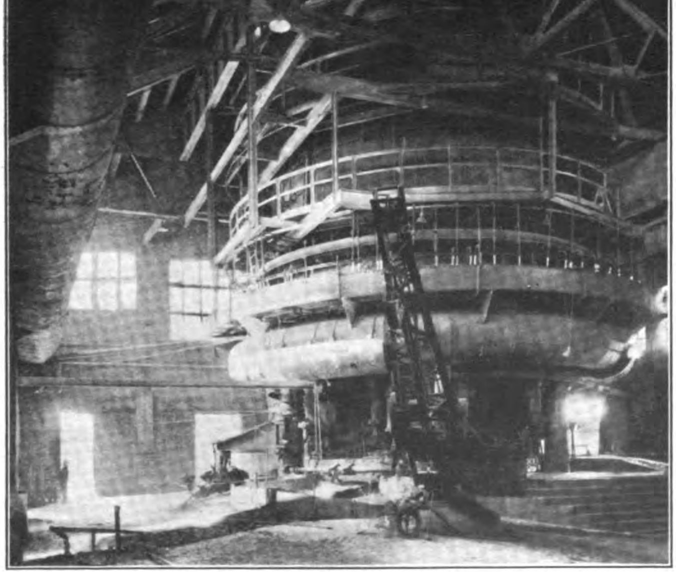


FIG. 9—BASE OF A BLAST FURNACE SHOWING THE STREAM OF MOLTEN IRON FLOWING TO THE LADLE THAT CONVEYS IT TO THE FOUNDRY

per cent is used in the heating furnaces around the Ford plants in the Detroit district, any surplus during light-load periods being turned into the Detroit city mains. The tar is used in heat-treating furnaces and under the boilers in the powerhouse. Ammonium sulphate is sold as fertilizer, and the light oil is mixed with gasoline and sold as motor benzol, 10,000 gal. of benzol being produced daily. The economy effected by operating one's own coke ovens is readily seen. At present there are 120 Semet-Solvay coke-ovens in operation; they use about 2000 tons of coal per day.

A general view of the coke-ovens and chemical plant is given in Fig. 5. The ovens themselves will be noted at the left with the coal-storage bin in the center, an inclined conveyor rising into its peak from the mixing building. The benzol plant is shown at the right. Fig. 6 is a view taken on top of the coke ovens and shows the electric charging-car that transfers a load of 16 tons of pulverized coal into the four openings at the top of each oven as it requires filling. The riser pipes seen at the

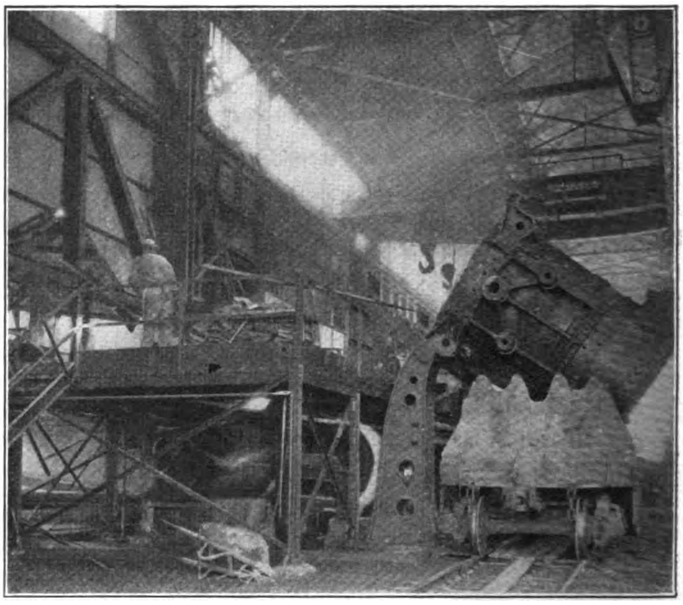


FIG. 10—THE 80-TON LADLE THAT TRANSFERS THE MOLTEN IRON FROM THE BLAST FURNACES TO THE FOUNDRY DISCHARGING ITS CONTENTS INTO A FOUNDRY LADLE

left collect the gases from each oven and feed them into the mains.

The side of one block of coke-ovens is illustrated in Fig. 7. The gas mains run along the top of the ovens and thence across the roadway to the pump-house and the by-products plant. The car shown in the center of the picture travels parallel to the ovens and carries a large plunger on the inner end of a deep I-beam. This plunger is pushed through each oven when the coking operation is completed, and the incandescent coke is discharged into an electric hopper-car on the opposite side of the oven. The hot coke is quenched and dumped into the coke bins, from which it is carried by mechanical conveyors to the coke and screen station for screening into blast-furnace, foundry and domestic sizes and distribution to other parts of the plant.

Fig. 8 shows one of the blast furnaces, its four air-stoves and the high-line or railroad trestle from which all material is fed into the skip-car that, in turn, travels up the inclined skip-bridge to the charging bell at the top of the furnace. There are two furnaces in the present

equipment, each having a capacity of 500 tons of metal per 24 hr.

We will consider next the benefits of the situation of the blast furnaces. Most people consider a blast furnace an instrument for producing pig iron only. Although its principal product is iron, its by-products are none the less valuable. Note particularly what is produced from a given charge:

Charge	*Products*
1 ton of coke	1 ton of iron
2 tons of ore	½ ton of slag
½ ton of limestone	6 tons of gas
4 tons of air	

The slag, which has been wasted, will be used very soon in the manufacture of cement, much of which will be consumed at the Ford plants for construction purposes. The gas, although of low heat-value, is a very valuable by-product. Nearly half of the blast-furnace gas is required for the four air-stoves which heat in turn the air blown into the blast-furnace. A part is burned to supply the power needed for operating the blowing engines and producing the electric power essential to the working of furnace skips, transfer cars, cranes and the like. Fifty tons of powdered coal and 36,000,000 cu. ft. of gas are burned daily under the boilers in the powerhouse, generating power for departments of the plant other than the blast furnace. This shows clearly the advantage of locating the powerhouse very near the blast furnace.

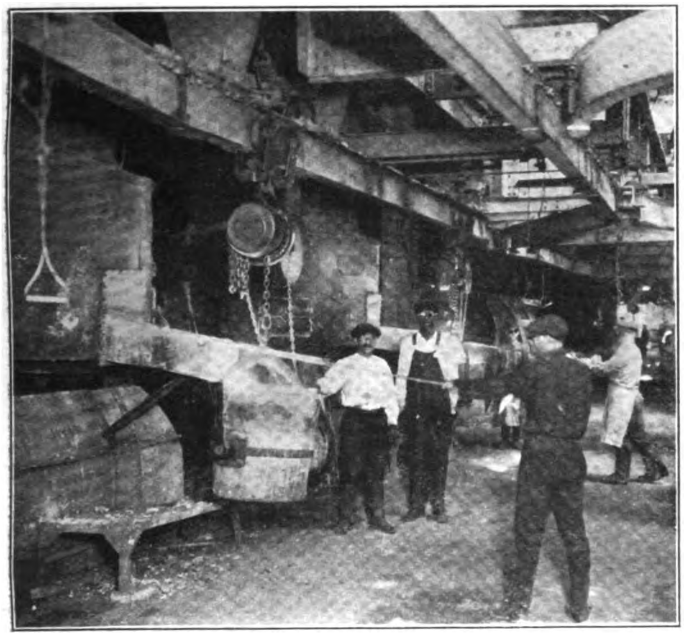


FIG. 11—POURING MOLTEN IRON FROM A CUPOLA INTO A FOUNDRY LADLE THAT IS SUSPENDED FROM AN OVERHEAD MONORAIL SYSTEM

POWERHOUSE AND FOUNDRY

The main powerhouse at the Rouge plant will eventually be the central power-source for all the Ford industries in the Detroit district. Its location at the source of large volumes of combustible by-products guarantees the generation of electric current at the minimum cost. The present equipment consists of four Ladd boilers rated at 2600 hp. each. They are fitted with a combustion system that injects liquid, gaseous or powdered fuel by air pressure. The amount of fuel is controlled electrically to meet the boiler load. Combustion is practically perfect, without ash, cinders or smoke. The customary dirt and disorder of the average boiler-room form a remarkable

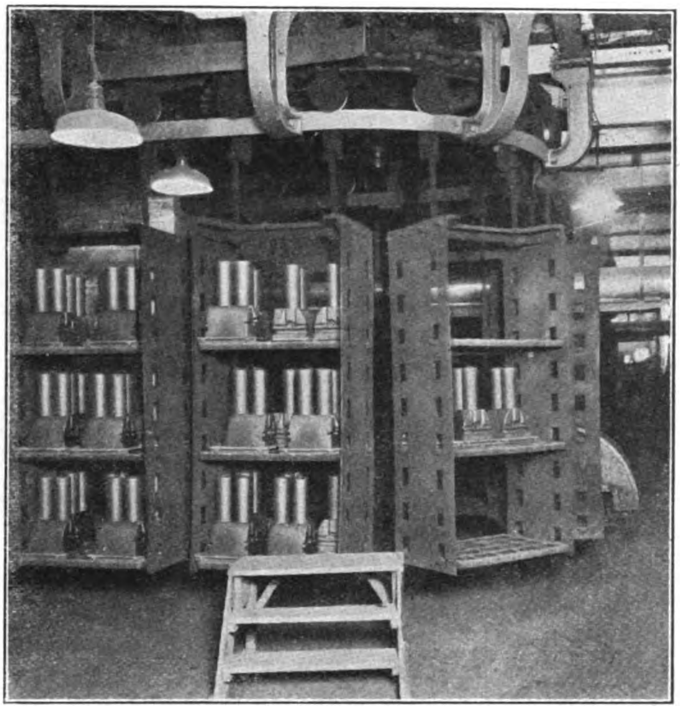


FIG. 12—ALL OF THE MATERIAL USED IN THE CORE ROOM IS HANDLED PROGRESSIVELY BY MECHANICAL CONVEYORS

contrast with the bright tiled floor and orderliness of this plant.

The electric generating equipment consists of two 12,500-kw. turbo-generator units. This capacity will soon be increased. Three turbo-compressors supply the blast for the blast-furnace line, their combined capacity reaching 45,000 cu. ft. of air per min.

The location of the foundry adjacent to the blast furnace is another basic consideration in the general scheme. Up to this time, pig iron has been taken from the furnace and cast into so-called pigs. These pigs are supplied to foundries and melted in cupolas. It was at this point, in the whole process of iron ore to finished cylinder, that a considerable saving was seen to be possible. One-half

FIG. 13—ONE OF THE CONVEYORS EMPLOYED IN THE CORE ROOM

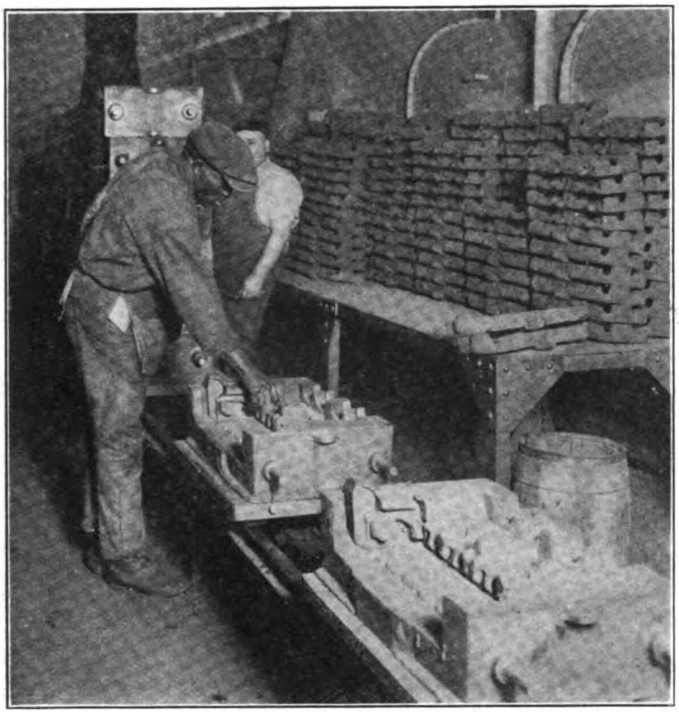


Fig. 14—Setting the Cores in the Drag Portion of a Mold as It Passes Along on a Mechanical Conveyor To Receive the Cope Portion

of the heat generated in the blast furnace is retained in the molten iron itself. Mr. Ford believed this heat should be conserved. In a foundry where approximately 1400 tons of iron is melted per day, this item of heat loss is a considerable one.

Through misinformation or misunderstanding, many writers have stated repeatedly that the blast-furnace iron is cast directly into molds without any additions or special treatment. This is possible; but it is not practicable because the iron produced by the modern blast-furnace contains too much carbon and its analysis is too variable for direct casting. To overcome this variation and to lower the carbon-content, a process has been adopted in the Ford foundry. This process enables us to use the foundry scrap and so-called back-stock that amounts to approximately 50 per cent of the iron poured into the molds. About 30 per cent of the iron poured into the Ford cylinder-mold is blast-furnace iron. The remainder is tapped from the cupolas and maintained at an analysis suitable for bringing the iron in the casting to a composition that agrees with our requirements. The percentage of manganese, sulphur and phosphorus is practically the same in both cupola and blast-furnace iron, silicon being the only element manipulated to make either soft or hard iron according to requirements. If an iron of 2-per cent silicon is required, for example, and the blast furnace is producing iron of 4-per cent silicon-content, the cupola would be run to produce 1-per cent silicon iron to give the desired chemical analysis.

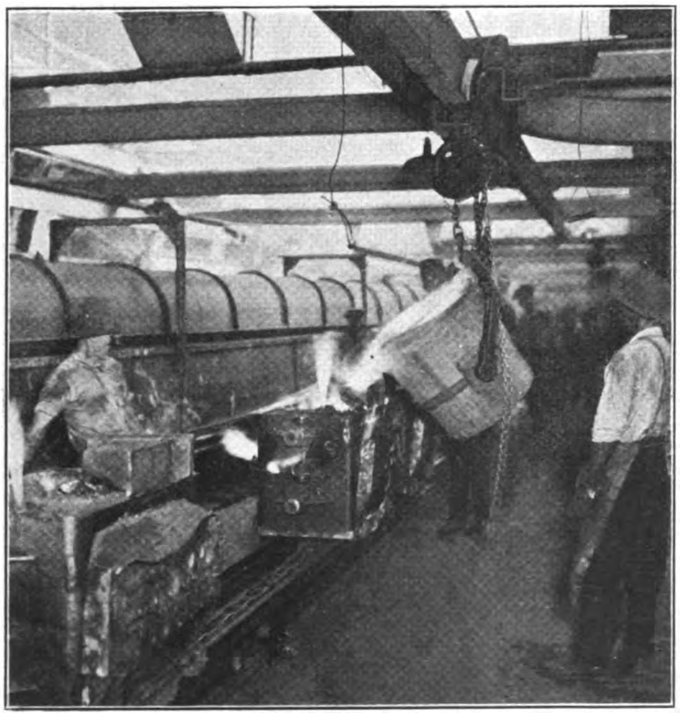


Fig. 15—Pouring the Molds

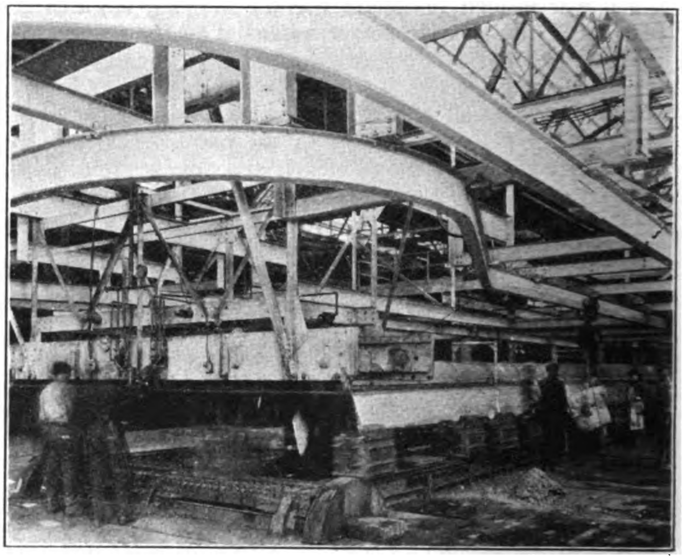


Fig. 16—After the Molds Have Been Poured They Travel to the End of the Outside Conveyor Where They Are Shifted Mechanically to the Center Conveyor That Carries Them to the Shake-Out Room

Fig. 9 shows the base of a blast furnace from which the metal is being drawn. The molten iron flows through a channel directly into an 80-ton ladle outside the building. The blast-furnace iron is conveyed by rail directly to the foundry in this ladle. Here it is tilted to the position shown in Fig. 10 by a gantry crane and discharges its contents into a cylindrical container mounted on an electrically propelled car. This car takes the iron to a point between two batteries of cupolas of four each and turns its contents into the foundry ladles hung from a monorail running over the cupola spouts and at right angles to them. The foundry ladles, after receiving a weighed amount of furnace iron, are taken to the cupolas to get their portion of cupola iron and then taken to the molds to be poured. Fig. 11 shows the foundry cupolas, ladle and monorail carriers. Twenty-four cupolas are arranged in batteries of eight each and these, with three stations for furnace iron, take care of the entire production.

Making the Cylinder Block

The foregoing is intended to show the magnitude and efficiency of the system needed to bring the iron to the molds at a low cost. Having provided good iron at a

minimum cost, additional economies must be effected in the molding end. Heretofore, castings have been made either in molds on the floor or on benches, from which they were transferred to the floor to be poured-off. This system was crude and wasteful in addition to being laborious. The molder was compelled to carry flasks from a pile or from the yard, fit his own cores and pour the iron. In the Ford foundry every workman has all the materials he works with brought directly to him.

The system or group of conveyors, molding machines and the like comprising one unit occupies a space only 50 x 300 ft. in area. Corerooms are situated between each two systems on the same floor-level. The shake-out, where the casting is removed from the mold, is at the rear of each system. Back of this is a large cleaning-room receiving the castings from the entire system.

The cores are made from new or green sand bonded with linseed oil or a mixture containing linseed oil, or with burnt or used sand bonded with compounds made principally from pitch. The green-sand cores are used where they are surrounded by metal, and the burnt-sand cores when they are only partially surrounded. All material in the corerooms is handled progressively by conveyors as shown in Figs. 12 and 13. The core-drying oven is placed near the center of the coreroom. Cores are made at one end of the room, dried in a continuous

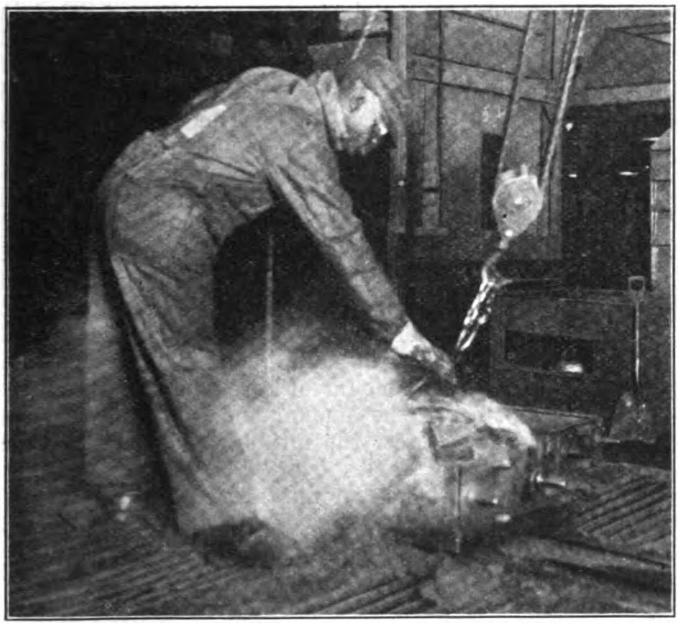


Fig. 17—After the Molds Are Shaken Out, the Loose Sand Falls through a Grating to a Sunken Conveyor

oven and carried by conveyor from the oven to the storage space at the opposite end of the room. The cores are taken from storage to the molding machines and used as required.

The cylinder is molded on the two outer chains of a system of three parallel conveyors. The two outside conveyors travel in the direction of the cupolas, while the center one moves in the opposite direction to the cleaning-room. Molding machines are placed on both sides of this system. The drag or bottom portion of the mold is rammed by hand near the starting point of the outer conveyor. Immediately after the drag has been rammed-up it is placed on the outside conveyor and, while in motion toward the cupola end, the cores are set. Fig. 14 illustrates this stage of the molding operation. Farther on the cope or top half of the mold is rammed-up and

Fig. 18—Apron Conveyor That Delivers Tempered Sand to the Hoppers Directly above the Molding Machines

placed on the drag. After this the copes and drags are clamped together and the runner or basin that receives the molten metal is made preparatory to pouring.

Pouring the molds takes place at the cupola ends of the outside conveyors as illustrated in Figs. 15 and 16. After being poured, the molds travel to the end of the outside conveyor and are shifted mechanically on a series of rollers to the center conveyor and started back toward the shake-out room. The iron solidifies and the castings become partly cool on their passage through the ventilated tunnel over the center conveyor. This removes a large amount of gas and smoke from the foundry. After

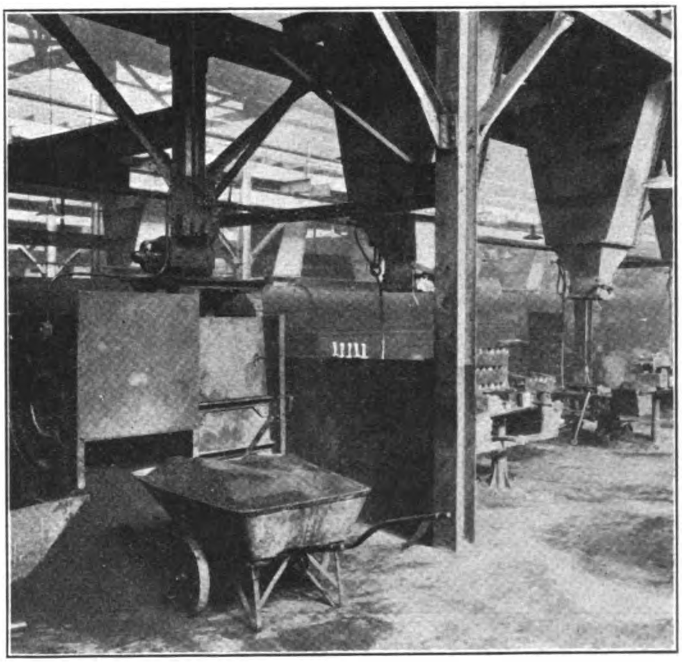


Fig. 19—Hoppers Supplying Sand That Still Retains Some Heat from the Previous Casting to the Molding Machines

the shake-out, the castings are transferred to a conveyor leading to a mezzanine floor and the loose sand drops to a sunken conveyor through a grating, as shown in Fig. 17. Enough new sand is added to the portion passing through this grating to replace the sand still adhering to the casting. The replenished sand is then automatically conveyed through a riddle or beater to break up the lumps and also through a mixer. The whole process is mechanical except the tempering, which is looked after by a man whose duty it is to add the required amount of water to dampen the sand. After this preparation, the sand is elevated by an apron conveyor shown in Fig. 18, which passes over the supply hoppers directly over the molding machines. Sand is drawn from these hoppers, which are shown in Fig. 19, through hand-controlled gates as each molder requires it. The sand has now completed a cycle and is ready for another mold. It is still warm from the previous casting.

The cylinder castings lie on trays until cold enough to handle and are then taken to the so-called knock-out where the bulk of the core sand is removed. This is accomplished at present by using pneumatic chisels, the cylinders resting on a grating similar to that on which the molds are shaken out. In this case, however, the burnt sand, when recovered, is used by the coreroom after proper tempering and preparation.

Cold cylinders, free from the bulk of sand in which they were cast, are then loaded on a conveyor which transfers them from the mezzanine to the ground floor where the tumbling-mills are located. A group of these mills is provided for each system. This particular conveyor also passes the cylinders between the tumblers that are grouped in two parallel rows. The cylinders are transferred by hand into the tumblers. The tumbling requires from 2 to 3 hr. The castings are removed from the tumblers by rope-hoists, returned to the same conveyor and transported to roller conveyors where the core wires are removed. From the roller conveyors, they are started on the last operations; these are performed on a slat conveyor. A crew of men on either side of this conveyor chip, grind and otherwise clean and prepare the cylinder-blocks for inspection. The rejected cylinders continue on the slat conveyor to a point farther on, where they are checked to determine the cause of rejection, removed to a truck and taken to the cupola-charging platform. The accepted castings are transferred to another slat-conveyor that runs at right angles to the cleaning conveyor; thence they pass into the machine-shop. Machining commences without delay, since the machine-shop is in the building that houses the foundry.

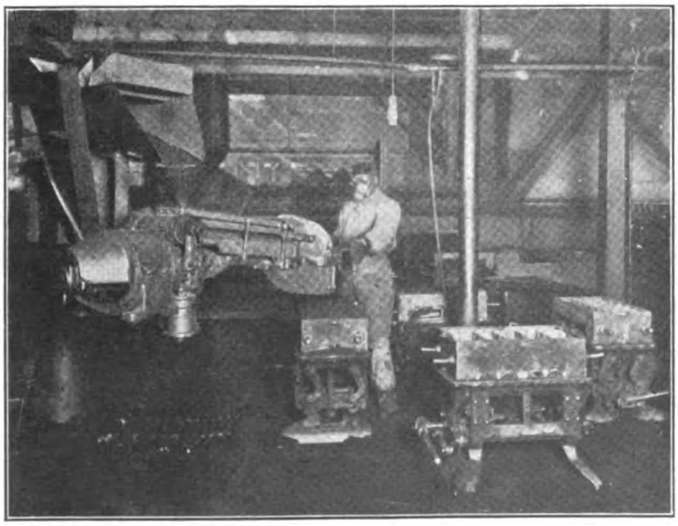


Fig. 20—An Experimental Molding Machine That Is Designed To Eliminate Hand Ramming in Which the Sand Is Packed by Being Thrown into the Mold at a High Velocity from the Periphery of a Revolving Wheel

The Ford cylinder, although simple as compared to other cylinders, is the most complex and difficult casting in the car. In its molding many troubles come up, which are ascribable mostly to its high production. Patterns wear rapidly due to abrasion by the sand and rough usage by machine-molders. They are being repaired continuously. The molding-machines also require attention nearly every week. The big problem of cylinder production lies in the constituents and usage of the sand and the iron. In a machine-shop most operations and materials are visible. The same thing is true of the patterns, machines and conveyors. A good mechanic who is always on the job is all that is required, but in the case of both the iron and the sand slight variations are hardly noticeable. Nature is not dependable when it comes to uniformity. The sand used in both molding and coremaking is ever-changing. What is right one day as regards mixtures may be altogether wrong the day following. Only certain grades of sand can be utilized successfully on a job where production is high and only common labor is employed. Sand varies with every different source of supply and even in the same pit. Shipments of sand are watched very closely. A certain amount of bond and a grain size have been determined upon and are adhered to as closely as possible. The sand must also be rammed properly. A mold rammed too hard or not enough will produce an inferior casting. Cores must be maintained at a certain composition; the voids between the grains must be sufficient to allow free passage of gases. The cores have to be strong enough to withstand rough handling and must not be easily destructible by the high temperatures of molten or very hot iron. No set rule is applied to our sand problems; the make-up of a core is modified to meet the difficulties as they arise.

Iron for the cylinder also requires constant watchfulness. The composition of scrap-iron is never dependable, due to a certain amount of foreign scrap that gets into our back-stock. The amount and condition of steel in the charge also affect the ultimate product, and the composition of coke varies over a considerable range. When melting back-stock for mixture with blast-furnace iron, we use about 1 part of coke to 6 parts of metal. The analysis of iron for the cylinders that is found to keep porosity at a minimum and at the same time permit easy machinability is as follows:

Silicon, per cent	2.20
Phosphorus, per cent	0.35
Sulphur, per cent	0.08
Carbon, per cent	3.30
Manganese	0.80

The process of manufacture as outlined serves us today, but it is continually being developed to a finer degree in accord with the policy of the company. The difficulties due to the human element have been reduced to a minimum and, although there are still some severe working conditions on the cylinder line, they are being eliminated one at a time. The hand-ramming probably will be replaced by a machine operation, upon which we are now experimenting; the cooling of the cylinder will take place while the latter is in motion, the present cooling-trays being eliminated. The experimental molding-machine is shown in Fig. 20. Sand is thrown into the mold at a considerable velocity from the periphery of a revolving wheel, and it seems to pack satisfactorily.

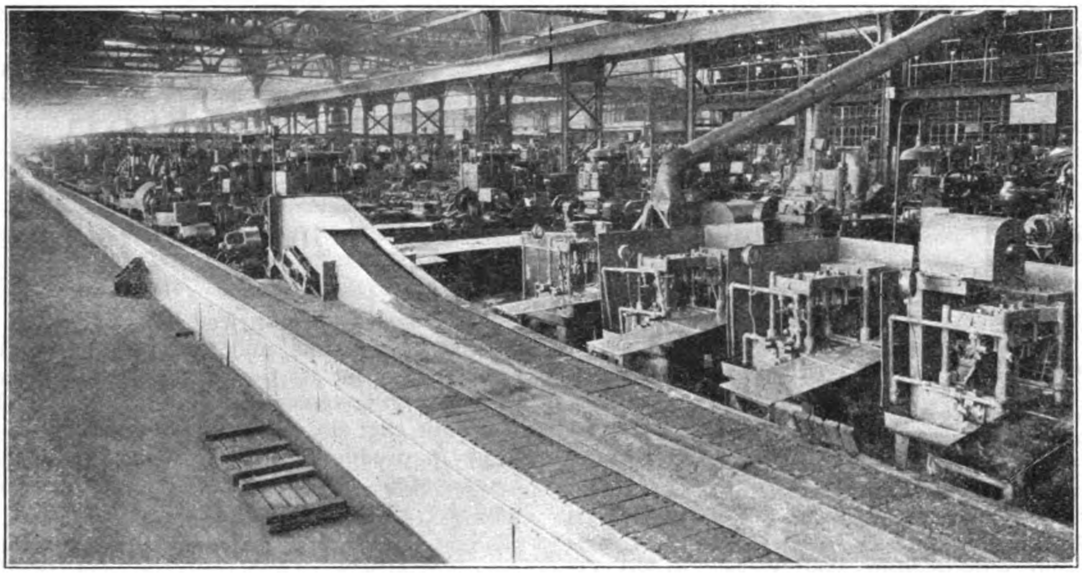


FIG. 21—GENERAL VIEW OF THE CYLINDER MACHINE SHOP

MACHINING THE CYLINDER

The cylinder block, as it comes from the foundry, is inspected for any foundry defects before being removed from the conveyor. The block is not allowed to touch the floor at any time in the actual operation, but moved from one machine to another by conveyors that are as nearly of the same height as the machines as possible. Two of these conveyors are shown in Fig. 21.

The first step in the machining operation is to locate four spots on the upper side of the block. This is done in two specially designed machines. The block then goes to a standard milling-machine to have the crankcase flange milled. After this the six main bearing boltholes, which serve as locating points throughout the entire operation of machining the cylinder block, are drilled and reamed. The top or head side of the block and the water-connection and the manifold side are next milled in one operation. The block is then placed on a four-spindle drilling-machine and a rough-cut taken out of the cylinder bore so that the casting can be tested under a water-pressure of 65 lb. to eliminate any castings that may be porous and will not stand pressure.

Several smaller milling and drilling operations come next. After this the block goes to multiple-spindle drilling-machines to have the port, valve-stem and push-rod holes bored. All of these machines are equipped with suitable fixtures so that accurately finished work can be turned out rapidly. The radius for the transmission cover is then turned on a standard lathe having special equipment to load and turn two blocks at once. Boring of the camshaft holes is next in order and this is done on a special machine that works from both ends of the block. The block is rebored and also reamed on a four-spindle machine that is similar to the one used to take the first rough-cut out of the bore, some changes in the feed and the speed of the tools of course being made on account of the difference in the nature of the work. The block is then ready for the three-way machine that drills five holes in the top, eight in the manifold side and two in the water-connection side. The block is then put in a four-way machine that drills 15 holes in the top, 4 in the front, 5 in the rear and 17 in the crankcase flange, the entire operation being completed in about 50 sec.

The next major operation is casting the babbitt in the main and crankshaft bearing and milling off the gates in a conveyor machine of special design. The babbitt is then planed to insure the proper fit in the cylinder casting and the correct density of the metal. The next step is the very important one of rolling or glazing the bore, which is done with a four-spindle standard machine driven by a reversible motor. The rolls, which are of special design, are ground very accurately and hardened very uniformly. The block is then placed in a specially designed two-way tapping-machine where 10 holes on one side and 2 on the other are threaded simultaneously. The next and last stage in the machining is the tapping of 15 holes in the top of the block, 4 holes in the transmission end and 3 holes in the front end at the same time by a three-way tapping-machine. From the completely machined blocks we receive approximately 70 tons of chips per day, which are returned to the foundry and immediately remelted to provide the cupola iron needed to make more castings.

After being completely machined the block goes through a specially constructed washing-machine and comes out on a conveyor to be inspected and given the final water-test. If it passes the inspection and test satisfactorily the cylinder block is stamped "ok" and travels along on a conveyor to the loading dock from whence it is shipped to the assembly plant.

MOLDING SAND INVESTIGATION

THE Bureau of Standards is conducting a series of tests to discover a sand with 100-per cent permeability. The advantage of finding a perfectly permeable sand or one that approximates perfect permeability is obvious. Having a standard sand with a known permeability, the suitability of every molding sand could be expressed as a percentage of the sand that was found to be 100 per cent permeable. To accomplish this result, several sands have been investigated. One commercial grade of sand, which is a very pure silica sand of a fairly uniform degree of fineness, has been found on a number of tests, both dry and with as high as 4 per cent of moisture, to be 100 per cent permeable. Further tests are being made to determine its colloidal matter or any other substances that might affect its permeability.

Standard versus Special Machine-Tools for Automotive Production

By R. K. Mitchell[1]

Detroit Production Meeting Paper

THE too prevalent tendency toward making large expenditures for special equipment when standard machine-tool equipment might well serve the purpose is deplored by the author, and a plea is made for the reduction of the altogether too large investment often carried under fixtures and permanent tools. Ill-considered plans may have as their objective only the design of some special fixture, but frequently result in a fixture or machine-tool that requires special driving and feed-mechanisms.

Some of the disadvantages that attend the use of special machines are stated and commented upon, and the benefits of using standard equipment whenever possible are set forth. Reference is made to possibilities of special jig-and-fixture design that would meet the needs of manufacturers of standard parts.

THIS paper is not an attempt to dictate a set rule or policy for the tool engineer or tool designer to follow in every problem that presents itself. Rather, it is a general adverse criticism of the present prevailing policy of making large expenditures for special equipment when standard equipment might well serve the purpose, and also a plea for the reduction of the altogether too large investment carried under fixtures and permanent tools.

There was a time when the automotive manufacturer found it necessary to build special machines for performing certain operations and making special parts. When an operation or a part of this description was required, the policy was to design special fixtures and machine-tools to meet the special conditions. But often when the original intention is only to design a special fixture, the ultimate result is a fixture or machine tool that requires special driving and feed mechanisms. Then comes the question of whether to design special drives and feeds for some machine that is already in the plant; and this is the critical point in the argument between special tools and standard equipment. In many instances the only machine tool that will accommodate the special heads is one designed and built purposely to meet this one particular difficulty; so, we arrive, perhaps unintentionally, at the stage we have so much desired to avoid, which is the design and fabrication of special machine-tools.

In the ordinary routine followed when building special machine-tools, we are confronted with numerous obstacles. The first is the fact that the average draftsman found in the general run of tool-designing departments has had neither the engineering nor the production experience essential to the proper designing of special machine-tools, and his lack of knowledge as to proper stresses, correct bearings, loads and the details to be employed, together with a lack of foresight in considering the interchangeability of parts, ease of replacement and the use, so far as possible, of standard parts, is reflected in the enormous first cost of the majority of special machine-tools that are built under private supervision. The actual construction usually is performed in the toolroom by high-priced labor, working an excessive amount of overtime, and the machine, finally completed, has yet to meet its first test. It will be acknowledged that very few special machines have ever been devised and built that did not demand much undue expense and delay in production, not to mention the many changes made before they began to function as originally intended.

Special Machinery

The governing motive behind the design of special machinery is usually economy in production, and this is very commendable; but lack of experience, errors in design and construction and the failure of a machine to work as intended do not pay dividends. For example, a special machine-tool for turning both sides of the flange and the face of a flywheel at one operation was designed and constructed recently at an expense of from $18,000 to $20,000. Three days after the machines were installed, they were abandoned; but, fortunately for the manufacturer, the old set-up was still available. This was not because the old set-up was more efficient; but because, although there was every opportunity to develop a machine that would give greater production, lack of foresight and poor design ruined the whole project. The worst blunder was that no provision had been made or could be made for the escape of chips. Chips from the upper cutters worked down and packed against the bottom face, impeding the two lower cutters and necessitating the removal of the chips with a chisel about every 10 min. The final outcome of this case was that the manufacturer had to go out in the market and buy standard machine-tools. If the outlay wasted in the design and construction of the special equipment had been applied to the purchase of standard equipment, it would have more than covered the standard machine-tools that were afterward purchased. The sum of the whole incident was that from $35,000 to $40,000 was expended, where $15,000 would have served the purpose. This is rather costly experimental work for the manufacturer. The money spent and wasted on this job alone should be a sufficient argument against the too prevalent weakness for designing special machinery on the least provocation.

A special machine-tool in production requires the services of a skilled or special operator, at least until those interested become familiar with its care and operation. If the operator should be absent for any reason, loss of time and production must result before another man can be broken-in.

Repair parts for special machine-tools are costly items. It develops not infrequently that patterns are broken, mislaid or left at the foundry and, when the casting is finally secured, it means day-and-night work in the toolroom with additional expense and delay. Perhaps all of the foregoing will be accepted as constituting problems that will arise in any trade or line.

The most forceful argument against special machine-

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tools at present is the unstable design and development of automotive parts. When a designer produces a special machine-tool to accommodate a certain part, he has no guarantee as to the life of that part. I venture to say that the average life of the majority of automotive parts without change in design is less than 6 months. Any change in the design of the part either obsoletes a special machine-tool or demands such expensive changes in its construction that the machine usually finds its way to the obsolete or salvage department long before its useful life has scarcely begun.

Standard Equipment

Let us consider now some of the advantages of using standard equipment and machine-tools. To-day, machine-tool builders have stocked the market with a large variety of simplified, standard machinery that can be adapted to special operations and parts with slight extra expense. In the first place, the standard machine-tool is very much cheaper than a special machine. It is built on a quantity-production basis, and designing and engineering charges are distributed over a greater number of units. The standard machine-tool is available for prompt delivery. It will have had a thorough trial in practical work before being placed on the market and passed out of the experimental stage. Reputable manufacturers of standard machinery build their machine-tools so that the parts are interchangeable and, in case of service requirements, are prepared to furnish any part promptly from stock. Consider for a moment the money that is tied up in special machinery patterns, extra castings and the like. With a standard machine-tool, in production in any large shop, if an operator is called from his machine, other men just as familiar with the operation of it are always available and capable of continuing production without loss of time.

Sometimes it is advisable to have special features on machine tools, such as a special number of spindles, possibly set at different angles, on a milling-machine. It has been proved that a standard make of milling-machine can be equipped with a suitable special head to take care of special work. If the special part made by the special head is ever changed or made obsolete, the manufacturer need not scrap the entire machine, but only the special tooling. In the case of a special machine-tool for this special work, the entire machine would become obsolete in such an instance. Also, the standard machine-tool with the special tooling can be secured in much less time and with far less expense than the special machine-tool. In the case of the standard machine-tool, if the special part only is scrapped, the machine itself still is adaptable to any other operation of a similar nature.

The foregoing example is cited simply to show what is possible with standard equipment on milling machines, but it is true also with lathes, grinding-machines and many other standard machine-tools. All I have attempted to deal with is the too-ready penchant for designing and building special machine-tools where the exercise of a little ingenuity and manipulation will produce a set-up that will serve the purpose just as satisfactorily, without the excessive outlay usually associated with the fabrication of special machine-tools.

Special Fixture Design

I believe there is ample room for improvement in the design of special fixtures. Too little attention is paid to the needs of manufacturers of standard parts whose product, if properly investigated, will be found to contain unlimited possibilities for incorporation in the design of special jigs and fixtures. In a recent issue of a popular weekly periodical there was a full-page spread advertising the merits and possibilities of standard bushings. This advertisement alone probably meant an expenditure of from $8,000 to $10,000 for that manufacturer. With dozens of companies in like manner placing their engineering staffs and experience in their particular line at our disposal, still we do not pay enough attention to their claims and the merits of their products to consider them when designing our own pet tools and equipment. So far as possible, when designing fixtures and tools, we should take advantage of all that the trade offers and attempt to simplify our creations. Frequent use of the three fundamentals of jig and fixture work, the clamp, the V-block and the angle-plate, is to be recommended.

As a recent instance, a large drum-type fixture was designed, built and installed on a machine. The cost was about $2,500, including special drive-gears and the like that were constantly breaking, delaying production and running up a continuous repair bill on this job. The annoyance and continuous expense demanded immediate action and the whole fixture was replaced with two small angle-plate fixtures on which V-blocks to oppose each other were fastened. One side was loaded while the part on the other side of the fixture was being milled. These two fixtures cost about $70 and actually increased the rate of production beyond that of the more elaborate and expensive fixture. This is only one of various similar instances that occur every day. I believe that the tool designer is so prone to become interested and intent on the design and construction of the fixture that he temporarily loses sight of the fact that the fixture or tool is not the ultimate issue, but only the means to an end.

MOTOR-VEHICLE KILLINGS AND THE ENGINEER

In the year 1921, the Bureau of the Census announces, 10,168 deaths from accidents caused by four-wheel motor vehicles occurred within the registration area of the United States, which contains about 82 per cent of the population. This is a death rate of 11.5 per 100,000, an increase of 28 per cent over 1917. Further than that, the increase in rate is itself increasing from year to year, and the rate in the 65 largest cities averages about 15 per 100,000. These are alarming statistics. Couple with them the statement just made by Chief Magistrate McAdoo of New York City, that before long all of Manhattan Island below 14th Street will have to be one-way streets barred to passenger vehicles and that there are 2000 unprotected crossings in the city where policemen are needed, and the seriousness of the motor-traffic problem will be realized. Part of the trouble is due to the laxity of the driver license requirements, part to the carelessness of drivers and their assumption of the right-of-way over the pedestrian at crossings, but mostly it is the inevitable result of an increase of motor use far beyond the capacity of a city street system laid out for slow moving traffic in small volume. This motor use will not decrease nor even remain stable. Driver and traffic regulations can only remove a part of the difficulty. The obvious solution lies only in a radical revision of our conception of what a city street is for, and this reduces to a problem for the engineer. Motor-vehicle boulevards, second-story streets and under or over crossings for pedestrians all are probabilities of the near future in our congested centers and engineers responsible for our city developments must take account of such things as actualities and not as dreams.—*Engineering News-Record.*

The Hot-Spot Method of Heavy-Fuel Preparation

By F. C. Mock[1] and M. E. Chandler[2]

Semi-Annual Meeting Paper *Illustrated with* Drawings

SINCE a number of verbal additions were made at the meeting to the preprinted text of the paper, particularly in the way of making clear first how far actual working conditions followed out the theoretical analyses, these are printed herewith to avoid the possibility of making incorrect assumptions from the previously printed text. For the convenience of the members a brief abstract of the paper as it was printed in the July issue of The Journal[3] is presented herewith.

ABSTRACT

THE development of intake-manifolds in the past has been confined mainly to modifications of constructional details. Believing that the increased use of automotive equipment will lead to a demand for fuel that will result in the higher cost and lower quality of the fuel, and being convinced that the sole requirement of satisfactory operation with kerosene and mixtures of the heavier oils with alcohol and benzol is the proper preparation of the fuel in the manifold, the authors have investigated the various methods of heat application in the endeavor to produce the minimum temperature necessary for a dry mixture.

Finding that this minimum temperature varied with the method of application of the heat, an analysis was made of the available methods on a functional rather than a structural basis. Three of these are discussed: (*a*) When the heat from the walls of the manifold is applied through the medium of the air; (*b*) when it is applied to the fuel alone, or partly to the fuel and partly to the air; and (*c*) when a spray of atomized fuel and air is directed against a heated surface. A device was constructed by which the three main variables, the exhaust temperature, the exhaust flow and the area of the heating surface, might be regulated and the three remaining variables, the quantity of air, the quantity of fuel supplied and the quantity of fuel vaporized, might be controlled.

Taking into account the wide range of temperatures that the air charge and fuel supply undergo before entering the intake-manifold system, a quantitative computation of heat transfer was made and the conclusions were drawn that only by a combination of centrifugal force, surface tension and the force of gravity could the unvaporized drops be separated from the fuel charge and that the conditions of combustion are governed by the rate of fuel feed from the manifold to the cylinder and not from the carbureter to the manifold.

THE DISCUSSION

F. C. Mock:—Supplementary to the paper, reference is made to an experiment with a hot-spot type of fuel heater in which the area of heating surface, mixture proportion and exhaust temperatures could be varied while observations of the condition of the fuel and temperatures in various parts of the system were made. This device is shown in Fig. 8 of the paper. It was mounted in the center of the exhaust manifold of a Continental Model 7-R 3¼ x 4½-in. six-cylinder engine, which was operated for several weeks with this device, on motor gasoline, high-test or aviation gasoline, kerosene and grain alcohol. It should be stated that this form of heater, having a horizontal heating surface, was not selected as being a desirable form for regular use in automotive service, but was chosen because of the convenience with which observations could be made.

Nature of Action at Hot-Spot

Our observations seemed consistently to show that whatever part of the fuel remains in contact with the hot-spot undergoes a sort of selective distillation, the light elements boiling off quickly and the heavier elements more slowly, being very much the same action as that in the distillation flash of Prof. R. E. Wilson's method for determining equilibrium solutions. Provided the temperature of the metal heating surface is above its boiling-point, each element of the fuel seems after boiling to depart from the pool of fuel on the heating surface as vapor at its own boiling-point. Very little heat is apparently communicated from the metal heating-surface and the liquid on the heating-surface to the airstream, and the final mixture-temperature is approximately such that its heat-content is the sum of that of the air part of the charge at its entering temperature and that of the fuel vapor at its average boiling-point. In other words, the heat balance and final mixture-temperatures with an exhaust hot-spot are substantially those obtained with the system shown in Fig. 4 of the paper. This combination then results in a fog mixture, the temperature of which depends upon the boiling-point of the fuel, its specific heat, the mixture proportion and the temperature of the entering air.

Fig. 10 shows the mixture temperatures that should result from the combination of gasoline and kerosene vapor with varying temperatures of the entering air. It should be borne in mind that such low mixture-temperatures as these can scarcely be obtained under a motor-car hood because of preheating of the air, and later heating of the mixture, from sources external to the hot-spot. With the greatest care that can be taken, the mixture will enter the valve ports from 15 to 25 deg. fahr. hotter than indicated by the curves.

The temperature values given for motor gasoline and kerosene are those computed from the observations of Prof. R. E. Wilson and Daniel P. Barnard, 4th, described in their paper on Condensation Temperatures of Gasoline and Kerosene-Air Mixtures,[4] which check very closely our own observations. For convenient reference, curves of the heat-content of gasoline and kerosene at different temperatures and at pressures corresponding to those of

[1] M.S.A.E.—Research engineer, Stromberg Motor Devices Co., Chicago.
[2] M.S.A.E.—Engineer of carbureter design and development, Stromberg Motor Devices Co., Chicago.
[3] See The Journal, July, 1922, p. 27.
[4] See The Journal, November, 1921, p. 318.

the vapor in the mixture at full throttle, are given in Fig. 11 in terms of British thermal units and degrees fahrenheit. These also were obtained from the work of Professor Wilson and Mr. Barnard and can be used to compute the rise in temperature necessary with different mixture-proportions, initial air temperatures and the like.

With the ordinary hot-spot, the fuel remains on the surface only at low air velocities in the manifold, and in customary use a considerable portion of the fuel goes by into the engine without having gone through the process of evaporation and condensation, so that the conditions described in the foregoing, and in the paragraphs following, are only partially carried out. Such fuel as does not boil at the hot-spot is carried farther along into the intake manifold where further evaporation takes place from its surface at a relatively slow rate.

When the temperature of the metal heating-surface was 200 deg. fahr. or more above the boiling-point of the fuel, there was a pronounced "spheroidal condition" and at times the drops of fuel would bounce around in the chamber like popping corn. Under this condition, a number of drops were caught up by the airdraft and swept out of the heating chamber into the intake-manifold without being evaporated. In our test, however, we found this only occasionally and as a temporary phenomenon, as it was only at the highest power outputs of the engine that the heating surface rose sufficiently beyond the boiling temperature for this condition to occur.

Final Limitations of the Hot-Spot Method

With kerosene and with alcohol, the temperature of the metal surface sometimes fell below the boiling-point of the fuel. Under such conditions the evaporation took place by surface evaporation rather than by the combined surface and internal evaporation of boiling, and consequently a considerably larger surface area was necessary at this time. It should be borne in mind that, during boiling, the limitation of heat transfer was probably the ability of the ribs to collect heat from the exhaust, as the rate of transmission from the metal surface to the liquid was sufficiently rapid to take away the heat as fast as it was collected from the exhaust. When the evaporation is only from the surface, however, the extent of surface presented is the main limitation and it becomes necessary to spread the fuel out in a very wide film or to recirculate and respray it on the hot-spot. Whether this surface evaporation can be obtained is, in our estimation, the consideration that will determine how low we can go in the scale of fuel elements, using the hot-spot method of preparation. It seems very likely that we will not be able to use successfully, in general service, fuels the boiling-point of which lies above 500 to 550 deg. fahr., so long as a low "idle" is desired; because, even allowing for the effect of the reduced intake pressure in lowering the boiling-point, the exhaust temperatures will not be adequate.

The heat for evaporation of fuel is obtained from the exhaust through the intermediation of the metal wall between, and the wall temperature is, therefore, lower than that of the exhaust but higher than that of the liquid film. The temperature of the heating surface, like the exhaust temperature, varies with the speed; but the percentage of variation is less. We found that the temperature drop of the heating surface under change of quantity or condition of mixture fed was a very good measure of the heat being taken up by the mixture. Any change of boiling-point, specific heat or latent heat of evaporation in the fuel is strikingly shown in the temperature of the heating surface. For instance, with the

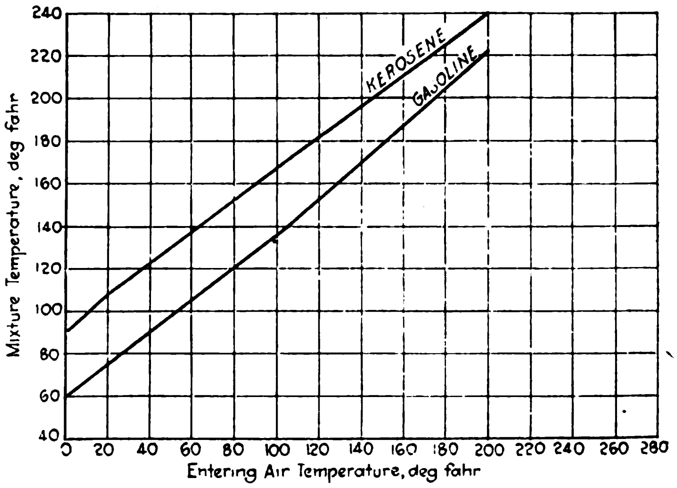

Fig. 10—Temperatures Resulting from the Mixing of High End-Point Gasoline and Kerosene Vapors with 15 Times Their Weight of Air at Varying Temperatures

exhaust temperature at 1000 deg. fahr., the metal-wall temperature at one point was with motor gasoline 595 deg., with kerosene 535 deg., and with alcohol 175 deg. fahr. This emphasizes what has been implied previously, that with alcohol and kerosene it is very important that an adequate amount of surface be presented to collect heat from the exhaust.

We were very much surprised to find how little heat was taken up when air alone passed through the heating chamber. With a surface more than adequate to vaporize a full charge when the fuel was taken into the heating chamber, if air alone were passed through the heating chamber, the fuel being taken into the airstream beyond the heating chamber, the temperature was only from 10 to 20 deg. fahr. above that of the entering air and the engine ran very poorly indeed, with every evidence of poor fuel-distribution. Another evidence of this point is that, at a speed and load at which the temperature of the heating plate was 400 deg. fahr. with no circulation above it, when the air alone for the engine was taken over the plate, the plate temperature fell 30 deg. fahr.; but, when both the air and fuel charge were sprayed on the heating surface, its temperature fell 150 deg. fahr., or five times the temperature drop when air alone was passing. Observation of thermocouples at various points of the heating surface showed that the air received most of its heat in making the right-angle bend to flow across

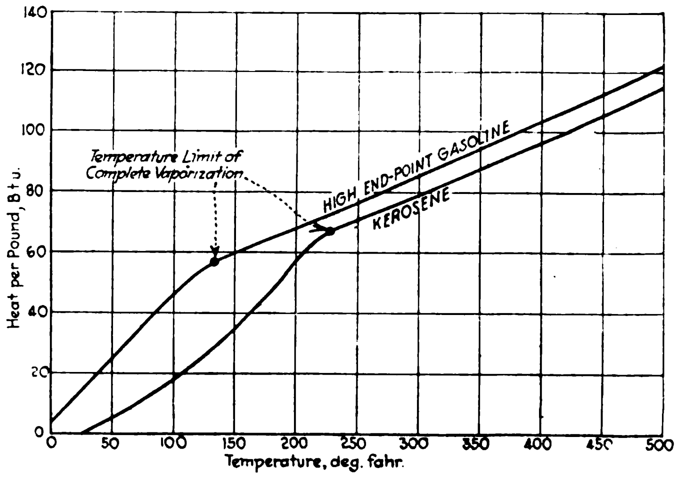

Fig. 11—Sensible Heat of High End-Point Gasoline and Kerosene at Varying Temperatures in a 15 to 1 Air-Fuel Mixture

the metal surface, and that there was very little heat-transfer beyond the bend.

Since the liquid fuel on the heating surface is at no higher temperature than at its boiling-point, while the surface is hotter than this, it seems reasonable to believe that the communication of heat directly from the liquid fuel to the air might almost be ignored.

The test heating-chamber was constructed so that the area of surface exposed could be varied as necessary to maintain a constant mixture-temperature at varying speeds and loads. We were pleased to find that a constant area of surface was required for any given fuel, at both low and high speeds, at wide-open throttle; and that the area of surface which was right for wide-open throttle, was adequate for part-throttle running. An exception to this rule has already been noted in that, when the temperature of the plate fell below the boiling-point of the fuel, considerably more surface was needed.

Under the conditions of our test, the area of surface presented to the fuel, as found adequate for creation of a dry fog, is given in Table 2. For idling with kerosene and alcohol, a surface approximately 40 per cent greater than is given in Table 2 seemed necessary when the fuel was spread out in a thin film.

TABLE 2—ADEQUATE HEATING SURFACE FOR EACH SPECIFIED VOLUME OF ENGINE PISTON DISPLACEMENT

Kind of Fuel	Adequate Heating Surface, Sq. In.	Engine Piston Displacement, Cu. In.
Motor Gasoline	1	6.5
Aviation Gasoline	1	11.5
Kerosene	1	4.5
Grain Alcohol	1	5.6

Under some conditions the surface collecting heat from the exhaust is a limiting factor of hot-spot capacity. In our experiments the surface presented to the exhaust was approximately three times that presented to the intake, giving the ratios of exhaust surface to piston displacement that are presented in Table 3.

TABLE 3—ADEQUATE EXHAUST SURFACE FOR EACH SPECIFIED VOLUME OF ENGINE PISTON DISPLACEMENT

Kind of Fuel	Adequate Exhaust Surface, Sq. In.	Piston Displacement, Cu. In.
Low-Test Motor-Gasoline	1	2.2
Aviation Gasoline	1	4.0
Kerosene	1	1.5
Grain Alcohol	1	1.9

The figures in Table 3 were obtained when there was a layer of soot about 1/32 in. thick on the exhaust surface. In their application to design it can be assumed that the whole mass of the exhaust manifold around and adjacent to the hot-spot is effective in collecting exhaust heat and conducting it to the heating surface. This can be used as a guide to the number of ribs necessary. It is of course essential that there be an actual circulation of exhaust gases over the surfaces included in the computation.

Laboratory versus Motor-Car Heat-Conditions

It should be borne in mind that there was a tremendous difference between the temperature conditions of our laboratory test and those existing under a motor-car hood. We had jacketed the intake-manifold with asbestos and placed asbestos shields between it and the exhaust pipe so that there was no radiation of heat from the exhaust manifold to the intake. We had a fan blast on the intake system so that the temperature was that of the room, 75 to 85 deg. fahr. instead of the 140 to 160-deg. fahr. fan-blast temperature that, in summer, is ordinarily directed onto the intake system under the hood; we also held the water temperature of the engine at 140 deg. fahr., which is somewhat lower than that of many engines when pulling a heavy load. The only heat applied to the mixture was at the heating surface.

When a hot-spot is applied to an intake system that has a long extended surface in proximity with the exhaust manifold, or one in which a large part of the intake passage is jacketed in the cylinder-head, there is bound to be a tremendous difference between the mixture temperatures of summer and winter operation, although no greater difference than would exist in mixture temperature without the hot-spot. If, in addition, the air enters the carbureter at 140 to 160 deg. fahr. in summer, and between zero and 40 deg. fahr. in winter, it is obvious that some sort of temperature regulation will be needed. But it would seem logical to place the control where the variation occurs, on the hood temperature or on the air entering the carbureter, rather than on the hot-spot, the temperature transfer of which varies very little between all seasons and conditions of operation. Indeed, thus far we know of no completely successful effort for correcting for atmospheric and seasonal temperature-changes of the intake system by change of the heat application at the hot-spot.

While on the subject of power loss from too much heat applied to the intake charge, it can be stated that the loss from expansion of the charge is less than is generally believed, and that the extreme and remarkable lack of power noted with some engines when there is too much heat on the manifold is due to a condition of detonation rather than the loss of so-called volumetric efficiency.

The Separating Hot-Spot

It is well known that many of the shortcomings now experienced with present hot-spots are due to the fact that the fuel does not stay on their heating surfaces long enough to be subjected to complete vaporization and conversion to a fog. The result is that the operation of the engine is efficient and correct only above certain limiting mixture-temperatures. The obvious step seems to be to incorporate with the heating surface a separator that will catch the unvaporized fuel-drops and return them to the heating surface. We have found that very good results can be obtained with such a device, but it is essential that the separating hot-spot have *adequate capacity for transmitting heat from the exhaust to the fuel;* otherwise there will be a time when the fuel, although metered in the carbureter, will stop and collect in the heating chamber instead of going to the engine, exactly as fuel "loads" in our present intake-manifolds at low air-velocities. But with the current type of intake-manifold, if the supply of vaporized fuel is inadequate to run the engine, it can always be increased by using the carbureter mixture control and raising the engine speed to a point where the air velocity carries a firing charge to each cylinder. With the separating hot-spot, this cannot be done, at least not until the separating chamber is filled with liquid fuel, but it is remarkable how well a passenger-car engine will perform on fuels so heavy that they cannot be vaporized and passed on to the engine except at part throttle. For truck and tractor usage, in fact for all heavy-duty usage, the separating hot-spot, if properly designed, presents the great advantage of abso-

(*Concluded on p.* 490)

Duralumin

By R. W. Daniels[1]

Cleveland Section Paper

THE author gives a short history and general description of duralumin and quotes the Navy specification of its physical properties as drawn by the Naval Aircraft Factory. The manufacture of duralumin is described and commented upon, inclusive of an enumeration of the improvement in physical properties produced at each stage. The physical properties are stated for annealed, heat-treated and hard-rolled duralumin and some of the possible automotive applications are suggested, inclusive of wormwheels, bearings, gears, connecting rods, rims and wheel parts and chassis and body trimming.

A report by the research department of the Fifth Avenue Coach Co. on the results of a test it made on duralumin wormwheels is included and the author details the advantages he claims as being attendant upon the usage of duralumin.

DURALUMIN is an aluminum alloy produced after years of systematic search to fill the demand for a metal combining the lightness of aluminum with the strength and toughness associated with ferrous metals. This condition has been met to a remarkable degree and the resulting physical characteristics make duralumin a most desirable material for extensive automotive application. As the commercial manufacture of this metal in this country dates back little more than 2 years, a short history and general description are given to afford a better understanding of the subject, although some information of this character already has been published.

Duralumin was first made in Germany and was developed by A. Wilm and associates during the years 1903 to 1914. The principal and unusual feature of this alloy is that after it has been hot, or hot and cold, worked, it can be strengthened and toughened further from 40 to 50 per cent by heat-treatment. This heat-treatment is somewhat analogous to that of the heat-treating alloy-steels, and consists of quenching from temperatures below its melting point, followed by an aging process. The increased physical properties are not all produced immediately on quenching, but increase during the subsequent aging. In addition to being made in Germany, the manufacture of duralumin was taken up in England by Vickers, Ltd., prior to the late war. During that conflict its use for structural purposes in connection with aviation brought the material before the eyes of the engineering world. To-day duralumin is recognized as occupying the same relative position to ordinary sheet or bar aluminum that heat-treated alloy-steel does to ordinary carbon-steel.

Duralumin is an aluminum alloy containing copper, manganese and magnesium. Its strength and toughness are comparable with those of mild steel, and are obtained with a specific gravity of 2.81 as against 7.80 for steel. The melting-point is approximately 655 deg. cent. (1211 deg. fahr.), the recalescence-point is 520 deg. cent. (968 deg. fahr.), the annealing temperature is approximately 360 deg. cent. (680 deg. fahr.) and the coefficient of expansion is 0.0000225 per degree of temperature centigrade (1.8 deg. fahr.). The chemical composition of the alloy varies within the following limits: copper, 3 to 5 per cent; magnesium, 0.3 to 0.6 per cent; manganese, 0.4 to 1.0 per cent; and the remainder is aluminum plus impurities. Small quantities of other metals are added sometimes for certain specific reasons. For instance, chromium can be added to increase the burnishing qualities of the metal.

The relative modulus of elasticity of duralumin is about one-third that of steel. The Bureau of Standards gives its value as being between 10,000,000 and 11,000,000 lb. per sq. in. Steel is quoted generally as having a modulus of elasticity of 29,000,000 lb. per sq. in. As the physical properties that can be obtained commercially from duralumin have not had much publicity, the following specification, as drawn up by the Naval Aircraft Factory, is of interest:

Material Specification for Duralumin

Use.—This specification is drawn to cover the requirements of duralumin sheet, rods and wire supplied to the Naval Aircraft Factory

General.—General specifications for the inspection of material, issued by the Navy Department, in effect at date of opening of bids, shall form part of this specification

Material.—This alloy shall show upon analysis the following chemical content:

	Percentage
Copper,	3.5 to 4.4
Magnesium,	0.2 to 0.75
Manganese,	0.4 to 1.0
Aluminum, minimum,	92.0 ..

Specimens for analysis or test shall be taken from the sheet, rod or wire selected as provided by the inspector

Manufacture.—No scrap shall be used other than that produced in the manufacturer's own plant and of same composition as the material specified

Workmanship and Finish.—The sheets must be of uniform quality; they must be sound, smooth, clean, flat and free from buckles, seams, slivers, scratches and other defects

Material in which defects are revealed by manufacturing operations shall be replaced by the manufacturer, notwithstanding the fact that the sheets, rods or wires have previously passed inspection

Physical Properties and Tests.—Duralumin is to be in the heat-treated condition. Its physical properties are to be as follows:

Specific Gravity, 2.80 to 2.85
Yield-Point in Tension, lb. per sq. in., 25,000
Tensile-Strength, lb. per sq. in., 55,000
Modulus of Elasticity, lb. per sq. in., 9,400,000

Selection of Test-Specimens.—At least one specimen for each of the tensile and bend tests shall be taken from a sheet selected to represent each individual melt of the material

The material shall be furnished in the annealed, quenched or "as-rolled" condition, as specified in the order

When material is ordered either in "quenched" or "as-rolled" condition, specimens for the tensile and bend tests shall be tested in the quenched condition. When material is ordered in the annealed

[1] A.S.A.E.—Baush Machine Tool Co., Springfield, Mass.

condition, specimens for the tensile and bend tests shall be treated in both the physical condition in which the material is received and also in the quenched condition

Specimens for the tensile and bend tests shall be prepared in accordance with the General Specifications for Inspection of Material issued by the Navy Department, except that the form of test-specimens shall be as shown in a sketch to be obtained upon application to the Naval Aircraft Factory

Tensile-Strength.—Tensile test-specimens cut in any direction from the sheets must have the properties specified in Table 1

TABLE 1—TENSILE TEST REQUIREMENTS

Physical Condition	Property	Sheets or Strips 0.05 In. Thick or Less		Sheets or Strips Over 0.05 In. Thick	
		Minimum	Maximum	Minimum	Maximum
Annealed	Ultimate Tensile-Strength, lb. per sq. in.	25,000	38,000	25,000	38,000
Annealed	Elongation in 2 In., per cent	10		10	
Quenched[a]	Ultimate Tensile-Strength, Minimum, lb. per sq. in.	55,000		55,000	
Quenched[a]	Yield-Point, Minimum, lb. per sq. in.	25,000		25,000	
Quenched[a]	Elongation in 2 In., per cent	18		18	

[a] Quenched specimens shall not be tested within 4 days after completion of heat-treatment. Annealed specimens shall be tested within 12 hr. after treatment.

Bend Test.—Specimens cut in any direction from sheets either annealed or quenched must withstand bending cold through an angle of 180 deg. over a diameter equal to four times the thickness of the sheet, without cracking

Dimensions and Tolerances.—The sheets shall be shipped in the lengths and widths called for in the order. The tolerances given in Table 2 will be allowed on the thickness of the sheets.

TABLE 2—ALLOWABLE TOLERANCES FOR SHEETS NOT WIDER THAN 18 IN.

Normal Thickness, in.	Tolerances, in.
0.0808 and more	±0.005
0.0808 to 0.0359	±0.003
0.0320 or less	±0.001

In duralumin forgings where the sections are heavy, it is advisable to lower the minimum tensile-strength requirements to 50,000 lb. per sq. in.; a proportional increase in elongation will be found. Duralumin is unaffected by mercury, is non-magnetic, withstands atmospheric influences and offers a remarkable resistance to sea and fresh waters. It is affected only slightly by numerous chemicals which, in the ordinary way, corrode other metals and alloys so readily; it does not tarnish in the presence of sulphureted hydrogen; and it takes a polish equal to nickel-plating and remains bright without cleaning longer than any plated or silvered article. It is the ideal substitute for aluminum, German silver, brass, copper, nickel-plated and silvered articles, and is the only substitute for steel where lightness combined with the strength of that metal is required. It is the only light metal that can replace steel in forgings, with a two-thirds saving in weight. Heat-treated duralumin forgings approximate mild-steel forgings in strength. Wherever weight is a deciding factor, duralumin is the most satisfactory metal for most shapes made by hot-working or forging. Naturally, duralumin forgings are especially desirable for reciprocating or moving parts where inertia, due to their own weight, forms a large part of the total stress. Duralumin machines and polishes very easily and, as it does not rust or corrode, it can be used in many places where weight is not the prime essential.

The manufacture of duralumin is somewhat analogous to that of steel and, in brief, is as follows:

(1) Manufacture of the alloy from its aluminum base
(2) Casting the ingot
(3) Hot-rolling or cogging in blooms, billets or slabs
(4) Hot or cold-working to final shape
(5) Heat-treating

The ingots are poured at as low a temperature as is practicable; that is, just enough above the melting-point to fill the mold and prevent cold-shuts. The ingots are then either hot-rolled or cogged into slabs, blooms or billets, similar to the manner of working steel. This hot-working is done at a temperature of from 450 to 480 deg. cent. (842 to 896 deg. fahr.), and care must be used not to perform any work on the metal above these temperatures because there is a danger of hot-shortness if the material is rolled or forged at higher temperatures. It is seen readily that such low temperatures cannot be judged by color; therefore, it is necessary to use accurate pyrometers in heating the metal, previously to working. The final rolling or forging can be done hot or cold, according to the character of the work being handled or the nature of the shape it is desired to produce.

The hot or cold-worked metal in its final shape shows greatly improved physical properties over those of the cast ingot, but the full development of its qualities is obtained only by a specific heat-treatment. To obtain this heat-treatment, the metal is heated to a temperature of 500 to 520 deg. cent. (932 to 968 deg. fahr.) for a period of time, depending upon the section of the piece, and immediately quenched. The heating and quenching immediately start to improve the physical qualities of the metal, but the maximum results are obtained only by the subsequent aging. During the aging period, which takes from 1 to 5 days, the alloy markedly increases in tensile-strength, hardness and elongation. Aging is sometimes accelerated by placing the metal in a hot-water bath up to 100 deg. cent. (212 deg. fahr.), or in a hot room. The above heat-treatment develops the remarkable properties possessed by duralumin, and these properties have not been obtained in like degree in any other aluminum alloy.

Improvement of Physical Properties

The various stages of manufacture, as related, increase the physical properties of duralumin by distinct steps. The cast ingot shows a tensile-strength of from 28,000 to 32,000 lb. per sq. in., with an elongation in 2 in. of from 1 to 3 per cent. The hot or cold-worked metal shows a tensile-strength of from 40,000 to 50,000 lb. per sq. in., with an elongation of from 6 to 12 per cent. These last figures are variable, depending upon the amount of working in the cold state. Upon subsequent heat-treatment and aging, the physical properties of duralumin show a marked increase, namely, 55,000 to 65,000 lb. per sq. in. tensile-strength and an elongation of from 18 to 25 per cent.

When it is required to put a considerable amount of work upon duralumin in its finished state, it often is found necessary to anneal the sheets between operations in precisely the same manner as in handling other metals. This annealing should be done at 350 deg. cent. (662 deg. fahr.). If several drawing operations are to be per-

formed, it may be necessary to anneal the metal between such operations. Annealed duralumin can be heat-treated and the maximum physical properties obtained, no matter what the shape or form to which the metal may be reduced. Conversely, heat-treated duralumin can be annealed.

Duralumin can be cold-worked after heat-treatment and aging. This operation produces a hard, smooth finish and materially increases the tensile-strength of the metal at the expense of elongation; that is, the tensile-strength will increase from 6000 to 10,000 lb. per sq. in. over that of the heat-treated metal, but the elongation may drop as low as 3 or 4 per cent.

In the annealed form it can be drawn, spun, stamped or formed into a great variety of shapes, as is the case of brass and mild steel. The physical properties in this state average as follows:

Ultimate Tensile-Strength, lb. per sq. in.,	25,000 to 35,000
Elongation in 2 In., per cent,	10 to 14
Brinell Hardness	54 to 60
Scleroscope Hardness	9 to 12

Duralumin in its heat-treated form can be slightly shaped or formed and can be bent cold to 180 deg. over a mandrel four times the thickness of the sheet. Its remarkable tensile-strength is here combined with its maximum elongation as follows:

Ultimate Tensile Strength, lb. per sq. in.,	55,000 to 62,000
Yield-Point, lb. per sq. in.,	30,000 to 36,000
Elongation in 2 In., per cent,	18 to 25
Brinell Hardness,	93 to 100
Scleroscope Hardness,	23 to 27

Heat-treated duralumin forgings have similar physical properties. Heat-treated and hard-rolled duralumin is used where no bending or forming is required. It is a very hard, strong, springy metal in this state and machines or polishes beautifully. Its physical properties in this form average as follows:

Ultimate Tensile-Strength, lb. per sq. in.,	67,000 to 72,000
Yield-Point, lb. per sq. in.,	55,000 to 65,000
Elongation in 2 In., per cent,	3 to 8
Brinell Hardness,	130 to 140
Scleroscope Hardness,	37 to 42

Having covered the general characteristics of the metal, a more intimate discussion of a few of the many automotive applications is given. Some of these applications are still under experimental observation and, in others, duralumin has been adopted as a standard material.

Automotive Wormwheels and Bearings

During the past 2 years much experimental work has been done along this line and the data are now available. Since the characteristics of the metal brought out in this class of service are highly desirable in other forms of gearing, bushings and the general replacement of bronze, these data are given at some length.

From the general description of duralumin it will be seen readily that here is an ideal material for wormwheels, provided the bearing or wearing qualities are satisfactory. For a given section, the weight is one-third that of the conventional bronze. The tensile-strength and the relatively high elastic-limit assure superior tooth-strength. The homogeneous structure and uniform hardness of heat-treated duralumin forgings obviate hard spots, porosity and spongy areas so common in bronze castings, entailing not only machining losses but uneven tooth-wear in service. The excellent machining qualities assure the manufacturer a saving in the machining costs, compared with those of bronze.

The wearing qualities of wormwheels for automotive purposes is best determined by actual road service, as bench or laboratory-test results do not always correspond. It is instructive, however, to compare results obtained from duralumin with those of other materials under identical conditions. The data from various laboratory tests under my observation on bronze and duralumin wormwheels can be summarized by saying that tests destructive to duralumin wormwheels were also destructive to those made of bronze. The results are always good where duralumin and hardened steel are run together. An example of this application is shown by duralumin connecting-rods running direct on the wrist-pins with better life at this point than with the conventional bronze-bushed connecting-rod of equal bearing area.

Comparative tests of bearings made from duralumin against bearings made of genuine babbitt metal show that, for shaft speeds exceeding 700 r.p.m. and loads over 200 lb. per sq. in., duralumin bearings develop less friction, remain cooler and show practically no loss in weight under the most severe conditions. For lower bearing pressures and slower speeds, babbitt metal is superior. Table 3 shows the details of this test.

TABLE 3—COMPARATIVE BEARING TEST

Loads, Lb. per Sq. In.	Speed, R.P.M.	Total Number of Revolutions	Final Temperature Deg. Cent.	Final Temperature Deg. Fahr.	Rise in Temperature Deg. Cent.	Rise in Temperature Deg. Fahr.	Friction, Lb.	Loss of Weight Grams
Baush Duralumin, Grade B								
100	632	37,920	39	102	18	64	21.15	.....
200	625	37,500	71	160	70	158	42.30	(a)
300	629	37,740	54	129	32	90	63.45	.....
400	623	37,380	62	144	39	102	84.60	(b)
Genuine Babbitt, Bureau of Standards								
100	694	12,230	89	192	53	95	22.00	0.028
200	706	16,510	102	216	58	104	29.00	0.021
300	686	15,150	125	257	100	180	38.00	0.018 (c)
400	603	5,500	139	282	94	169	79.00	0.054 (d)

(a) Bearing roughed and ran warm in 10 min.
(b) No measurable loss of weight
(c) Belt slipping
(d) Bearing seized, smoking

In regard to road tests, a considerable number of duralumin wormwheels are now actually in regular service in trucks ranging from 1 to 3½-ton capacity. These wheels have been in service from a few weeks to over 2 years without any failure. As these wheels are all running, complete data are not available, but through the courtesy of G. A. Green, vice-president and general manager of the Fifth Avenue Coach Co., New York City, I quote from the report of one of its preliminary tests under date of Aug. 2, 1921, as follows:

FIFTH AVENUE COACH CO. REPORT

General.—The greatest possibility of effecting weight-saving lies in the employment of aluminum or some of its alloys. With this idea in mind it was decided to test in road service a rear-axle wormwheel fabricated from an aluminum alloy commercially known as duralumin

Object.—To determine by road test the merits of a duralumin wormwheel, especially noting its resistance to wear, relative weights and the like, as compared with the standard bronze unit

Description.—Duralumin is a light aluminum-alloy having a specific gravity of 2.82. It can be forged, stamped, drawn or spun. The product is highly resistant to corrosion. The metal is heat-treated by the producer in a manner that is not made public. The following physical properties are claimed for this material:

Tensile-Strength, lb. per sq. in.,	55,000
Elastic-Limit, lb. per sq. in.,	30,000
Elongation in 2 In., per cent,	18

Bend cold over 180 deg. on a mandrel four times the thickness of the sheet.

the motor car and, if they are not available, the building of a wye is not an insurmountable obstacle. Possibly we had better solve some of the other problems before we attack the solution of this double-end-car proposition too seriously.

Another thing that we must keep clearly in mind if we want to make a real success of rail motor-cars is that the idea of using them must be sold to the public. There are certain objections in the public mind in regard to the riding qualities and some other things connected with the rail motor-car; they may be imaginary, but they constitute a real problem.

SOME REQUIREMENTS FOR THE RAIL MOTOR-CAR

BY W. L. BEAN[1]

THE rail motor-cars now used by the New York, New Haven & Hartford Railroad are illustrated and commented upon, and statistical data regarding their operation are presented. The features mentioned include engine type and size, transmission system, gear-ratio, double end-control, engine cooling, heating by utilizing exhaust gases and exclusion of exhaust-gas fumes from the car interior. A table gives revenue data.

THE usual steam-railroad coach is heavy. The impacts of that car on the track are sufficient to cause a yielding of the roadbed, track and ties; whereas a light vehicle has little or no effect. The smaller rail-pressures reflect themselves, for instance, in the fact that we have trouble in operating electric crossing-signals, particularly on Monday morning when there has been no traffic over the line on Sunday. That is not altogether due to weight, it is a matter of wheelbase and speed; but it shows that the wheel pressure on the rail is so light in relation to what usually operates over the tracks that it is relatively negligible. That has its effects on the riding qualities, yet we must keep the vehicle light to propel it economically with the gasoline engine.

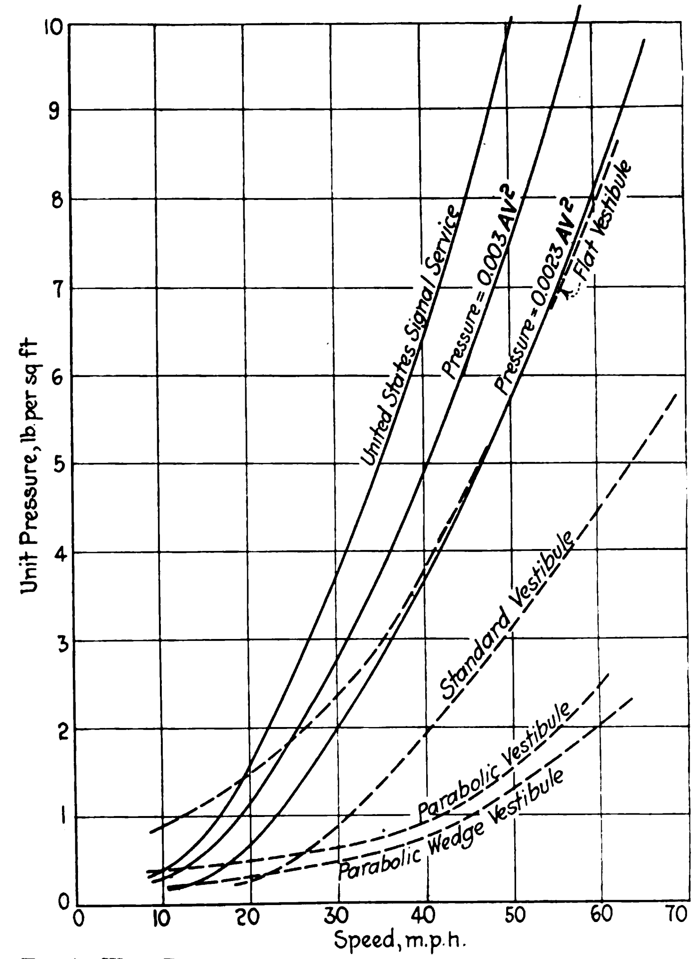

FIG. 1—WIND-RESISTANCE CURVES FOR CARS OF DIFFERENT FRONTAL SURFACES OPERATING AT VARIOUS SPEEDS

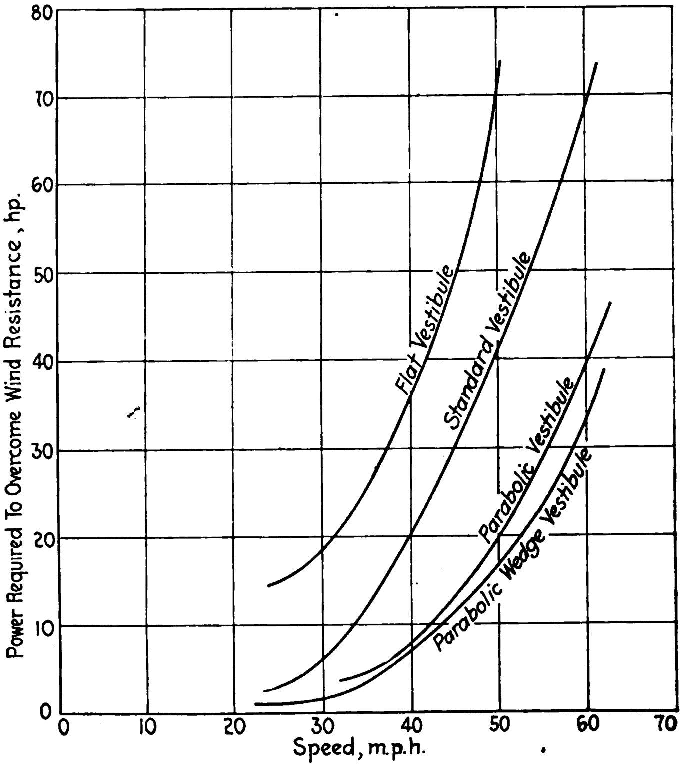

FIG. 2—HORSEPOWER REQUIRED TO PROPEL CARS HAVING DIFFERENT CONTOURS OF FRONTAL SURFACE AT VARIOUS SPEEDS

We think the rail motor-cars now used on the New York, New Haven & Hartford Railroad constitute an exceedingly good beginning and that, as a foundation from which to work, a much better unit can and should be developed. But there is no use in disguising facts. If one will ride in one of those cars for a number of hours, one must admit that the wear and tear on one's nerves is more than in riding on a steam car; and the patrons bring up that proposition. So, while we are working for more power, we need the greater flexibility together with perhaps a six-cylinder engine, because I believe that we should not consider more than six cylinders for some time to come. We must get a six-cylinder engine that is designed for rail motor-car service; this means continuous heavy duty and not a light throttle and drifting,

[1] Mechanical assistant to the president, New York, New Haven & Hartford Railroad Co., New Haven, Conn.

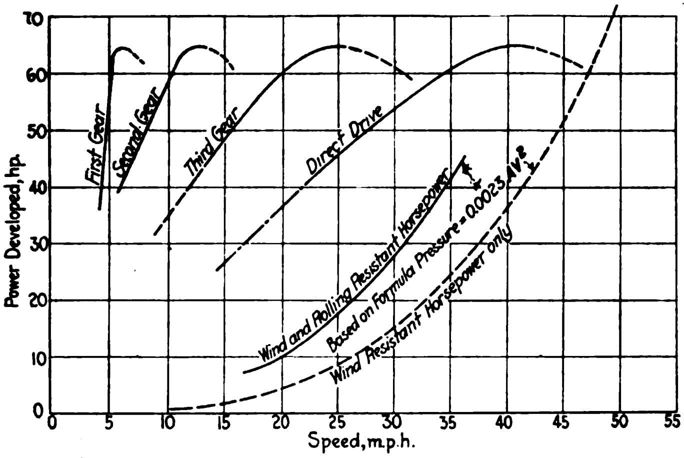

FIG. 3—RELATION BETWEEN THE HORSEPOWER DEVELOPED BY THE FOUR-CYLINDER MACK ENGINE AND THE CAR SPEED

ating standpoint, than a single-end car. We have terminals on our railroad where a car that could come in and shuttle out would be vastly superior to one that would have to go out to a turntable or a wye; in fact, it would be almost impossible to carry out some of the schedules that are contemplated for such a double-end car, when using a single-end car.

The matter of cooling the engine requires special consideration. We find that the heavy, continuous service requires a greater ability to dissipate heat than in highway cars. The matter of heating by utilizing exhaust gases efficiently and satisfactorily demands considerable study and arrangements that will keep gas fumes outside of the car body are necessary and have not altogether been worked out.

Table 1 gives data covering the revenue service up to recent date. Incidentally, the car had made considerable mileage before that. Under the heading Average Passen-

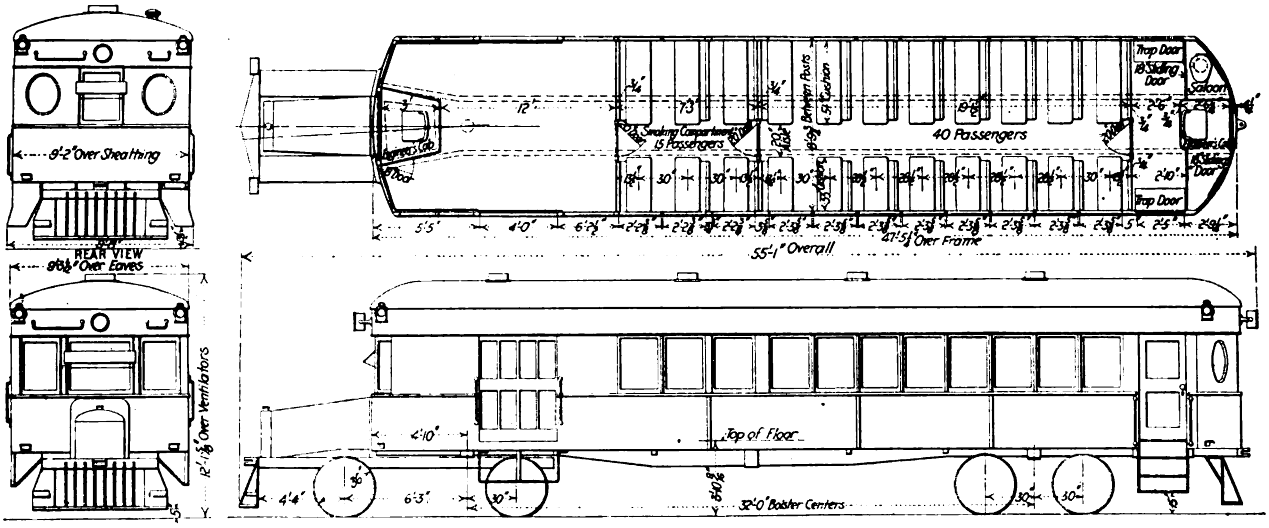

FIG. 4—PLAN VIEW OF A 55-PASSENGER MOTOR RAILCOACH

slowing up for traffic or turning corners. It means full throttle the majority of the time. The engine must be designed for that usage not only from the angle of ability to stand up but with regard to minimum vibration, which of course includes the suspension of the engine. That is important in a rail motor-car because of the sort of drumhead effect given by the roof, floor and sides of the car.

Transmission systems should be designed to permit operating the engine at less than normal speed; that is, at favorable speeds for economy and quietness, when the demand on the engine is less than normal. For instance, a car may drift successfully for 15 to 20 miles down a water-grade. That should be done through a gear-ratio properly adapted; we should be able to let that car drift at less than the full engine-speed. The car may require only power enough to propel it from 20 to 30 or 35 m.p.h. but, if its engine must turn over just as fast as if it were making 35 m.p.h. on the flat, developing a corresponding horsepower, we get conditions of considerable vibration. I think that can be avoided in a measure through the use of proper gear-ratios, and we intend to conduct further experiments with our cars in that direction.

As to the double-end car, I wish we could consider developing cars now without that feature; but the operating requirements in some localities are such that a double-end car would be much superior, from an oper-

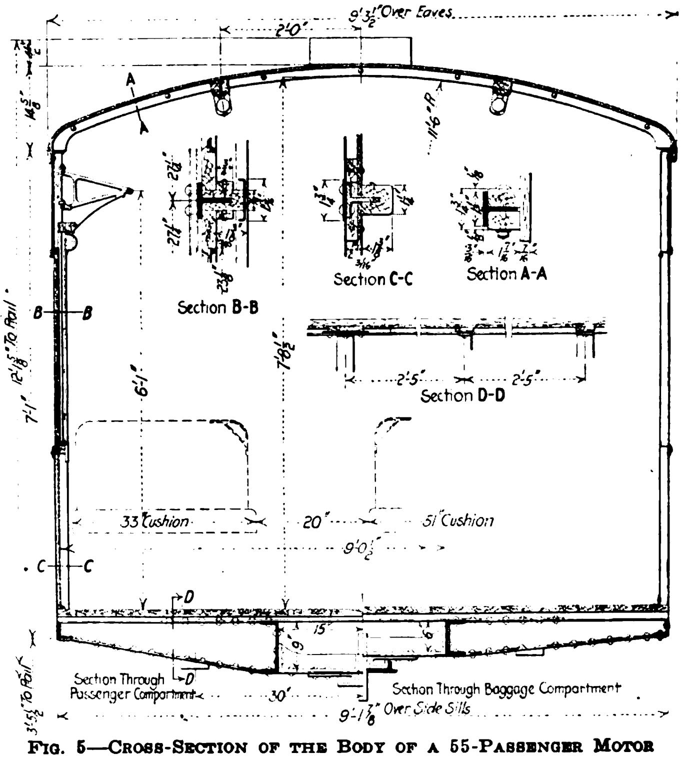

FIG. 5—CROSS-SECTION OF THE BODY OF A 55-PASSENGER MOTOR RAILCOACH

TABLE 1—STATISTICAL DATA ON GASOLINE-DRIVEN RAIL MOTOR-CARS

	Car No.		
Items	9,000[a]	9,001[b]	9,002[c]
Placed in Service, 1922	Jan. 4	Jan. 18	Jan. 30
Daily Total, miles	146	59	139
Total Mileage to May 6, 1922	12,441	5,479	10,396
Number of Revenue Passengers to May 6, 1922	11,115	6,822	11,013
Average Number of Passengers per Trip — Revenue	20.0	19.6	39.6
Average Number of Passengers per Trip — Non-Revenue	8.5	3.8	3.6
Average Number of Passengers per Trip — Total	28.5	23.4	43.2
Total Number of Trips	533	366	301
Total Delay during Period, min.	213	340	179
Total Delay per Trip, min.	0.4	0.9	0.6
Number of Stops per Day	42	12	25
Car Trips Replaced by Steam Train	3	10	25
Number of Trips per Day	6	4	4
Average Speed, m.p.h.	28	25	20

[a] Operated between Derby, New Haven and New Hartford, Conn.
[b] Operated between Fairhaven and Tremont, Mass.
[c] Operated between Litchfield, Danbury and Waterbury, Conn.

gers Per Trip, the total average of non-revenue and revenue is 28.5 for Car No. 9000, 23.4 for No. 9001 and 43.2 for No. 9002. Those are averages per trip, not the maximum on the car at any one time. Therefore, it will be noticed that two of the cars are handling on the average from 3 to 12 people less than the nominal seating capacity of the car and yet, at times, the cars are badly crowded. The delays per trip are 0.4, 0.9 and 0.6 min. The average speed in miles per hour is shown at the extreme right; in most cases the cars run on an average from 35 to 38 m.p.h., because of the number of stops those cars have to make. I rode in one car at 42 m.p.h., but that is too fast for comfortable riding, so far as engine vibration is concerned.

Figs. 1 and 2 show studies that were made of wind resistance and are not intended to be anything except an attempt to show in a rough way the relation between wind resistance and the different contours of the frontal surface. Fig. 3 shows the horsepowers developed and their relation to the speed with the four-cylinder AC Mack type of engine. The vertical distances measured between the horsepower curves represent the rolling-resistance approximately, and it is important to note the rapid rise in proportionate power resulting from the engine, because of the fact that, in the unit car, the relation between frontal area and weight is very different from what it is in a steam train. The problem of wind resistance is real in unit-car resistance, whereas it is decidedly minor in steam or heavy electric types.

Fig. 4 shows approximately our idea of the floor area and arrangement of the larger car that we feel would cover a very substantial field. That car would seat 15 passengers in the forward compartment, which would be in the rear of the baggage compartment, and the seats would be used by the smokers; in the rear of that there would be seats for 40 other passengers.

Fig. 5 is a cross-section of the car body, showing the lightness of construction that one must get into in designing bodies to keep within the power limitations of gasoline engines. At the same time it would give some insulation to help solve the heating problem and eliminate vibration and noise.

AUTOMOTIVE RAIL-CARS AND THEIR FUTURE DEVELOPMENT

BY L. G. PLANT[1]

THE many improvements effected in gasoline-engine construction during the war for airplane, heavy truck, tractor and tank usage have done much toward making the gasoline-driven rail motor-car a practical possibility to-day.

The gasoline-electric cars built by the General Electric Co. are mentioned and light rail motor-car construction is discussed in general terms. Reliability and low maintenance cost are commented upon briefly, and the requirements of service for rail motor-cars are outlined.

FROM the start, developments in automotive engineering have inspired attempts to adapt the same principles of propulsion to railroad cars. There always has existed a field for equipment of this description, due to the fact that the operation of a steam locomotive and a train of cars involves certain elements of cost that cannot be curtailed in proportion to the size of the train; so, in passenger service, light local steam-trains have been operated at an expense that is excessive in proportion to the revenue received. It is only recently that the most vital factors contributing to the commercial success of the automobile truck have been applied in the construction of self-propelled rail-cars, but within the past year developments in this field have been moving rapidly toward a type of self-propelled car that can be substituted successfully for a steam-train in certain classes of service. No recent development in railroad equipment has aroused such universal interest on the part of manufacturers and nearly all railroad departments within a short space of time. The successful adaptation of automotive principles to railroad cars will, I believe, prove a very great benefit to the railroads in enabling them to reduce the cost of light local passenger-service and increase their gross revenue by augmenting and improving the character of this service.

In view of these circumstances, the question arises as to why previous developments in this direction have not met with more permanent success, and it is still something of a mystery why so obvious a solution of the problem as is found in the modern rail motor-car should not have been discovered earlier. But, before discussing the more fundamental causes that retarded this development, it is pertinent to say that our railroads have never been more severely pressed to devise operating economies than within the past year and have never been more keenly alive to the possibilities of any equipment designed to reduce operating costs. Coincident with this attitude, the situation with the truck builders has also been propitious for progress in this direction.

During the period of the war, self-propelled rail-car construction came to a standstill, and many of the cars previously purchased by the railroads were withdrawn

[1] Associate editor, *The Railway Review*, Chicago.

from service, due to their high maintenance-cost and unreliability in operation; but, while the war apparently retarded development along this line, in reality the many improvements effected in gasoline-engine construction, designed not only for airplane use but for heavy trucks, tractors and tanks, have done much toward making the gasoline rail motor-car a practical possibility to-day. In distinction from the relatively slow-speed heavy engines used originally in the McKeen and Hall-Scott cars, we now have in the heavy-duty truck-type of engine a very much lighter high-speed engine capable of exerting a high torque through a wide range of speeds. This has an important bearing on rail-car construction, not only on account of the reduced overall weight of the engine but because the reduced weight of the reciprocating parts obviates the difficulties occasioned in the earlier cars by the inertia of the heavy engine parts. This will account for some of the difficulties encountered on the earlier types of rail car which, although they may appear somewhat crude in the light of present-day practice, represented a real mechanical achievement at the time of their construction.

This is illustrated best in the construction of the gasoline-electric cars built by the General Electric Co. involving the design of a special gasoline engine with two sets of four cylinders each, forming the V-type arrangement that has since been used extensively and indicating that, at the time these cars were first built, they embodied the most advanced engine construction available. Now, however, the variable-speed characteristics developed in engines of the type now considered for gasoline-rail-car operation, together with the variable speed-ratios in the transmission mechanism employed with these engines, afford an element of flexibility in speed control that obviates the necessity for the interposition of electric drive from the standpoint of speed control; and it is believed that the additional weight and first cost involved in an electric generator and motors preclude the economical use of this form of transmission in rail motor-cars of the type now under consideration.

Light Rail Motor-Car Construction

It is apparent, therefore, that from the standpoint of the motive-power unit, the type of rail motor-car now discussed is fundamentally different from that developed prior to the war and represents a distinct advance over these earlier cars. But while the development of light high-speed gasoline-engines capable of operating continuously under heavy loads has been advancing rapidly, there also has been under way a development, inspired partly by what the automobile builders have accomplished through the use of alloy-steels and also by the trend in street-railway-car construction toward lighter weight, such that the builders of this equipment are now able to design very light cars for operation where the power limitation is severe. One of the most remarkable examples of this construction is a double-truck car-body weighing 11,000 lb. that has seating capacity for 46 passengers. This car is 42 ft. long and has a baggage compartment. Broadly speaking, therefore, it can be said that the modern rail motor-car is the embodiment of an improved powerplant and refinement in car construction.

While it is believed that in the numerous designs of self-propelled rail-car in service or under construction there is available to any railroad a type that it would be justified in buying at present, it is admitted that the design of these cars is still in a progressive state principally with respect to details in construction that will insure their reliability and low cost from a maintenance standpoint, and also with respect to increasing the capacity of the equipment. It is not possible to determine from the figures now available the largest number of passengers that can be handled more economically in self-propelled rail-cars than in a steam-train and, of course, this figure would depend upon local conditions; but it is safe to say that under ordinary circumstances, the operation of self-propelled cars with as many as 80 passengers would show a considerable saving over a steam-train carrying the same number of passengers.

Although the question of using trailers or a single large car would need to be decided in this connection, this is not regarded as fundamental to the solution of the problem that, in reality, lies in the design of a motive-power unit of sufficient capacity without sacrificing any of the characteristic features in commercially available types. From a theoretical standpoint, the use of a motor car and trailers in place of a single large car will increase the frictional resistance and dead-weight per passenger slightly; but, practically, operating conditions peculiar to the railroad on which this equipment is operated will prove the determining factor, so that it would be a mistake for any manufacturer who is looking toward the development of greater carrying-capacity in this type of equipment either to depend entirely upon the use of trailers or to commit himself to a design that would preclude the use of trailers.

The application of more power to self-propelled cars presents a real problem, since there are few commercially available engines of the type adapted to this service that exceed 60 hp. at normal speeds. It is in this connection that the unit steam-car has an unique advantage, since it is capable of developing as much as 300 hp. with a flash type of boiler. While it is understood that the unit steam-car has some very special advantages in connection with the subject of self-propelled cars, it is recognized that this type has reached a more advanced stage in relation to its ultimate development than the gasoline-engine car, so that further discussion of this subject will be confined to the latter type which must still be regarded as being in a formative stage.

With gasoline engines of 60 hp., the best that can be anticipated appears to be a car that will seat approximately 40 passengers, carry baggage and operate normally at a speed of about 40 m.p.h. To effect any considerable increase in the size of this car or render it capable of pulling a trailer at the speeds required in main-line service will necessitate more power, either through the use of a larger engine, which ordinarily involves special and expensive construction, or the use of two engines, which involves certain special problems in their control. Assuming that it were practicable to design an individual transmission of sufficient flexibility to enable the simultaneous operation of both engines, and that it were possible to control the operation of these engines satisfactorily, the use of two 60-hp. engines would have a theoretical advantage over a single engine of larger capacity since, whenever the power requirements dropped to the capacity of one engine, it would be possible to run a single engine at full capacity and thus realize more efficient operation than when a larger engine is operated at a fraction of its capacity. The gasoline consumption of a rail motor-car seating 40 persons and carrying baggage will approximate 0.2 gal. per mile, and it will be desirable to maintain a proportionally economical rate in larger cars. Another factor that should encourage development in the direction of using two engines is the element of reliability afforded by two independent driving engines since, ordinarily, one engine will continue to operate

should the other fail. Probably no other factor proved so discouraging to the successful use of both the McKeen and the General Electric Co. cars than engine failures encountered in the operation of this equipment.

Reliability and Low Maintenance Cost

Reliability and low cost from a maintenance standpoint as already referred to undoubtedly constitute the most difficult problems with which designers of this equipment will have to contend, since they are matters in which the railroads are most exacting. Maintenance of cars and locomotives already costs the railroads as much as either train wages or locomotive fuel, and any failures in train service add to this expense and the difficulty of operation. The problem is complicated further by the fact that equipment of the type under consideration often could be operated to the best advantage between points that are isolated from shop facilities. While the substitution of a hard and smooth rail would seem to facilitate the adaptation of automotive equipment to rail service, the absence of that element of flexibility afforded by a resilient tire operating over the ordinary highway surface in reality makes the adaptation of automotive equipment to rail service a more difficult problem. Not only do uneven joints and cross-overs introduce more severe vertical shocks, but abrupt changes in the alignment of the rail, as at switches and on curves, cause far more severe lateral shocks than are ever encountered in highway service. Moreover, the absence of any element of elasticity between the engine and the driving-wheel tread, as provided in automobile construction by a resilient tire, operates against the efficiency of the gasoline engine that can be operated only to the best advantage when the transmission is capable of absorbing the ordinary pulsations of the engine and cushioning the shocks occasioned by any abrupt variation in the speed between the engine and the wheel tread.

For these reasons it is believed that the most successful development in this class of equipment will tend toward standards in truck, axle and wheel construction that many years of railroad service have demonstrated as safe and economical; that the use of rotating axles mounted in special journal boxes fitted with frictionless bearings will become general; and that the most desirable form of transmission will prove to be one that is flexible with respect to the vertical and lateral blows transmitted through the driving-wheels. But in whatever development work that is undertaken, either in this direction or with respect to enlarged engine capacity, it is safe to say that the most successful results will be obtained where the experimental work is conducted in conjunction with the railroads.

Finally, in discussing the future of the self-propelled rail-car, the question of greatest importance to the manufacturer and of some moment to the railroads as it may affect the cost of this equipment, is the matter of production which, in turn, depends upon the prospective field for this equipment. It is not unreasonable to assume that in short-line railroad-service, wherever it is possible to dissociate freight-car movement from passenger service, the gasoline-driven rail motor-car will supersede the steam passenger-train eventually. The rate at which this transformation will take place will depend not only upon the finances of the railroad, but upon the attitude of the manufacturer toward financing this purchase. Reliable figures are available to show that wherever cars of this description have been operated by the independent short-lines, they have reduced the cost of operation in comparison with steam service; and wherever the road was incurring a loss with steam service, the gasoline rail motor-car enabled it to make a net profit. Moreover, there are now available several types of car admirably adapted to any class of passenger service ordinarily operated by the short-line railroads.

Trunk-Line Requirements

Turning to the trunk-line railroads, the question cannot be answered as easily; first, because the railroads do not themselves know the extent to which self-propelled rail-cars can be substituted profitably for steam service and, second, because types designed to carry more than 50 passengers at the desired speeds are not yet available.

It is safe to say that there is an immediate field for possibly 1500 cars of the types already available. It is understood, of course, that many railroads will want to delay the purchase of these cars pending the development of new types, while others will want to observe the operation of this equipment on other railroads before committing themselves. Also, the question of financing the purchase of these cars will come up for consideration and it is reasonable to believe that, once a depreciation rate on this type of equipment has been determined reliably, some form of equipment trust applying to a number of these cars would facilitate their purchase. Altogether, the development is yet so new that it may be several years before we can look for any volume of purchases that mean the production of these cars on a large scale, despite the fact that a great field of usefulness awaits them.

THE DISCUSSION

R. B. ABBOTT[4]:—I am chairman of a committee of the Philadelphia & Reading Railway Co. on the question of the rail motor-car for use on branch lines. After we determine what type is best suited to our purpose, we will then make a study of all cars that approximate this particular type.

A car to suit our purpose must be capable of usage as a motor car with a trailer or, perhaps, another car in addition to the trailer, because the demands on our branches vary so on different days in the week and for different times of the day that a single unit car would not of itself solve many of our problems. The car must also be reasonably comfortable and not noisy or ill-smelling. It should, we think, be capable of control from either end, so that it will not be necessary to turn the equipment at the end of each run.

G. C. HECKER[5]:—When the electric railway people first adopted the light-weight cars, they worked on the theory that they would not replace the seats, seat for seat; that is, they would not attempt to give exactly the same service that had been given previously with large double-truck cars but, rather, increase the service considerably by running cars on closer headway. The people in the different communities found, after they overcame their first dislike for these cars that, perhaps, were not so comfortable as the large double-truck cars, that they were really getting much better service. On a branch line of a steam railroad where very infrequent service is now being given by steam operation, it might be possible, with the gasoline rail-motor-car operating in single-unit light-weight cars, to give a very much improved service. I believe the public can be made to realize that they will get very much better service if they will put up with a little less comfortable car. As the gasoline rail motor-

[4] Assistant general superintendent, Philadelphia & Reading Railway Co., Reading, Pa.

[5] Special engineer, American Electric Railway Association, New York City.

car is developed, there unquestionably will be many refinements that will reduce the objections of the riding public to this form of transportation.

J. E. BURRELL[6]: — The Pennsylvania Railroad has a committee that has been investigating the various types of rail motor-car that are in service at a number of points. The company does not operate any cars of this type on its lines. One car, however, is operated on a branch line by another company. The car is similar to those used by the New York, New Haven & Hartford Railroad, and it has been giving very good satisfaction. We are, of course, somewhat in the same position as the Philadelphia & Reading; we are trying to find the car that will suit our purpose best and, after ascertaining what car that is, we probably will install it on the line.

ARTHUR J. SCAIFE:—It is our understanding that the great need to-day is for some kind of a combination baggage and passenger car that will take the place of the present equipment used on many short-line railroads where it is necessary to use a passenger car, a baggage car and a locomotive, with a full train-crew. This equipment cannot be operated without a loss and the company usually runs one train a day because it is required to do so.

Very little work has been done on rail motor-car equipment by our company, and that has been only within the last few years. We are trying to find out first just what the requirements are with reference to seating capacity. It will be necessary to go at this proposition with an open mind. The thing that we have run up against is that men have been thinking in railroad terms. They immediately criticize a rail motor-car job and ask how it compares with the present railroad equipment and Master Car Builders' standards. If the automotive railcar builders and the railroad operators go at this problem with open minds, I believe that something can be accomplished.

L. G. NILSON:—The present International car, which has a good appearance and is doing very well, naturally has the earmarks of the ordinary motor-truck. I believe that when we consider the larger sizes that the railroads undoubtedly want, we will come back to something like the McKeen car; that is, the power and transmission, the whole drive and equipment should be on the forward truck or at least on one truck. The car body proper could be made very light, with a light trailer, arranged so that it could be uncoupled in a very few moments. In that way the power unit could be gone over at regular intervals once a week, or even inspected once a day, and it would not be necessary to tie up the car body. The car bodies could be run 24 hr. per day or as long as desired.

I believe we will find that a driving unit of this kind equipped with spur gears will give better satisfaction than one equipped with bevel gears. The bevel gears are doing very well in ordinary sizes, but the use of too large sizes causes many difficulties, not so much on account of the gears as on account of the mounting. Unless the mounting and the housings are very rigid, the teeth simply tear themselves out; with spur gears, there is less trouble of that kind.

I would like to predict that we will see an internal-combustion engine almost as elastic as a steam engine in its action, possibly within 3 years. Then the problem of transmission and control will become very much easier.

HENRI G. CHATAIN:—Some 20 years ago, Mr. McKeen induced E. H. Harriman to invest some money in the construction of a rail motor-car that had a mechanical drive. At about the same time I persuaded the General Electric Co. to engage in the construction of gasoline-electric rail motor-cars. Mr. McKeen built some 150 to 200 rail motor-cars and, I believe, they are in successful operation to-day. The General Electric Co. built 100 rail motor-cars and approximately 98 are in operation to-day.

I have listened with great interest to the gentlemen who have spoken in regard to the number of miles that the newer types of car are making. It is interesting to know, and I think the information is correct, that one of our cars recently completed 1,000,000 miles in service, and there are a number of them that have gone over 600,000 miles.

At the time we began to build rail motor-cars, if we had had superintendents on the railroads who would listen to arguments in favor of light weight and not insist upon having many things hitched to the car that belonged to the steam locomotive, we would have built small light cars a number of years ago. But the railroad representatives could not agree with such a viewpoint, and I am not sure that they can agree to-day. Each superintendent wants a different kind of car. Some want trailers; some want to go faster, and others want to go slower. All of this involves differences in design and attendant high cost of production.

Mr. Bean studied the proposition of the light-weight rail motor-car and has convinced me of its merits. He is willing to do without couplers and many other things, provided he gets a good and safe rail motor-car.

I am not an advocate of the mechanical drive. I followed the McKeen rail motor-cars very closely and have the facts and figures covering thousands of miles of their operation and also similar data for the cars built by the General Electric Co., covering a comparable number of miles of operation. The mechanically driven car will operate on less gasoline per mile, but it will cost more for maintenance and repair, because it does not possess what I like to call "squashiness." It is not an automobile with rubber tires, but runs on rails that are not flexible from the transmission viewpoint. No engineer has yet developed ways and means of attaining suitable flexibility between the engine and the track, and this is the important factor so far as the upkeep is concerned.

The gasoline-electric drive has four points of advantage:

(1) The engine can be loaded at all times. As the gas engine is governed by changing its compression, it is obvious that if it can be loaded properly at practically all speeds, and all through its operating range the efficiency can be increased by increasing its average working compression
(2) It is more flexible. The speed changes blend from one to the other because of the nature of the electrical units employed
(3) The prime-mover can be operated at a speed below its normal rate and yet maintain a high car-speed. The high engine-speed is maintained during the accelerating period and for grade work, but is reduced during the period of free running on the level or down a slight grade. These conditions can be well taken care of by the gasoline-electric drive
(4) It makes possible double end-control

I will not take any very decided stand on whether the transmission should be capable of a complete conversion of energy or not. I think that there are a number of transmissions that do not completely convert the energy. They possess all the desirable features such as the flexibility, loading and the various speeds of operation of the

[6] Superintendent of passenger transportation, Pennsylvania System, Eastern region, Philadelphia.

engine, with but small losses of energy. The installation of an engine in a rail motor-car is an extremely difficult thing. It cost us a large sum of money before we found out how to do it. A combination of felt and springs seems to be the most effective; we are using it and it has been reasonably successful.

We built eight-cylinder engines, but they are not as desirable as those of six cylinders or multiples thereof.

The position of the exhaust is an important matter. Mr. Bean points out that there must be some means of preventing gaseous odors in the car, because they are very disagreeable to passengers. We tried every conceivable position for an exhaust and found that the best place to put it is directly overhead, with not too much muffler, so that the gases will go up as high as possible due to their velocity. Exhaust at the rear of the car is prone to roll up and come in through the back windows.

The greatest need in the rail motor-car field to-day, to make it an economic as well as a manufacturing proposition and therefore desirable for both the manufacturer and the user, is the standardization of requirements. This probably can best be brought about by the Society working in conjunction with railroad representatives of authority.

CHARLES O. GUERNSEY:—The gasoline-propelled motor-coach equipment should be only of such size as can be operated with commercially proved engines and handled by a crew of two men. With larger engines that have not been proved out in severe duty and under commercial conditions, we may get into some mechanical difficulty. For cars larger than can be handled by two men, the saving in cost will not be sufficient as compared to steam equipment to justify the use of gasoline-propelled cars. Like any other broad statement, this is undoubtedly subject to some limitation.

Generally speaking, the gasoline engine above about 5-in. cylinder-bore has not been proved in commercial automotive service. It is true that large engines have been used in various installations, such as aircraft or private yachts, but for such service the first cost, operating cost and maintenance are not of prime importance. Engines of this type would not be satisfactory in motor-coach service. If we assume then a 5-in. cylinder as being the maximum that can be used safely, we are confronted immediately with a limitation of horsepower dependent upon the number of cylinders that are used. Four-cylinder engines of this size have been well proved and undoubtedly will be successful in this service. It is possible that some six-cylinder designs which are now on the market may also be successful. For larger powers, nothing has been developed as yet, and I doubt whether there will be a sufficient demand to justify the development of 8 or 12-cylinder engines for this service, to say nothing of the complications incident to such a multi-cylinder design. If the foregoing assumptions are correct, we are limited in the case of a four-cylinder engine to about 70 hp. as the maximum that is available; or, in case we accept the six-cylinder design of about the same cylinder bore, we can expect to get about 100 hp.

The designer of these cars should bear in mind that the car must represent a combination of automotive and railroad practice. The railroad standards as to safety, comfort, steadiness of riding and low cost of maintenance and operation must obtain. Because of the limited power available, the weight must be kept to a minimum. This indicates, therefore, the use of alloy-steels, light-weight designs, anti-friction bearings and the like, as customarily used in automotive practice and as already proved in such service. The weight of car that can be handled satisfactorily with the engines of the powers mentioned will depend, of course, upon the speed required, the road conditions, the number of stops and the acceleration that must be had.

In general, it is my opinion that, with the four-cylinder engine, the outside limit of weight for general all-around satisfactory performance is about 18 tons loaded, and for the six-cylinder engine the outside weight should not exceed from 22 to 25 tons. In a properly designed car the four-cylinder engine should handle about 45 passengers in combination with a baggage or express load of about 1 ton satisfactorily. The six-cylinder engine probably would handle a passenger load of from 55 to 60 passengers in combination with about 2 or 3 tons of baggage.

Demonstrations of a four-cylinder railroad motor-coach developing 61 hp. show the results given in Table 2.

TABLE 2—ACCELERATION OF A FOUR-CYLINDER RAILROAD MOTOR-COACH[7]

Acceleration from a Standing Start to Speed, M.P.H.	Time Min.	Time Sec.
25	..	30
29	1	..
35	2	..
41	2	40

[7] Light weight of car, 13 tons; loaded weight, 17 tons.

The gasoline consumption varies from 5.2 to 7 miles per gal., depending upon the conditions. The normal speed of the coach at the rated speed of the engine is 35 m.p.h. and the maximum speed with full load is 48 m.p.h. The operating cost, including a crew of two men at standard wages, gasoline, oil, maintenance, depreciation and interest on the investment, is about 29 cents per mile. This figure will, of course, vary with the local conditions.

W. G. BESLER[8]:—In my opinion, there are certain places where a gasoline-propelled vehicle finds its proper application in railroad service, but in those cases where a cement highway costing from $40,000 to as high as $120,000 and in some cases even more per mile, is constructed at public expense, paralleling a railroad, why should branch-line service, which is the only place where a gasoline rail-car finds its proper use, he continued? In such an instance the railroad company had better stop operations, invest its money in motorbuses and continue service upon a highway provided for it free of expense, than subject itself to the burdens of expense for rails, which require renewal, maintenance, supervision and all that goes with railroad service.

GEORGE L. SHINN[9]:—Our designation of the practical application of the gasoline-driven rail-car at present is that it can be used with great advantage for light traffic conditions. I say this because on our road we have substituted a White combination passenger-and-baggage car where we formerly used a steam locomotive and two passenger coaches to handle the traffic. We have not yet arrived at a definite figure but a conservative estimate indicates that, by the use of this gasoline-driven car, we will effect a saving of $15,000 per year. We believe that, by the use of this gasoline-driven rail-car, we are giving service superior to that formerly rendered, and we note from the expressions of opinion that have reached us that the patrons on the line are much better satisfied. We maintain the same schedule as when the steam trains

[8] President, The Central Railroad Co. of New Jersey, New York City.

[9] President, Pennsylvania & Atlantic Railroad, New Egypt, N. J.

were in operation and find that we could give even greater service should it be necessary and desirable.

The car that we have in operation is governed for a maximum speed of 33 m.p.h. and, due to its excellent acceleration and easy handling, we are maintaining the former steam-train schedule without difficulty and could, if desired, stiffen this schedule. We have no hesitancy in saying that our experience with the gasoline-driven rail-car is very satisfactory in every way.

J. W. CAIN[19]:—In making our investigation of gasoline-propelled rail motor-cars for the member lines of the American Short Line Railroad Association, we did not go into the subject technically but, instead, considered the different cars more from the standpoint of practicability as evidenced by actual service. We approached the subject from three different angles and our final conclusion was based on a summation of the information thus received.

Our membership consists of some 500 different railroads located throughout the United States, and for a great many years they have been the proving grounds for the different rail motor-cars brought forth. Indeed, there has seldom been a gasoline-propelled railroad-car built that has not at some time or other found its way to one of these properties. We, therefore, had a source of extremely valuable information and sent to each of these lines a questionnaire, of which the following are the principal questions:

(1) Are you using motor equipment on your line, and if so what make?
(2) How long has it been in service?
(3) What is your average operating cost per train mile?
(4) Approximately what mileage do you get per gallon of gasoline?
(5) Do you experience any trouble from slippage in rainy or snowy weather?
(6) Have you had any serious trouble from derailments on curves?
(7) Of the different commercial designs now on the market, which do you consider the most satisfactory?
(8) Do you expect to be in the market for rail motor-car equipment in the near future and, if so, what equipment will you need?
(9) Give us your suggestions as to the necessary compartments and toilets as suggested by the demands of your service or required by your State Railroad Commission

There was a most gratifying response, indicating great interest in the subject, and from these answers we were able to arrive at certain definite conclusions.

We have a used-equipment department in the Association, which is a sort of clearing-house among our member lines as well as some of the trunk lines, and from this we secured a tabulated list of all the rail motor-cars offered for sale. This threw a most interesting spot-light on the entire subject.

We made a personal inspection of the most successful cars available and spent a great amount of time in making demonstrations and in looking over the manufacturing facilities of the firms proposing to build them. There were some cars offered that we did not examine because we considered them impractical or not soundly financed.

Summing up these three phases of our investigation, we found that the most successful cars in service were those of light design, using a thoroughly tried and proved make of motor-truck power-unit or chassis. One make in particular showed a preponderance over all others in the ratio of probably 5 to 1. We were furnished records of cars that had made as high as 300,000 miles, and been in practically continuous service for a period of 5 years. The operating cost varied from 10 to 25 cents per mile, and the gasoline consumption from 5 to 10 miles per gal.

We found the maintenance cost surprisingly low, averaging about $15 per month on these smaller-type cars and only slightly above this on the larger ones. By smaller type I mean those using a 2½ or 3-ton motor-truck chassis, and by larger cars those using 5-ton chassis. The operating cost of 10 cents per car-mile was, of course, confined to the former, which were being operated by one man. But some of the larger types using two men were being operated as low as 20 cents per car mile, as given in Table 3.

The figures in Table 3 were made on a basis of $12,500, the purchase price of the car, and an operation of 100 miles per day.

TABLE 3—COST OF RAIL MOTOR-CAR OPERATION

	Cost per Mile
Gasoline	$0.030
Labor, two men at $125 per month	0.085
Depreciation, rate 12½ per cent	0.042
Interest and Insurance	0.022
Maintenance	0.021
	$0.200

It was revealed that the majority of cars on the short lines were being operated most successfully by younger men, who were trained as mechanics and who were thus able to take care of practically all of the necessary light repairs. I think this point should be emphasized strongly, as the labor cost is one of the principal single items in the operation of rail motor-cars, and the payment of standard wages would defeat the object to be accomplished. At least this is true on the short lines. We take the position that the operation of these cars does not require the skill or training necessary to operate a steam locomotive. They have never been classified by the Interstate Commerce Commission, and I feel sure that our position would be upheld.

The consensus of opinion was that all cars should be equipped with a pivotal lead truck for safety and that a single pair of drivers with the proper weight distribution gave satisfactory service, though the riding qualities of the car were naturally not as good as though a four-wheel truck with swing bolster were employed.

The different rail motor-cars offered through our clearing-house revealed that practically every road owning the old heavy and now obsolete types, some of which are not now being built, desired to sell them at prices ranging from $500 up. Of all of the cars offered, however, there was not a single one of the modern light adapted truck type.

In our personal examination and inspection of the different cars offered, we found that the light six-wheel type cars up to a length of 36 ft. and a weight of about 20,000 lb. could be operated successfully at a speed of from 30 to 35 m.p.h., making from 5 to 6 miles per gal. of gasoline. Beyond this, there is too much vibration, and the single-driving-wheel arrangement makes the car ride uncomfortably; but for a capacity up to 35 passengers and about 2000 lb. of baggage, we found this the most successful car of the present time. Above this capacity, we examined a car 43 ft. in length, equipped with two four-wheel

[19] Manager of purchases, American Short Line Railroad Association, City of Washington.

pivotal trucks that was capable of making a maximum speed of slightly better than 40 m.p.h., at which speed it rode very comfortably. The weight of this car was about 30,000 lb.

In conclusion, I believe that these cars will prove the salvation of many short-line railroads, as well as the branch lines of the larger systems; and, as a large number of our member lines have stated, they have changed their figures from red to black. While the cars that we have been discussing are absolutely successful and will faithfully perform the duties imposed on them, I feel that efforts should be expended toward the development of a higher-powered engine, as the present ones have none to much power. I do not mean to increase the bore of the cylinders or go to the slow-speed marine-type of engine; but, instead, to increase the number of cylinders and adhere strictly to the successful and proved type of automobile engine.

HOT-SPOT METHOD OF HEAVY-FUEL PREPARATION

(*Concluded from p.* 476)

lutely preventing crankcase and cylinder-wall lubricant-dilution.

Successful application of the separating hot-spot demands only an ordinary knowledge of the laws of physics relating to heat, and presents much less difficulty than a number of other problems that our automotive engineers have solved. It is only a question of getting an adequate heat-supply from the exhaust and of excluding heat communication from other sources. As suggested in the foregoing, the entrance-air temperature should be the lowest that can be obtained, and care should be taken to avoid conduction of heat to the intake system beyond the hot-spot. In particular, careful attention must be given to the heat insulation between the heating surface and the remainder of the enclosing walls of the intake system, as a tremendous amount of heat can be conducted across the ordinary flange-joint.

Our experience indicates that the separating hot-spot should always be located in the main exhaust line and that it is hopeless to attempt to pipe the exhaust across a T-head engine, or the like, as the temperature drop will result in too low an exhaust temperature at the lower speeds. Also, the actual flow of exhaust to the hot-spot will be a function of the muffler back-pressure, which will result in exaggerated temperatures of the mixture at high car-speeds.

DURALUMIN

(*Concluded from p.* 480)

and length to do more than describe the physical and metallurgical properties of duralumin and touch on some of the applications of interest to the Society. However, the automobile engineer should realize that now he can avail himself commercially of this extraordinarily light yet strong material as the aeronautical engineer has done. The development of the Zeppelin was dependent upon the development of duralumin, and the rapid progress of the all-metal airplanes since the war has been due largely to the commercial availability of the same metal. While the automobile is not as dependent as the airship or airplane upon a material of this class, nevertheless modern automobile design tends toward the elimination of unnecessary weight. The extensive introduction of duralumin into automobile construction, especially in unsprung and reciprocating parts, will permit of a complete redesign, effecting economies that will offset its greater cost as compared with steel. Such a car will bring nearer the ideal combination of road performance and economy of upkeep and operation.

Oil Consumption

By A. A. BULL[1]

SEMI-ANNUAL MEETING AND DETROIT SECTION PAPER

THE subject of oil consumption was discussed at the meeting of the Detroit Section held Sept. 29, 1922, following the presentation of the paper bearing this title that was prepared by A. A. Bull for the 1922 Semi-Annual Meeting of the Society. An abstract of Mr. Bull's paper precedes the discussion. Members who desire to refer to the complete text as originally printed and the illustrations that appeared in connection therewith will find these in the June 1922 issue of THE JOURNAL.

ABSTRACT

THE object of the paper is to consider some of the fundamental factors that affect oil consumption; it does not dwell upon the differences between lubricating systems. Beyond the fact that different oils apparently affect the oil consumption and that there is a definite relation between viscosity and oil consumption, the effect of the physical characteristics, or the quality of the oil, does not receive particular attention.

The methods of testing are described and the subject is divided into (*a*) the controlling influence of the pistons, rings and cylinders; (*b*) the controlling influence of the source from which the oil is delivered to the cylinder wall. The subject is treated under headings that include the piston-ring; the effects of oil-return holes, side-clearance and ring motion; thin rings; influence of piston fit; efficiency of the scraper-ring; ring and cylinder contact; carbonization and spark-plug fouling; oil-supply control; influence of oil viscosity; effects of dilution; external oil-leaks and breather discharge; and influence of controlling lubrication in proportion to throttle opening.

THE DISCUSSION

A. L. CLAYDEN:—I have devoted much thought to scraper-ring action, because there seems to be so much difference of opinion about it. It appears possible to me that in some cases it would work effectively and in others be actually detrimental. The ring mentioned is arranged at the bottom of the skirt so that it runs half out of the bore. It is useless unless it does run half-out, because there is no space for the oil it drives before it to escape. I wish to emphasize that on the down-stroke of the piston the scraping action of the lower one of the several compression rings builds up a very high pressure in the oil-film that is being scraped before it; so, even if the cylinder film is not any too complete, that scraping is enough to insure that the space behind the ring will be completely filled with oil. The depth of the space behind the ring is probably not a very important function, with the amount that is pumped, because if you regard the piston-ring as a pump piston or as a valve, it will only pass the quadrant, or 90-deg. position, of its stroke. But one can be sure it will be fed full, all it can take; hence the extreme value of the relief holes or grooves immediately beneath the ring and the great value of the ring with sidewise expansion; by "sidewise" I mean fitting the groove expansion.

The crank drilling that Mr. Bull mentioned is probably one of the most important things to study. To some slight extent, a study of the development of various aviation engines was made. It was found to have a very profound effect upon the oil consumption and on the lubrication of all parts of the engine. In fact, all kinds of changes could be made by simply moving the hole around the pin. In many of certain new engines, we finally hit upon a 90-deg. position as giving the desired results; but, of course, a very considerable quantity of oil must circulate for the purpose of cooling.

O. C. FUNDERBURK:—I was connected with the designs of some marine engines ranging in cylinder size from 6¼ to 7¾ in. and developing from 300 to 450 hp. In the multiple-bearing crankshaft we used, it was necessary on account of the high powers to carry oil pressures in each individual crank control. That is, we would have to cut-off the communication all of the way through the shaft, as is apparent in the Liberty engines because of this long crank-arm. We run this 7¼ by 9-in. engine at 1650 r.p.m. The centrifugal force and the load constitute very large factors in the distribution of the oil, and cause over-oiling in the particular cylinder that has the loosest journal and crankpin bearing. We found it necessary to put a plug in the main-bearing journal of the crankshaft and to make individual crankpins for each cylinder. That greatly decreased the over-oiling in any cylinder where there were loose bearings in proportion to the adjacent cylinder. We found also, as Mr. Bull did, the necessity for moving the oil-hole from the outer position on the crankpin to an inward position. The position we use is 10 deg. from the underneath position. That, we found, made a great difference in the distribution of oil. We also found a great change in consumption due to pressure. We have experimented with pressures of from 10 to 250 lb. per sq. in. We found that a pressure of about 30 lb. per sq. in. gives the highest horsepower with the best oil-consumption. The excess power required to drive the pumps at the exceedingly high pressure and the cooling of the oil absorb sufficient power to note the difference on the dynamometer. We obtained the best oil-consumption in our engines when the oil had a temperature of about 108 deg. fahr. The temperature sometimes ranged from 100 to 120 deg. fahr. with Mobil-Oil B, and Veedol extra-heavy. The temperature of the oil ran as high as 150 deg. fahr. and there was a marked increase in the consumption. If the oil was colder than 80 deg. fahr., we found an excessive amount of oil on the spark-plugs. This proved very conclusively Mr. Bull's statement that the film of oil on the piston, if three rings on the top and one scraper ring are used, could not be discharged on the ring, and the unit pressure between those upper and lower rings would discharge it on the piston. As soon as we got temperatures of the oil as low as 50 deg. fahr. the fouling of the spark-plugs with over-oiling became very much more apparent.

T. J. LITLE, JR.:—I am not a believer in the velvet, or rough, finish on the cylinder bore that some companies have practised, and the rough rings. At least one engine company in this Country has practised finishing the bore, grinding it four times and honing the surface. The final honing operation is done with a stone of very fine grade, leaving the cylinder wall in a polished condition. Many of us have looked into engines after they have been driven several thousand miles and greatly admired the boring. That is the way it should be done at first, and

[1] M.S.A.E.—Chief engineer, Northway Motor & Mfg. Co., Detroit.

the pistons and rings should not be used as laps to do it.

A thick piston-head is necessary if a cast-iron piston is to operate very satisfactorily in a passenger car. Unless a thin-headed piston is used, the piston is heavy and the engine vibrates excessively. Therefore, I do not believe in using cast-iron pistons.

There has been a very great development in aluminum pistons recently in this Country. I refer to the aluminum-alloy pistons containing about 10 per cent copper, and heat-treated to increase the hardness from 75 to 80 up to 175, almost as hard as cast iron. That has an indirect bearing on this whole problem, because when a light piston of great hardness is produced, the ring grooves will not wear. It is when the ring grooves wear that the engine starts to pump oil excessively.

We all know that, if the depth of the ring is increased, its life is increased, for it will not wear the groove wide so quickly. Many companies are careless in fitting the ring in the groove. Some rings are tight and some are loose at the start, right from the factory. The inertia of the ring hammers the groove wide, particularly where the former does not fit tight.

The construction of the ring itself affects oil consumption most. A plain one-piece ring that has a real scraping-edge on the bottom will control the oil-flow. I do not mean the conventional 90-deg. corner, but that if the ring is cut in at an angle and a scraping edge is established at the lower side of the ring, the oil consumption will be changed wonderfully. On a given engine that is consuming oil at the rate of 1 gal. per 300 to 400 miles, the oil consumption can be decreased to 1 gal. per 2000 miles by simply modifying or sharpening the scraping edge of the bottom of all of the rings. The scraper ring does the most good when a line is cut under the edge of the piston. In other words, if the piston is right up to size, if the liner under the piston ring, the lower ring, is the same size as the rest of the skirt, it does little good. But if that point is cut under, its efficiency is increased greatly.

If a little gash is cut in the lower part of the ring at rather an acute angle the scraping effect is very marked. It is just like that of the ring itself, and it does not provide any oil-holes in through the cylinder or employ these in every ring. We experimented on the dynamometer about a month to determine this angle. We varied it every 5 deg. and the difference in the scraping effect was remarkable. With an acute angle I think there is a certain flexing of the edge. In other words, it acts like a chisel going down an oil-stone; you can scrape all of the oil ahead of it, and leave the oil-stone dry. But if a plain piece of metal with a square edge is put down there, it will leave a film of oil.

CHAIRMAN GEORGE E. GODDARD:—I think that the great advantage of that angle feature is that it provides an edge which sharpens itself.

MR. LITLE:—I believe that the greatest scraping, of course, is on the lower ring below which there is a gash right through to the interior, but no small holes. We had difficulty in getting enough oil through the holes. It is an expensive operation to cut a little slot around the skirt and drill a number of little holes in it. A gash cut right through is, I think, the best construction. It is used very largely on the aluminum-alloy piston and is very effective.

MR. FUNDERBURK:—In connection with the discharge of oil from the rings through the holes, we have had the experience Mr. Litle relates. We started with 12 No. 40 drilled holes below the scraper hole. That was the third ring from the top. We have increased the size of those holes, and finally we have now 22 holes of 3/16-in. diameter, which, of course, is greatly in excess of the area of the annulus represented by the clearance between the cylinder and the piston.

In a design we prepared for the Government we inaugurated some experiments in which we ran the shaft on the same bearing pressure as when the engine was on full power. We found it necessary in connection with the oil pressure to use a refrigerating system to cool the oil, so that we could lubricate at very high speed without scoring; the bearing is 5¼ in. in diameter and runs at 1400 r.p.m., which is far above anything in my experience which had been undertaken in our line before.

J. E. WHITE:—We find in many cases that, if some of the so-called heavy oils are used, the engine cannot be cranked at more than 30 to 40 r.p.m. If an oil that has the proper base is used in winter, one can get as high as 50 or 60 r.p.m. I am speaking of the Packard, the Lincoln, the Cadillac and the Lafayette types of engine.

L. M. WOOLSON:—So far as the minimizing of oil consumption goes, we have found that we can get practically anything we want. We can get an engine to run 5000 miles per gal. of oil if we so desire, or we can do even better than that. But the fact of the matter is that an allowance must be made for sufficient oil-consumption so that the average owner will maintain his oil supply. We have had owners proudly boast about running 10,000 miles and not using a drop of oil. To avoid that condition, we have fixed our engines so that they will use some oil. That is the only possible way of getting fresh oil. We must have fresh oil until we get this dilution problem solved. I think that none of us is really working on this dilution problem as seriously as it deserves. There are two general ways of keeping the oil consumption down, but I think we cannot use either of them until we get the dilution problem solved because a job that will run 5000 to 6000 miles per gal. of oil in the course of the use of a car by the average owner, will be ruined in fairly short order, especially in the cold winter months.

To my mind there are two ways in which we can really control the oil consumption best; by baffles and by scraper-ring construction. We used a scraper-ring construction on high-compression engines with very great success. If we go too low with the oil-consumption we find we get into trouble from hot pistons and hot exhaust-valves. In high-compression aviation-engines some oil must be passed into the combustion-chamber to keep the exhaust-valves cool.

The ordinary type of baffle consists of a slotted plate fitting the cylinder bottom, the rod working through the slot. A baffle like that is worse than useless, because of the high velocity of air going by the slot which results in carrying a great quantity of oil with it, and the result is that instead of decreasing the oil supply it is generally increased. If the baffles are arranged in the form of semi-circular guards that have the crankshaft center as their center and these guards are extended over the ends of the connecting-rod bearing, the oil can be practically prevented from reaching the cylinder. I do not know why that is not a very much better way than trying to get extremely accurate fits between the piston-rings and the cylinder and the ring grooves, which cannot possibly be maintained during thousands and thousands of miles of travel.

We have been taught for many years that the place to feed the oil is on the slack side of the bearing; yet Mr. Bull tells us that the place to feed it is on the tight side or, in other words, so that the bearing will plug up the hole. I have had some intensely practical experience

with just that construction which enables me to make such a positive statement that, from a bearing standpoint, it is positively the worst thing one can do.

We had a 600-hp. aviation engine on a 50-hr. test. At the end of 25 hr. we found the bearings in pretty bad shape. We then went through this same analysis of bearing pressure that has been discussed and found just what Mr. Bull found. Therefore, we decided to put the hole just where he said we must not put it, on the slack side, so that we would surely get plenty of oil there. We ran that engine another 25 hr. with new bearings and the new oil-hole location but everything else the same, and those bearings stood up. Since that time we have always located the oil-feed holes on all our jobs at 45 deg. leading, which represents the zone of least pressure. That gives us an ample flow of oil through the bearing, and helps to dissipate the heat. You can trap the dirt just as well by pressing a tube in the oil-hole as long as desired.

F. E. WATTS:—I think that the principal value of Mr. Bull's paper is possibly not in the conclusions he reaches so much as in the method he follows. I believe it is the method that must be followed in working out the oil problem in any engine. He starts with the oil coming up to a point where it can possibly be drained without giving any trouble. Then he follows that oil through all of the various passages and the places it can go to get into the cylinder. I believe that is the way we must lay out any engine. Lay it out on paper and theorize upon it. Possibly, make models and study them as much as you study the engine because, if the engine once gets to running at high speed, so many things happen in it that the different things cannot be segregated.

I believe that the ideal oil-consumption at present is about 1 gal. per 1000 miles. Keeping between that figure and 1 gal. per 400 to 500 miles is doing pretty well. Using large quantities of oil keeps the grit out of the cylinders and the bearings, and the engines last enough longer so; that pays more than the oil costs. It is perfectly easy to study the combustion-chamber and locate some point in it where the spark-plug will keep reasonably clean even if there is a quantity of oil coming onto it.

CHAIRMAN GODDARD:—In some of our experiments for reducing oil consumption we got so low on the oil that we began to get spark knocking. We found that the amount of oil had been cut down so that the carbon deposit we get from the poor gasoline was not moistened. I think that with filtered gasoline we get better results than we do, even to-day, with hot-spots and the like. Our experience is that there must be a little bit of oil in the combustion-chamber to keep the top of the piston moist.

H. S. MCDEWELL:—As some of you may know, in the Navy Liberty engine a scraper with holes is provided. The oil consumption was reduced as a result of that. The 1700-r.p.m. consumption was cut from 14 to between 5 and 7 lb. per hr. In starting that work we went through a rather laborious research to determine what shape that scraper ring should have, and as to whether this feathered ring was sufficiently better to warrant the increased cost of production. Therefore, we devised a hot plate and a block of 1-sq. in. section. One edge was right-angled, another edge was at 16 deg. and the third edge at 30 deg. We provided a load such that the pressure would be 8 lb. per sq. in., which is the rate of pressure on the piston-rings. We then measured the thickness of the oil-film, and scraped this slot across the plate, which was maintained at a temperature that we assumed to be practically that of the oil-film. We found that under the same unit-pressure, while there was a difference in favor of the sharp edge, it was not sufficient to pay for the increased cost of producing such a ring. Consequently, we adopted the small square ring; and it provided ample space into which to scrape the oil and ample drain-holes. The size of the hole we used was 3/16-in. in diameter. We used 14 of them. I think the cause of the failure of holes of very small diameter is the high surface-tension of the oil, so that an oil-film bridges across those holes and offers too much resistance to the flow of oil draining out to the other side of the piston.

Particularly in the aluminum-alloy piston with cast-iron rings the thing that must be guarded against is not so much the initial clearance as the differential clearance; that is, the increase in the clearance that is due to the different ratios of expansion, the coefficients of expansion of the groove and the ring. The narrower the ring is, the less the actual change in the clearance will be. Consequently, very much better results are obtained with the narrower rings.

In regard to the location of the scraper ring at the bottom of the skirt, it has always seemed to me that this would be very analogous, in the case of steam-engine practice, to installing some device to prevent lubrication of the crosshead. The piston, or the skirt portion of the piston, is the crosshead in the gas engine and it should be lubricated. The same thing applies to the use of baffles to prevent the throwing of oil up from the crankshaft. The real solution is to prevent the oil from working past the piston-rings, and to provide ample lubrication for the crosshead surface itself.

MR. LITLE:—Placing the feather edge in the bottom of the ring requires 14 sec. for the actual operation. I would rather determine its value by actually knowing how long and how well it produces on the job than in a laboratory machine such as Mr. McDewell used for that purpose.

I agree that the piston and the edge of the crosshead should be oiled copiously. I think it is not advisable to place guards around the bearings to prevent the oil from splashing up into the cylinder bore. It should be just slathered with oil. There should be plenty of oil for the cylinder and plenty of oil to pass to the skirt of the piston. The rings should be used to control passage of the oil.

My experience with engines that use 1 gal. of oil every 300 to 400 miles is that they are using too much oil. It requires too frequent visits to the service-station to have the carbon cleaned out.

CHAIRMAN GODDARD:—We were ready to put in baffle-plates but found that in using the aluminum piston with the large slot we kept the oil down enough so that we did not need them. Also, we are driving away more cars than ever before due to the railroad congestion and it would not be well to keep cylinder lubrication down in these new cars that many times are abused by irresponsible drivers.

We have, however, provided a place in the cylinder where baffles can be added after 5000, 10,000 and 15,000 miles, to keep the oil consumption down if this becomes necessary. By using a volume of oil, the oil lasts longer, and it will stay in better condition. Some of those here have spoken about the rings with the scraper groove in them being in the bottom groove of the piston. I assume that they mean the lowest groove above the wrist-pin. Our experience has been that, if they are put there, not enough oil gets above them to lubricate the two upper rings satisfactorily. Our practice in using scraper rings has been to put them in the top groove, but we could not always get enough drain-holes in there.

A. A. BULL:—It is unfortunate that in considering this question of oil consumption we are usually inclined to pick out some particular feature, instead of trying to consider the matter as a whole. Consequently, as has been evident, in almost every instance, changes or modifications may, under some conditions, prove absolutely ineffective under others.

The thing that counts is what happens in service. The purpose in locating the oil-discharge hole as I recommend is to get a condition that will exist throughout the life of the engine regardless of the bearing clearance. In other words, there is a certain discharge from the crank when the engine is new, and it is desirable if possible to keep that quantity of discharge the same regardless of the bearing fit. It is inevitable that the bearing will get loose. If you have an excessive bearing-clearance there is absolutely no guarantee that the oil will reach the place where the bearing pressure exists for the greater part of the time.

So far as providing adequate lubrication is concerned, the time that you get the discharge is when you change the direction of the pressure on the crankpin due to the influence of the pressure in the cylinder; the greater the load is, the greater the interval will be; consequently, the greater is the supply of oil. Mr. Woolson said that it was shown that the bearings on some engines do not stand up as well. I maintain that was not because there was not sufficient lubrication or that the oil was not sufficiently distributed, but rather that the temperature reached under these particular operating conditions required a larger portion of oil to be circulated. That may be because the clearance was inadequate. We must recognize that high temperatures call for a different clearance.

We must have a definite amount of lubrication in the cylinders. I grant that absolutely. But again I say that if, when the engine is first built, the lubrication given the cylinder is a definite amount, and if we agree that is sufficient, all we want to do is to maintain it. In placing the oil-hole at the top of the pin there is no question that the discharge from the cylinder increases as the bearing wears. If, in order to control the oil going into the cylinder, baffle-plates are placed over the bearings, the effect of which is to do just what we are trying to do with the oil-hole location, I would like to know how we expect, when the engine is new, to get any oil in there at all. Under conditions where the job is new, we should have the most oil; and, after the engine is worn, we need the least. With the ordinary construction and oil-hole location, we take steps to give it more when it needs less.

On the question of how much oil we should use, I believe that 1 gal. per 1000 miles is good all of the time.

On the question of ring wear and piston hardness, I am a champion of the aluminum piston. Mr. Litle believes that we will eliminate the troubles with an aluminum piston in regard to the side clearance of the ring when we make the piston as hard as cast iron. My arguments on this are that, while we may do things to the piston and rings that will more or less limit the wear, sooner or later it will occur. What I would like to do, if possible, is to provide something that will take-up wear automatically and maintain the condition that we know should exist.

We argue that making the ring thinner will reduce the force of inertia that is responsible for the wearing down of the grooves. If that is the predominating cause, why is it that the top ring in the piston, and the second ring, will invariably wear at a considerably greater rate than the rings below it? I have seen instances where, after both top rings were worn as much as 1/32 in., the third ring was in fair condition and the bottom ring in the same piston and subject to the same inertia process practically in the same condition as when originally installed. I think that there are some other factors affecting this wear that we do not appreciate fully.

As to properly finished cylinder bores, I used to argue that it is useless to put on a very fine finish and make a nice round cylinder bore, because I did not know just what the shape of the cylinder would be under operating conditions. Subsequently, in the development of pistons, we found that the cylinder was a peculiar shaped one under operating conditions. Afterward, we made an engine with cylindrical sleeves the same thickness all of the way around and machined both inside and out. Then we found that whatever we put in the cylinder to start with was maintained pretty well under operating conditions. Rings and pistons that would not produce a compression in the ordinary cylinder, no matter how well it was finished, would work perfectly well in the inserted-sleeve type of cylinder bore because, under the conditions in which the engines were operating, we had a fairly constant relation between the ring and the cylinder.

In an experiment we made 4 or 5 years ago, we drilled a number of holes in a piston immediately below one ring-groove. We put in the ring with just an ordinary mechanical fit, cleaned the piston on the inside and painted it white, so that we could trace the flow of oil through the holes. We made a partition on the bottom to prevent any oil from being splashed inside. We ran this piston in the cylinder until there was evidence of oil passing up to the top surface of the piston. We expected, of course, to find that the oil had been scraped off the cylinder wall by the ring and pushed through these holes, but the oil was not there. It did not push through until we made the ring a real tight fit in the groove. When you make a large hole or slot such as you have with the slipper type of piston, it makes it that much easier for oil to pass through. Probably it will be more effective than the ordinary 1/8 or 1/32-in. holes would be under the same running conditions.

I made a fairly definite statement in my paper to the effect that I do not believe piston clearance of itself has anything to do with oil-pumping. I think that the fit of the piston in the cylinder does have an effect on the way the ring functions. In that respect I agree that the location of the piston-pin has considerable to do with that because, if you have 0.006-in. clearance and the piston-pin is located near the top of the piston, the angle of the piston in the cylinder will be much greater than if that same piston were provided with a pin in the middle of its bearing face. There is a too prevalent opinion that this question of clearance is the real cause for oil-pumping. Let us consider for a moment the sloppy type of piston. It is exposed completely on two sides. The oil-film would come up 1/2 in. thick if it could, but it cannot. If it be true that clearance in itself will permit a larger quantity of the oil to cling to the walls, then I would say that the slipper type of piston would be a very poor job from the standpoint of oil-pumping. Yet it has proved to be very good and for no other reason, in my opinion, than that it is much easier with a large clearance actually to displace the oil-film or roll it up off the cylinder bore and push it through the holes.

The character of the lower edge of the ring is important. I made a statement to that effect in my paper,

(*Concluded on p.* 519)

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The Modern Airplane and All-Metal Construction

By William B. Stout[1]

Metropolitan Section Paper | *Illustrated with* Photographs and Drawings

THE author emphasizes that the best type of airplane combines in its make-up a complete solution of structural problems with the best aerodynamic compromise, and that the eventual airplane will expose no parts to the air that do not give back a resultant lift for their resistance.

After outlining the structural problems, the progress of the development of thick-wing and all-metal airplanes with which the author has been identified is reviewed and illustrated. Thick-wing problems are discussed and the requirements of all-metal airplane-construction are stated. The author believes that future commercial airplanes will have all-metal construction.

A DISCUSSION of the airplane of to-day when the new industry is progressing so fast must, of necessity, include planes now being built and in the course of laboratory development as well as those actually in use. Just what the modern airplane really is, as to detailed description, depends largely upon which particular engineer describes it; each one will state what his research shows, in his judgment, to be that combination of structure and aerodynamics which constitutes the best airplane. Again, the best airplane for some specified usage may be the worst airplane for some other specific purpose.

In comparison with the marine field, the aerial speed scout of to-day is represented by the racing hydroplane or speed boat, which, with a tremendous amount of power per pound of vehicle, still carries but very little load and sacrifices everything to speed. This speed type of airplane is by no means a commercial type, and I believe it is the commercial type in which the Society is interested primarily at present.

Directly opposite in the comparison is the old type-B Wright biplane which, with a 30-hp. engine, flew at slow speed and carried a weight of about 50 lb. per hp. This, in aerial parlance, compares in a way with the rowboat that has a motor attachment which, with a very small amount of horsepower, carries a heavy load per horsepower but travels at very moderate speed.

The attempts of designers to-day are largely toward reducing the amount of horsepower required for flying by two methods; (*a*) by reducing the amount of dead weight carried for a given useful load and (*b*) by seeking a minimum of "parasite" resistance toward forward movement, as against useful resistance which gives back a resultant lift.

As I have pointed out to the Society before, there are two major branches of airplane design; the first is structural, and the second is aerodynamic. An airplane comprising poor aerodynamics and very light successful structure will fly, but one of perfect aerodynamics with faulty structure is of no value whatever. The structural division of research, therefore, absolutely must be solved, and *that development is the best airplane which combines in its make-up a complete solution of structural problems with the best aerodynamic compromise.*

In the past, too many designers have approached the airplane from the standpoint of wing curve, lift-drift ratio and the like; forgetting that, after all, he who can build the lightest areas for a given strength has produced at least the fundamentals of the best airplane. Airplane design to-day, therefore, is largely a structural problem.

Structural Problems

The structural problems that arise are subdivided into the necessary materials, and the arrangement of those materials. The original Wright efforts were biplanes made of spruce, cotton shirting, piano wire, stove bolts, tire tape and the like. The original Bleriot design, using almost the same materials, was of monoplane arrangement with a different type of landing gear. From that day, until recently, the airplane designer added but little either to the arrangement of the structure or the materials used. Our DeHaviland war airplanes were built of the same spruce and ash and cotton cloth, but saved a few pounds of weight on the airplane by the use of expensive nickel-steel bolts instead of stove bolts costing but a few cents each. The details of airplane construction have improved wonderfully, but the materials and the arrangement of the structures developed but little for a long time.

It must be admitted that the present-day airplane, as now in use, carrying about 28 to 30 per cent of useful load and a pay load of 3 to 5 lb. per hp., cannot hope to be the eventual commercial-passenger or freight-carrying machine. Ways must be found, if a real commercial airplane is to be available for air lines, for improving the structural part of the present type of airplane to make it lighter and stronger and cheaper. This can be done by using better materials and providing better structural arrangement, and then combining with them improvements in aerodynamics. Aerodynamic improvements include better wing curves and the elimination of parasite resistances by better arrangement of the design and better streamlining of those parts that must be exposed.

The eventual airplane will expose no parts to the air that do not give back lift in return for their resistance. This airplane will be practically nothing but wings. Such an airplane was outlined to the Aircraft Production Board with drawings and technical data in 1917, and marked the beginnings of our own experiments in actual construction of so-called thick-wing, or cantilever, monoplane-winged airplanes.

Progress of Development

Our own analysis of the best methods of solving the problems outlined above can best be visualized by reviewing first the history of our work to date in thick-wing and finally in all-metal airplanes. Our original

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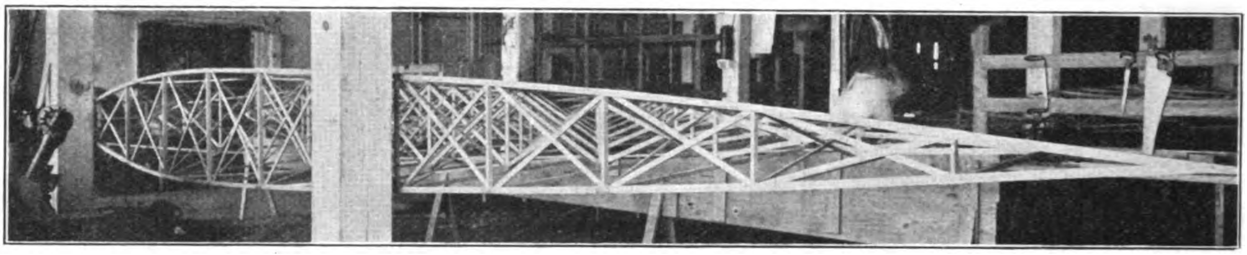

FIG. 1—FIRST MOCK-UP OF THE THICK-WING PLANE MADE IN JUNE, 1918

idea was that, if parasite resistances form two-thirds of the horsepower resistance of airplanes of that day, the first line of attack for the engineer to follow was to eliminate parasite resistance. The simplest and most obvious way was to make nothing but a wing and put everything inside the wing. Thus, for every pound of resistance to forward motion for which we must expend horsepower, we would get back 8 to 20 lb. of lift. To be able to do this would be to double or quadruple airplane performance at one jump.

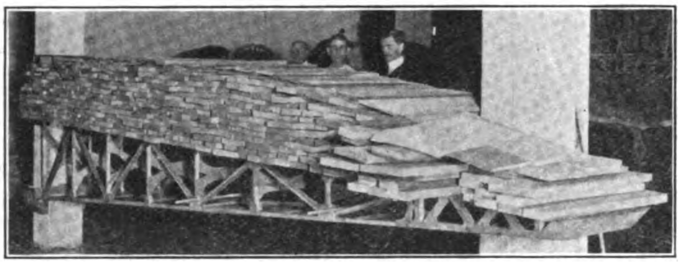

FIG. 2—FIRST CANTILEVER TEST OF A SPAR WEIGHING 7 LB. UNDER A LOAD OF 2200 LB.

Fig. 1 shows our first mock-up, built to show the idea we had in mind. This gave us a view of some of the engineering problems we were up against, and a peep into the new structural possibilities that had come with a search for mere aerodynamic advantage.

I will not follow our wind-tunnel research, or choice of a wing curve and the like. The type of spar was the first thing developed. Fig. 2 shows the wing spar as tested, supporting 2200 lb. of evenly distributed load from the root to the tip, the spar does not touch the outer post, and in a weight-per-spar of 7 lb.

Fig. 3 shows how the spars were fitted to the body and the general scheme of transverse structure. On proceeding with our patent work in connection with the structural progress we found that a similar fundamental had been in the mind of Hubert Latham of France, who had built a peculiar pouter-pigeon-like plane of thick-wing design which actually flew but was soon crashed. Latham, therefore, seems to have been the real inventor of the thick-wing plane later claimed and developed by Junker of Germany, but all from the same fundamental idea. Junker's patent views, uncovered after our first planes were in the air, showed an almost similar front view to that shown in Fig. 3, but it differed materially in other features. Latham's machine, however, antedates Junker's applications by several years.

When this work started, we had no thick-wing curves and we, with others, had the idea that a thick wing would have more resistance than a thin wing, forgetting that a wing's value is in the amount of air that it displaces, as well as its minimum of drag. Our first ship therefore was designed with a long thin curve with a fineness ratio of about 1 to 12. To get the wing thickness and still have depth enough for our spars, we lengthened the chord of the wing at the fuselage to extend the entire length of the fuselage, as shown by the photograph of the completed plane in Fig. 4. This gave us spar depth and structural advantage, but introduced problems of center of pressure movement that it took some time to solve even on paper. Fig. 5 is a front view of the same structure.

The airplane as shown had an area of 480 sq. ft., was made entirely of wood and three-ply veneer, the first veneer plane built so far as we know, and with a 150-hp. Hispano engine weighed complete but 1542 lb. It had retractable radiators, a full factor of safety of 6, and its lightness proved that at least we had hit upon a good structural fundamental even before we tried it in flight. This plane was hopped at Dayton at McCook Field in 1918 even though the engine we had been furnished had a broken pump shaft and hence could attempt no serious flights. The plane seemed normal as to lift and fore-and-aft balance. Its ailerons were ineffective and the vision abominable. The lessons learned from this plane, however, were well worthwhile and led to the next step under civilian auspices after the armistice.

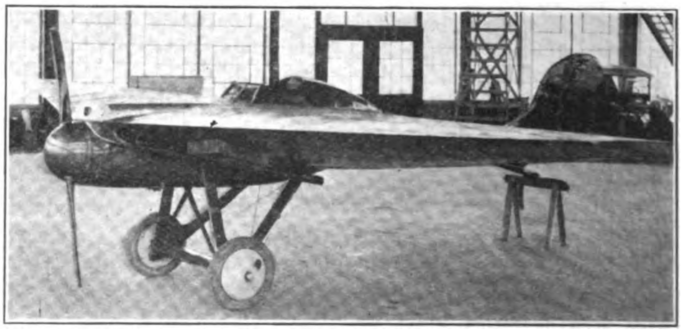

FIG. 4—SIDE VIEW OF THE FIRST VENEER BATWING AIRPLANE
The Area of This Plane Was 480 Sq. Ft., the Total Weight Was 1542 Lb. and the Engine Was a 150-Hp. Hispano

The war being over, we laid out and started a four-passenger commercial job along the same lines, but influenced by the lessons learned from previous work. A large number of changes were made for sales reasons, but it is my own opinion that the airplane of the future

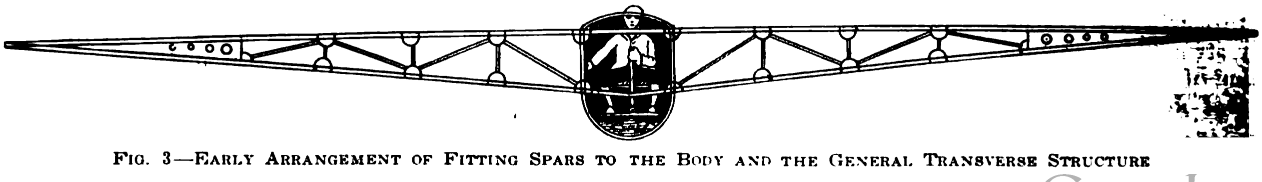

FIG. 3—EARLY ARRANGEMENT OF FITTING SPARS TO THE BODY AND THE GENERAL TRANSVERSE STRUCTURE

will look more and more like the original "Batwing" as it was dubbed at McCook Field. The new design, however, using a small-lift, high-speed curve, had a gap between the tail and the long-chord wing at the demand of opinions, and had added a cabin and features to fit it to commercial use.

Fig. 6 shows the commercial sedan, also of veneer throughout. This plane, piloted by Bert Acosta, flew at the first attempt. The wing curve was poor in lift and the climb was bad. The pilot was enclosed and this was undesirable. The vision was not ideal.

Our light-lift wing-curve had been chosen because of its small *CP* or center of pressure movement as we, and all of our advisers, were afraid of the action of the long chord with a really cambered curve. The poor lift of this airplane forced us to take it back to the shop, however, and change the wing nose to give us a really lifting-wing profile. Trials with this airplane, as changed, surprised us. We had more lift and climb than we expected. With a 200-hp. Packard eight-cylinder engine, the machine with 750 lb. of load made an official speed of 112 m.p.h. and climbed 4800 ft. in 10 min. It was a tricky flier, however, and in the glide "hunted" in a way to alarm any but a pilot of strong nerves and wide experience.

Fig. 6—The Four-Passenger Enclosed Sedan Airplane Which Is Driven by a 200-Hp. Engine

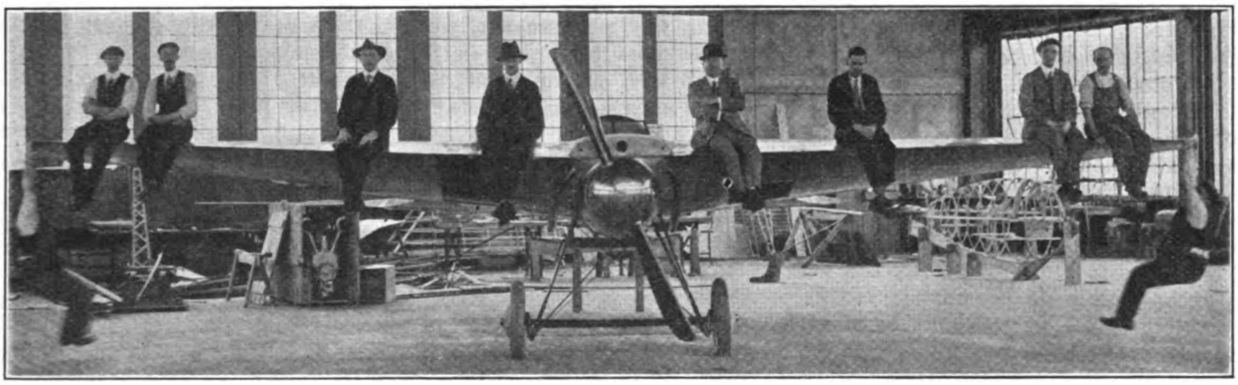

Fig. 5—Front View of the Airplane Shown in Fig. 4

This airplane was flown from 30 to 40 times with a load and without, and with new tails, new flippers, larger and smaller rudders and what not, but still the hunting due to the *CP* travel continued. At last the idea came, and by a change not occupying more than 10 min. a new airplane was born. The hunting was cured, the controls all became more effective and the flippers too much so, and our "batwing" problems were solved. Later this 36-ft.-spread airplane flew with a 1070-lb. useful load, and got off with real snap and ginger. Our future large type commercial airplanes will be of this long-chord type, but of all-metal rather than veneer construction.

Following our experiments this far one can see that an original search for a new aerodynamic advance brought with it as more or less of a surprise a considerable advance in structural possibilities, so that it was possible to build lighter areas of the same factor of safety.

Thick-Wing Problems

The entire problem in a thick-wing job seems to be not so much the strength of the spar members involved, as this is comparatively easy to obtain, but the *wing rigidity* so that there is no distortion or warping of the wing against aileron action, nor change of angle of incidence at the tips at different angles of attack of the plane. Foreign designers, particularly of the German school, have gone either to the thick-wing curves, such as the Junker, or to outside trussing, such as the Dornier. In the thick-wing Junker type, with an almost rectangular span of wing, the airfoil becomes so very deep at the fuselage in proportion to the chord of the wing that considerable speed is sacrificed to structural depth. To get the twist out of the wing, it is necessary to use very long interlatticed spars of more or less tetrahedral connection. This binds the structure into a solid unit but adds considerable weight, although, in the metal Junker, the wing is about the same weight per square foot as wood-and-cloth wings of the same general factor of safety and maximum spread.

In our type of wing we used a much more tapered plan view, so that for equal spar depth at the center section, as shown in Fig. 7, we had a wing of much better fineness ratio for high-speed work, at the same time the amount of structural space in the thick-wing, or German type of wing, extended only as far as the shaded portion *A* in Fig. 7; whereas, in our long-chord type, our structural space was almost three times the volume, as at *B*, allowing considerable advantage particularly for wing-tip rigidity. It was possible to build larger areas of equal weight by this method. The advantage of getting the greater amount of area toward the fuselage is obvious, from both structural and aerodynamic standpoints.

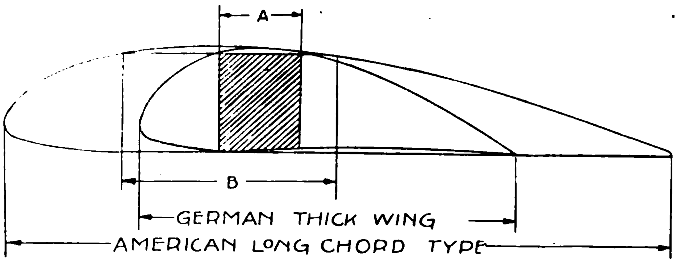

Fig. 7—Comparison of the German Thick-Wing and the American Long-Chord Types

Advancement along this line began, as stated, back in the early days of flying when Hubert Latham flew his thick-wing design with a fuselage on top and an enclosed landing gear, shortly after Bleriot made his famous Channel-crossing flight. The plane was crashed, however, after about its third flight and abandoned. Our experiments in this Country on the thick-wing type date from 1917 when we started, through the Aircraft Production Board, experimental activities on an airplane which, on account of its appearance, was quickly dubbed the "Batwing." So far as we now know, only one firm at that time, in any way paralleled the work we were doing. This was the Junker firm in Germany, which started its experimental work some time before our activities, but without our knowledge. Photographs show that the Germans started with the same type of wing-truss experiments as we later tried out in this Country, with the same method of building spars, mounting them and loading them, that is shown in Fig. 7. The only difference is that Junker worked in metal, originally in steel, instead of wood.

Starting from the same fundamental of enclosing as much as possible within the wing surface, Junker nevertheless adopted the old conventional arrangement of wings, fuselage, tail and rudder, and with the usual aspect ratio of wing, moment arm on the tail surfaces, and the like, making no attempt to produce a new aerodynamic plan other than that the wings should be thick and have the structure inside. The real original part of his development work was metal construction and all credit should be given to Dr. Junker for the very remarkable structures he finally developed, structures that are admirably suited to the particular type of airplane that he built, airplanes that have been flying in this Country under the name JL-6. Dr. Junker's work in metal indirectly grew from that of the Zeppelin Co. in duralumin in connection with dirigible construction, and it is only natural that it should be paralleled by the work of the Zeppelin Co. represented by its engineer, Herr Dornier.

Figuring also on a rectangular wing and noting the disadvantages Junker had in maintaining wing rigidity by the thick wing at the root to the limitation of speed, Dornier adopted a monoplane type using the same wing-spar thickness from tip to tip approximately, and a rectangular plan view of wing but fitted with outside brace struts, as shown in Fig. 8. This view shows the typical Dornier construction. The Junker, or JL-6 construction, with deeply corrugated metal surfacing, is well known in this Country, and it is doubtful if the Junker structure has been equaled in ships of its size.

Dornier obtained, by the use of a smooth section with inset ribs sticking above the surface every 8 to 10 in., a slightly better speed advantage but with no advantage of lift so far as the wings were concerned. In the end, through using external trussing, he obtained not quite so good a performance per horsepower due to the fact that his wings were heavier per square foot and that the airplane had more parasite resistance.

Another metal airplane, built by the Germans, was known as the Staaken Giant, a very remarkable, four-engine airplane, built originally to fly between Berlin and Friedrichshaven. The airplane was not a complete success due to overweight, but was a remarkable development. With a wing loading of 16 lb. per sq. ft., this machine flew successfully, but had a landing speed of 83 m.p.h. The Germans are not dropping their experimental work on these airplanes, however, because the first ones were overweight and faulty in detail, but are only awaiting the permission of the Allies to go ahead on further development structures. The Staaken Giant is the most pretentious example of all-metal structure ever undertaken. The Junker, however, as a commercial venture is the most successful.

All-Metal-Construction Requirements

I have used considerable space in explaining thick-wing design and the various schools involved, since it is this type of design that seems to fit all-metal construction best. To get the utmost out of the strength of the wing section, it is an advantage, with most of the designs to date, to use tapered spars so that the section itself can form part of the structure. In analyzing for all-metal construction in airplanes, there comes up at once a question of steel versus duralumin.

When we started our all-metal construction work, 2 years ago, little or nothing was known of duralumin, and designers were fearful as to its stability under conditions of weather, corrosion and vibration. Enough is known of it to-day so that we can speak openly and with knowledge of the real properties of the metal. As to the greater merits of metal versus wood-and-cloth, it is understood that in the statements I make I am voicing only our opinions as the result of our own experiments and accumulation of data during intensive work in this metal during the past 2 years, and the personal investment on the part of our company and associates of over $100,000. The properties of duralumin are stated in Table 1.

TABLE 1—PROPERTIES OF DURALUMIN

Specific Gravity	280
Weight per Cubic Inch, lb.	0.102
Melting Range	
Deg. Cent.	540 to 650
Deg. Fahr.	1,004 to 1,202
Modulus of Elasticity	10,600,000
Coefficient of Expansion	
Per Deg. Cent.	0.000002260
Per Deg. Fahr.	0.000001255
Yield-Point, lb. per sq. in.	30,000
Strength when Tempered, lb. per sq. in.	
Compressive	44,000
Shearing	30,000
Tensile	50,000 to 60,000
Elongation when Tempered, per cent	16 to 20

It is true that steels can be had of much higher tensile-strength per pound than dural, as duralumin has come to be called colloquially. It is also true, however, that these steels, heat-treated in very thin sections, are even more of an unknown quantity due to inaccuracies in the heat-treating, particularly in experimental structures, and therefore have variable physical characteristics. Dural, treated in a bath of nitrates at temperatures well under control and fabricated quickly before the tempering has begun to take effect, is thoroughly reliable and can be depended upon for a 55,000-lb. per sq. in. tensile-strength, with an 18-per cent elongation.

If steel spars are used in connection with dural structure, provision must be made for differences in the coefficient of expansion and for the production difficulties of riveting metals of different hardness. There is an advantage in making a complete dural structure in that no expansion difficulties are encountered, and production problems are much simplified.

Our first problem in building an all-metal airplane was the same as in building our first wooden one, the development of a tapered spar of sufficient strength for wing requirement. The contract we were working under required an airplane of 60-ft. span, maximum, and 600

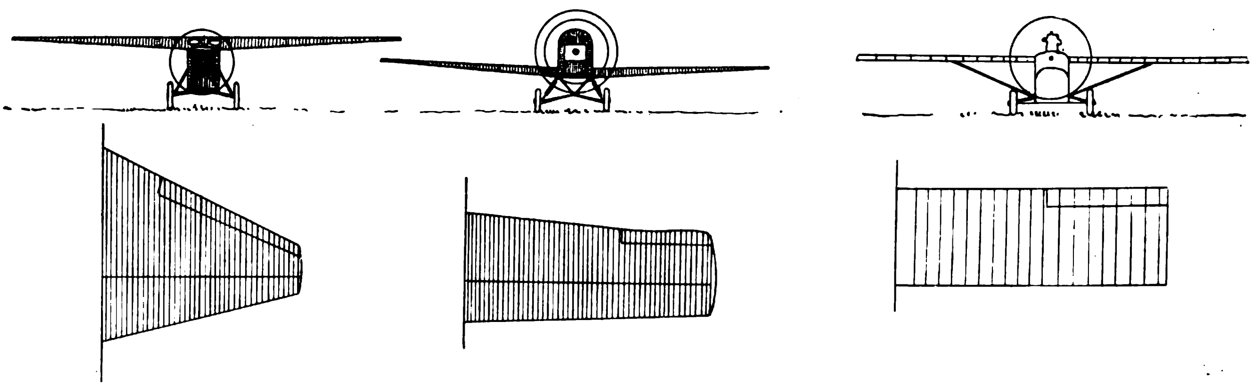

Fig. 8—Front Views, from Left to Right, of the Stout Commercial, Junker and Dornier Airplanes and Their Wing Plans Underneath

hp., to be arranged either for wheels or floats, and to carry approximately a 2-ton load at 105 m.p.h. In the trials the airplane, with its full load, officially, made 113½ m.p.h. and was the first American-built, all-metal machine to fly in this Country. Our "Batwing" airplane at McCook Field was the first thick-wing machine to fly in America, and possibly the first veneer airplane built on this side of the Atlantic.

The first requirement for the wing spar was the development of what we originally called the "spar longerons" but which later were termed "chord" sections. These are the top and bottom members of the latticed girder, as shown in Fig. 3. Development work was done on 19-in. hand-made sections, starting with the conventional U-shaped members and developing through the various convolutions, and changes found necessary in each until the master section developed, as shown in Fig. 9. The upper left view is the spar chord-section. In the 19-in. column this weighs 7½ oz. and will support 8000 lb., or 4 tons. The longeron section, weighing about 4 oz. in the test column, and supporting in column load 4400 lb., is illustrated in the upper right corner. The drawing in the lower left corner is that of an ordinary U-section used for ribs, and that in the lower right is a special diagonal section for the lattice of the spar girders which, in the 19-in. column, supported 2100 lb.

In these pieces, the advantage of dural over steel was shown. The tension member is not the problem in building a spar; otherwise we could make our spars of piano-wire cable. The real problem is the compression member. Dural, being so light and with a much thicker section in proportion to its strength than steel, has a considerable advantage in rigidity, hence these light weights in proportion to the strength. Our entire airplane substantially was made up from these four sections in various arrangements, and with different fittings and connections. Aside from the fact that the parts were all metal, the general structural arrangement and spar

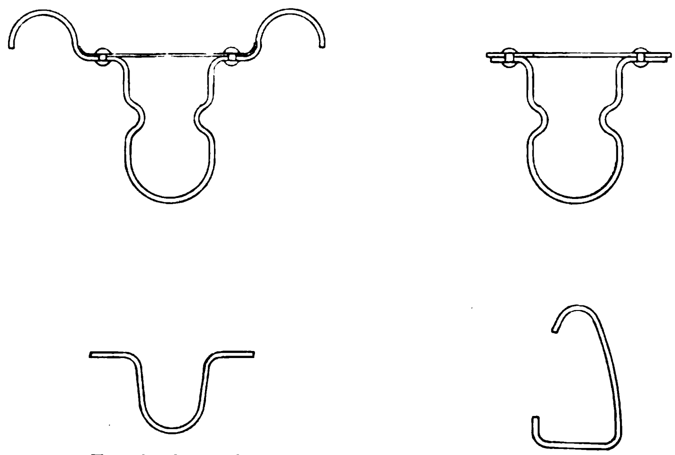

Fig. 9—Some Structural Shapes of Duralumin

layout followed very closely the airplanes we had previously built in veneer.

The Junker airplane uses dural tubing for spars and places these, with diagonal latticing between and stamped from dural sheet and riveted in place. The Dornier uses the U-shaped spar-members with side plates and with plate ribs running fore-and-aft between each, the flat surface pieces with their upturned edges being left to form external ribs. The Staaken Giant used metal of a heavier gage, flat sheet for surfacing with plate ribs inside. Both the Dornier and Zeppelin or Staaken designs were heavy compared to the Junker.

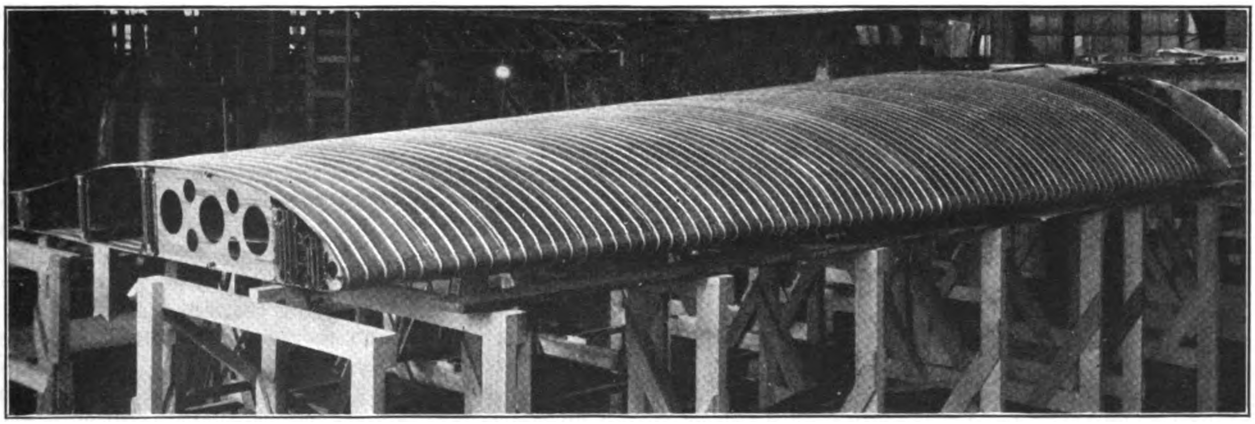

Fig. 10—An Airplane Wing That Was Constructed Entirely of Duralumin

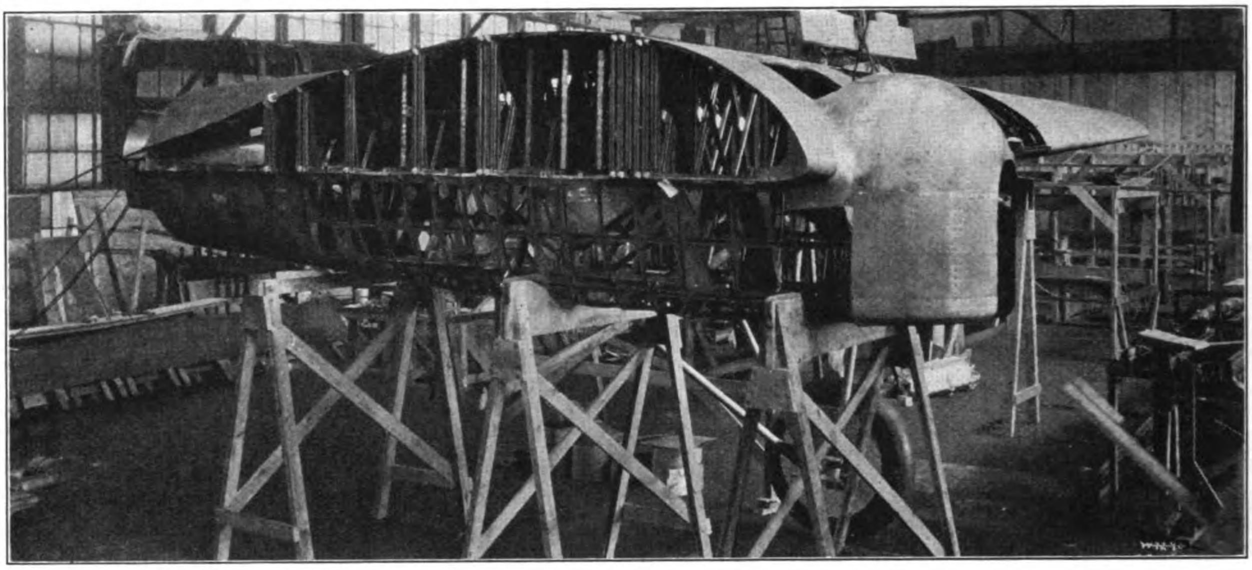

FIG. 11—A SIDE VIEW OF THE FUSELAGE IN COURSE OF CONSTRUCTION IN THE SHOP

In our plane we allowed 10 spars by their depth to form the contour of the wing fore-and-aft, and by merely attaching our wing skin of 0.020-in. dural, ribbed every 2 in., to these spars in a fore-and-aft direction, we obtained wings of great rigidity and strength, and in a weight equivalent to the best Dornier and considerably ahead of the Staaken weights, although heavier than the Junker wings, on account of our greater area and loading per square foot made necessary by our limitation to a 60-ft. span for the load to be carried.

Fig. 10 shows one of the wings complete, giving an idea of the smooth contour obtained and the size of the structure. Fig. 11 is a view of the side of the fuselage in the shop. This shows the arrangement of the spars, and gives an idea of the wing curve, the pilot's position and how the fuselage is divided just back of the wing for storage purposes. The weight of the airplane complete and fully loaded approximates 5 tons. Table 2 gives the comparative areas, weights and the like, of other metal airplanes previously built.

TABLE 2—COMPARATIVE AIRPLANE DATA

	Dornier Comet	Dornier Dragon-fly	Junker	Stout Ordi-nary	Stout Com-mercial
Span, ft.	58	28⅛	49	60	35
Length, ft.	30	23½	29	..	31
Height, ft.	8	7⅝	10	..	10
Area, sq. ft.	459	150	417	790	280
Chord, ft.	9	5¼	8½	13½	8
Weight, Full, lb.	3,960	1,652	4,101	9,817	2,025
Weight, Light, lb.	2,500	992	2,341	6,557	1,250
Useful Load, lb.	1,460	660	1,760	3,260	775
Useful Load, per cent	36.5	0.4	43.0	33.0	34.0
Engine Type or Size	BMW-233	80 hp.	BMW	600	90-OX
Wing Load, lb. per sq. ft.	8.65	9.70	9.60	12.40	7.20
Weight per Horsepower, lb.	17.0	20.7	17.4	16.0	22.5
Useful Load per Horsepower, lb.	6.2	8.2	7.2	6.1	8.6
Ton-Miles per Gallon	3.10	..	4.25	3.61	4.40

SUMMARY

All metal planes, to date, can be called experimental. The future commercial airplane, however, will undoubtedly be an all-metal construction. Metal planes mean greater safety to pilot and cargo; a possibility of considerably lighter weight; less production cost, particularly as quantities go up and the demand increases; and easier servicing and simple repairs provided the airplanes are designed with this idea in view.

It cannot be expected that the first metal-construction attempts of any manufacturer will be a success in every particular but, if America is to lead in aircraft, it must lead in metal aircraft, as it is my opinion that, in a comparatively few years from now, wooden airplanes in the air will be scarcer than wooden ships on the sea, and that all airplanes flying under insurance rulings will be of all-metal construction.

Thick-wing airplanes are developing fast, both in monoplane and biplane types. Retractable chassis, wing-type radiators and all those items that the recent Pulitzer events have shown to be practicable, will appear shortly in commercial airplanes and increase their profit-paying possibilities. But if safety and low cost are to come with these items of greater performance, then must metal construction and production methods be applied to the producing of an airplane for American air-services that shall be safe, cheap, economical and long-lived. Only by the production of a real commercial airplane can commercial aviation come in America.

THE DISCUSSION

COM. H. C. RICHARDSON:—I am unable to agree with Mr. Stout in his conclusions that "the eventual airplane will expose no parts to the air that do not give back lift in return for their resistance." The statement is not incorrect, but the inference to be derived from it is incorrect. At least I infer that Mr. Stout intends that no elements are to be exposed which are not designed to give lift in a normal attitude. Practically anything but a sphere will give a lift if properly set to the wind, but very few shapes give lift efficiently except wing-shaped sections. Struts, floats and fuselage sections can approximate wing-sections in form, but in most cases the aspect ratios are bad and the lift-drift ratios are not high. Any attempt to give them lifting sections will result usually in an expansive life on this account; whereas, to give them minimum resistance, but little lift is sacrificed and, in a normal attitude of flight, their resistance is lower than would be the case if made a lifting section. That the aerodynamic qualities must be as clean and efficient as

possible and that the structural design that offends against this must be modified is, I believe, recognized by anyone familiar with aircraft design. The parasite resistance must be reduced to the lowest factor compatible with the purpose of the design, and this implies structural efficiency. But questions of stability, maneuverability and arrangement require that an aircraft shall be more than wings alone, and the additional members, I contend, will be more efficient if carefully streamlined and carefully disposed to meet the conditions of maximum performance, than would be the case if the attempt were made to get a lift as the prime feature in the design of these elements. I believe it better to gain a lift from the most efficient lifting member and to give the parasite members the best streamlining possible.

Present practice indicates a minimum wing-drag at from 2 to 4 deg., but when this is associated with parasite resistance the minimum drag for a complete airplane is usually found to be at from 8 to 10 deg. With thicker wings, these data require some modification, as some of these sections are most efficient at angles in the neighborhood of 0 deg.; but the effect of additional parasite resistance will be of the same nature as that indicated. In still air the maximum efficiency for cruising is at an angle of attack of about 7 deg., and this angle changes very little with the load carried. From this it appears that it is of paramount importance to commercial craft to dispose the fuselage for minimum resistance at such an angle of the wings. Depending on the purpose for which an airplane is designed, there will be an optimum angle for the wing setting to give the best result whether it is climb or speed, or some other quality that is considered most important; and this should be given consideration in locating the wings on the fuselage so that the resistance of the fuselage will be a minimum for the chosen condition. Relative to wing efficiency, I am not unaware that a head or a tail-wind will change the optimum angle for cruising, but the slight variations involved will not affect the fuselage resistance seriously.

For the time being it appears that the wing loading which can be used is controlled by getaway and landing speeds; but, for commercial aircraft, I believe the greatest efficiency of transportation requires high power-loadings, although here again is a limit, as the design must not be sluggish. To combat head winds or bad air requires a reserve of power, and that means a reserve of speed and of fuel and oil. There is, therefore, a limit to power loading, and I place the desirable maximum speed at least 50 per cent above the landing speed and preferably 100 per cent above it.

We still have something to gain in propeller and engine efficiency, and much to gain in structural efficiency. Regarding structural efficiency, I believe we may well haul in our horns on the factors of safety and gain a worthwhile improvement in useful load without danger. A factor of safety of 4 appears ample, and even 3 should be sufficient for a carefully designed airplane if the pilots will confine their maneuvers to those required for commercial purposes and avoid stunting. Bad air and forced landings require consideration, however, in determining this reduced factor.

I believe in metal construction and that ultimately it may replace wood and wire, but it requires much research. I believe it will be warranted on a production basis only. Both research and production to be warranted require established service and a demand incident thereto, and I believe that both will develop more rapidly with the demonstration of efficient wood-and-wire construction than can be hoped for in the immediate future from metal construction. Also, it appears that, for the present, the light alloys have an advantage over steel, because the thicker sections used for equal strength have greater resistance to secondary failures.

Maintenance is important and it is here that metal offers important advantages from a deterioration standpoint but, from the viewpoint of field repairs and minor crashes, I anticipate much anxiety in regard to a metal venture until a production basis is reached and readily available spare units permit damaged parts to be replaced easily so that they can be returned to central shops for repairs where heat-treatment and proper facilities are available.

I congratulate Mr. Stout on his paper, but more on the valuable experience he has had in this pioneer work. I congratulate him on his vision and faith and persistence, and truly hope he will see and participate in the development that is bound to follow the efforts which have required no little financial and personal courage. He has important assets in his experience which I hope will materialize to his advantage and to the advantage of aeronautics.

HARLAN D. FOWLER:—Refinement in the detail design of structural parts sufficient to develop the required strength demands experience and good engineering principles. It was necessary in the past to construct portions of an airplane of more than the necessary weight to insure proper strength. This was due to the uncertainty of the conditions to be met. With sand-load and flying tests as they have been developed up to the present, it is possible to design parts that have just sufficient strength and so obtain a very light member. An example of this is the hull of a flying-boat. The conditions met with on landing, porpoising, taxiing or getting off are uncertain quantities. The British made some very difficult tests[2] in 1920 by subjecting the hulls of the F-3 and the H-16, weighing respectively 10,600 lb. and 11,600 lb., to all of these conditions; and, by ingenious measuring devices, they were able to determine local and distributed stresses along the entire bottom of the hulls ahead of the step. These measurements revealed the localities where much excess weight could be eliminated, and this is one reason so many of the seaplanes have poor useful-load capacity. Within narrow limits, a seaplane should be as light as a land machine.

We are greatly concerned with the reliability and durability of our engines. The attempt to develop large power and light weight is justifiable for military purposes. But are they practicable for commercial usage? At present it is necessary to have a large reserve of airplanes on hand to maintain regular service such as existed in the Forest-Fire Patrol on the Pacific Coast. It was necessary there to have at least three machines tied up to keep one in the air, and the greater number required engine attention. The vertical type of engine represented in the German machines certainly gave more uniform service.

In the last few years a very important phase of development has been recognized as being essential to increase the usefulness of airplanes. The rigid or inflexible type of wing construction must be relegated as belonging only to the past progress of the art. Efficient as thick wings may be, the advantage of the variable area, variable camber or auxiliary wing-surfaces presents too great a possibility to be allowed to lie dormant much longer. To obtain a safe landing-speed or to get off with a heavy load, we are obliged to carry a certain amount of excess wing-

[2] See British Advisory Committee for Aeronautics Report and Memorandum No. 683.

surface and an inflexible wing-section. Having once cleared the earth, why do we pack with us on a journey occupying hours all of this parasite or airfoil resistance that demands such unnecessary power as the cube of the speed requires? It is apparently something we must rid ourselves of very quickly. It is also apparent that the variable area is one of the most important features offering the most substantial gain. There may be some mechanical difficulties at first. So also there was difficulty with the first automobile gearshifting device, which is closely analogous to our problem. To be able to increase the passenger load 100 per cent, or to reduce the power required 25 per cent, or to increase the high speed 20 per cent, or to reduce the landing speed 15 per cent, certainly offers an interesting goal.

With few exceptions for steel, there is no doubt of the advent of the duralumin airplane. The lightness, strength and ease of handling of this metal eventually will simplify our constructional problems. With experienced workmen and the gradual increase in demand for duralumin tending to cheapen its manufacturing cost, it will be a serious competitor with wood. As Mr. Stout points out, duralumin is no longer a metal of mysterious properties. Aside from exercising care in the annealing and tempering of this metal, it is nothing more than a high-class tinner's-job. To see the use of this metal in the framework of the ZR-1 airship under construction at the Naval Aircraft Factory at Philadelphia is very conclusive evidence of its possibilities. If a suitable brazing compound could be found, the use of duralumin for fittings would be more extensive. However, duralumin in the forged state has been used, to a lesser degree, with some success.

H. M. CRANE: — I know from past experience with light-boat construction and similar work that it is essential that every bit of material in a light structure bear its just proportion of the load. If you intend to have a factor of safety of 6 and there is a large part of the structure that has that factor and another large part that really has a factor of 2, the structure is inefficient.

The most difficult feature of any problem of structure is the design of the compression members. The Quebec bridge disaster was due solely to failure of the compression members. The failure of the dirigible R-38 was caused unquestionably by the failure of a compression member. The joint between the different members is another important detail. When dealing with thin sections of metal, it is very hard to make sure that the load is distributed over the whole structure. It is very apt to be very much localized, with a certain attending failure. You can make tests on 19-in. pieces in a testing machine and get an excellent line on what you must do; but when you design a 70 or 100-ft. span on an airplane, it is very difficult to translate the results of the 19-in. tests into terms that apply to the full length of the machine. It is equally difficult in construction to see that each of the hundreds and hundreds of pieces that go into that wing is really bearing the load it is designed to carry.

The fact that the winning machines in Detroit were all American machines from the engine, the radiator and the powerplant installation to the whole airplane construction, is a thing that we all can be justly proud of. I felt during the war, and it has been proved since, that the sooner we in this Country stopped looking abroad for inspiration and began to look at the airplane and the airplane engine from our own point of view, the sooner we would progress. I believe that, with the opportunities this Country has had, the progress in aviation has been astounding. That does not apply to commercial aviation, but I think we ought not to feel too much discouraged about commercial aviation. Commerce is a traffic, it is a barter and trade that is healthy and of real value only when something of value is sold to somebody who requires that thing.

The Aeromarine Airways, Inc., has made a strong start in that direction, taking the most favorable opportunities where the competition with other methods of transportation has been reasonable. The Air-Mail Service is making a record from day to day that is being surpassed nowhere. This is being done with a very moderate amount of support from the Government and by using salvaged machines.

The Liberty engine has been one of the most adversely criticized powerplants that was ever produced, but its history proves that it is probably the most remarkable feat of design and construction done by any country during the war. The first race at Detroit was between five Martin bombers, each powered with two Liberty engines. The winning machine averaged 105 m.p.h., the difference between its fastest and slowest lap being less than 0.5 per cent. The other four machines, while not so fast, ran with almost the same degree of uniformity. A Liberty engine recently drove an airplane for 35 or 36 hr. consecutively, raising the duration record in the air something like 12 hr. over the previous record made with a German engine.

HENRY E. BRUNO:—During the 3 years in which we have flown more than 1,000,000 passenger miles and carried over 20,000 people without a single serious accident, we have demonstrated that commercial aviation in this Country, so far as over-water flying goes, is not behind that of Europe. We inaugurated the first double daily service between two large cities in the United States in the summer of 1922. I refer to Cleveland and Detroit. We used three 11-passenger enclosed-cabin ships; two were in active service and one was in reserve. We started our double daily service July 17 and finished Sept. 17. We had no forced landings, no interruptions of schedule, and the boats left on time twice daily from each end of the route. The first 2 weeks these big 11-passenger boats carried about four passengers. It was heartbreaking to send those ships across the Lake that way, but we realized that we had to prove something to the people of Cleveland and Detroit. Toward the end of the service, we could not accommodate the people who wanted to fly.

CHARLES M. MANLY:—Mr. Stout has hammered steadfastly at the very important subject of parasite resistance. He awakened a greater realization of the enormous importance of parasite resistance when he indicated what a large portion of the total resistance of the DH4 machine is parasite resistance. Mr. Stout has wanted to put everything inside the wing. Commander Richardson questioned whether that may not be carrying it a little too far and pointed out the importance of streamlining certain portions and possibly leaving them exposed in their more efficient form. The importance of doing one or the other is very firmly established by the results of this speed test at Detroit. At an air-speed of 248 m.p.h., every square inch of normal surface has a pressure of 1 lb. on it. The winning machine, with the horsepower that was required to propel it at 248.5 m.p.h., was really experiencing a total head-resistance of very little more than 3 sq. ft. of normal exposed surface. That brings home to us very forcefully the advantage of minimum parasite-resistance and the importance of applying accurate scientific data and thoroughgoing engineering training and experience to the design and construction of a machine.

HERBERT CHASE: — Why did Mr. Stout give up the veneer type of construction in favor of the all-metal.

WILLIAM B. STOUT:—In figuring on commercial aviation and on safety and cost as the fundamentals of commercial aviation in comparison with maneuverability and some other things for military ships, our analysis was that metal would of necessity be the eventual material, if only to secure safety in crashes. Airplanes must operate in all climates. For Europe or for our Northern climates I think a veneer ship would be fine, but in a hot climate such as that near Mexico, where the casein glue would harden and the veneer dry-out and split, veneer is out of the question. Also, I would rather trust metal and rivets because it has been demonstrated that there is less possibility of serious personal injury in a crash.

I disagree with some of those who have spoken on the cost of metal construction. I admit that the experimental work we have done to date is expensive. But, having been through that, we know now how to design more economical structures. I believe that we can repair our newer-type structures in an emergency practically as quickly as we can repair wood.

Another advantage in all-metal construction is that the entire airplane can be made of comparatively small units. You save cost in the shop and, if you damage a unit, it is generally cheaper to replace it than it is to fix it. I believe that to-day we can build airplanes of certain passenger or load capacity more cheaply than we are now constructing passenger cars and motor trucks of the same capacity, up to 1 ton, if we can market them in equally large quantities.

We are flying now with one Liberty engine and carrying loads that required two Liberty engines during the war and at higher speeds. The airplane that made the world's endurance record recently left the ground with some 4200 lb. of gasoline aboard. Its engine had the same horsepower, practically, as that of the airplane that made the 248-m.p.h. speed; so, we are translating horsepower now not only into tremendous speeds, but also are able to translate the same horsepower, if we wish, into load carrying at slower speeds. I think we will see within very few years a nice four or five-passenger commercial airplane selling for less than $3,000. When we do see that, I think commercial aviation will have begun.

LEON OTTINGER:—It is only recently that really scientific work has been done in the manufacture of plywood in our different lines. For instance, the veneers are being chemically tested after being cut. It has been found that they can be treated chemically so as to wash out the cells fairly well. Then they are treated with chemicals that lower their coefficients of expansion and contraction to a very marked degree, when under the influence of either water or heat, without injuring the strength of the individual veneers. That is very important because the ability of plywood to withstand weather, due to the waterproof glue, means that the glue, in its soaked condition for instance, should be stronger than the expansive power or stress produced between the various parts of the veneer. If the veneer is reduced, the plywood will stay together with a weaker glue. Furthermore, the stresses on the various structural parts of the airplane must be definitely effected by a plywood that will not expand or contract to the extent that the old types of product did.

The matter of what woods to use for making the veneers and the manner in which they are cut provides a large field for research in plywood. I have seen a number of braces made of chemically treated plywood; the process is of German origin. The veneers were used before being fabricated, but I have seen pieces of that kind that had an angle of 90 deg. thrown into the water, and that remained at an angle of 90 deg. when taken out and dried. I think that the lack of development of plywood has been due partly to taking emergency Government specifications too much for granted; the plywood factories have not developed anything new in the way of airplane plywood, because it is very poor judgment to alter material from specifications on Government orders.

S. WARD SEELEY:—I believe the time is coming when radio telegraphic equipment will be as essential on an airplane as the landing-gear, for instance, and in view of the fact that all-metal equipment might militate against the successful employment of radio equipment, I am wondering whether Mr. Stout has made any tests on the effect of the all-metal construction on radio signals?

MR. STOUT:—No, we have not. However, I think that would have about the same relation to the development of metal airplanes as the question that came up when iron ships were first made, that the compass would not operate properly, and that therefore it would be useless. I think the radio will be the compass of the airplane and that every airplane will of necessity be equipped with it to travel through fogs and any kind of weather and for night flying. The radio engineers undoubtedly will solve any interference problem.

I believe that a factor of safety of 4 in a metal airplane is safer than a factor of safety of 6 in a wooden airplane. But if you intend to build in wood, I think plywood is the thing for the cantilever type of airplane. Since the wooden members that are used are of such small section in multiple units, if they are made of plywood, you know you have a pretty good average in the piece anyway, although the piece of plywood in itself will not pull as well as the best piece of spruce you can find; but it can be depended on to have uniform strength.

H. E. DAVIES:—Does Mr. Stout consider the riveted seams a source of weakness?

MR. STOUT:—No, provided the riveting is watched to be sure that it is a thorough job, and that the duralumin is watched when the holes are drilled so that no cracks are started. If heat-treated duralumin starts a crack, the crack is liable to run, just as with any tempered steel. We ream all of the holes after drilling and before we rivet, and are very careful to start no cracks and get as perfect riveting as we can.

HENRY R. SUTPHEN:—What effect has salt water on duralumin?

MR. STOUT:—In the natural state, duralumin is affected by salt water; in the heat-treated state, normally, it is not. I should say it is affected about as much as iron is with water. Some rust or surface corrosion comes but, when properly heat-treated, we have had absolutely no trouble with it. It is perfectly possible to weld it, but the welded joint seems to go back almost to aluminum and is not rust-proof. Salt water will affect it. We have put pieces of duralumin into a saturated solution of salt water for 3 months without having the metal destroyed; in fact, it showed practically no signs of corrosion. We hear from other sources that salt water does have effect, but we have had no trouble ourselves with corrosion of any kind.

MR. SUTPHEN:—Is that the reason you rivet the sections?

MR. STOUT: — Yes. We rivet the sections and heat-treat the rivets. One thing about the metal is sometimes confused with corrosion. In some of the very thin sheets from the German airplanes, what appears to be corrosion in spots where vibration has occurred can be noticed.

That is where parts are over-stressed through poor design. It is due to the fact that the metal in these airplanes was obtained during the war, when Germany could not get pure aluminum to make the duralumin, and there was too high a content of zinc in both the copper and the aluminum. When zinc is present in any aluminum alloy and there is vibration, the metal will tend to granulate. Under that condition, there is trouble. It looks like corrosion, but it is due to the presence of zinc in the aluminum. That is why our American duralumin seems to be superior to the foreign duralumin; we have a pure aluminum to start with, and at least as good copper, if not better.

MR. CHASE:—Does duralumin suffer from what is commonly called crystallization or fatigue to any greater extent than other metals? How does the metal construction compare with wood in respect to durability, when affected by vibration?

MR. STOUT:—As with any metal structure, that depends very largely on the design. For example, with any alloy-steel, if it is located at a point where there is too much vibration and overstress on the metal, there is bound to be an effect on the metal in time. The idea that duralumin becomes fatigued and crystallized by vibration originates from faulty design.

Regarding the metal floats, water soaks up in the wooden floats after leaving a seaplane standing in the water for 3 months. The actual soakage is about 600 lb. for an F5L hull. With a metal float, that weight could be saved and put into the pay-load at a certain amount per pound.

COMMANDER RICHARDSON:—We have not gone extensively into vibration tests at the Naval Aircraft Factory, but we have information that vibration is just as important with duralumin and no more important than it is with other metals. The elastic-limit must not be exceeded and so long as the stresses are maintained at a moderate value, there is no reason to suspect that duralumin will give way any more than any other metal.

MERRILL C. HORINE:—It has been my impression that duralumin is more economical to work in building a complete airplane than the conventional wood, wire and fabric.

MR. STOUT:—That depends upon the design. If an attempt is made to reproduce a wood, wire and fabric design in metal, it will cost a large sum of money; but if the structure is designed to suit the material that is to be used, and the cheapest production processes that fit that material are employed, the structure can be designed and made so that it will be as cheap or cheaper than the wood constructions, even in experimental ships. However, if the design calls for a lot of bumped work and toolmakers' work and equipment of that sort for the first airplane, the costs will run up far higher than those of any wood construction. It is all a matter of engineering.

THE NEED OF RESEARCH FOR THE TRACTOR INDUSTRY[1]

EVERY commodity must meet an economic need in our lives and in industry, to be worthy of, and to win, the support of the buyer and consumer. That economic need may be manifold. The commodity may fill a need that adds to the health and happiness of a community by giving people a diversion from the deeper and more strenuous problems of life, or it may be purely economic in its effect on industry itself. The tractor, no doubt, can be classed only as filling purely an economic need in industry and, in so classing it, we must be sure that it fills such a place. It may be going far and it may shock many to question that it actually is filling such a place to-day.

For some time I have had a vision of research for the tractor industry. Perhaps others are having the same thoughts. The possible development of this dream of the future calls for research of a somewhat different nature than is ordinarily talked of in laboratories and factories. To most of us, engineering research means a highly technical study of some phenomenon, the fuel problem or the study of the operation of some new device in its adaptation to work. The thought I have is back of all of this and means a restudy of the old problems.

Before the tractor became a factor in the agricultural world, the methods or processes of the agricultural industry were pretty well established; therefore, the one chief thought of the man who was trying to introduce power farming was to make a machine to replace the horse. So the machines were made and the adaptation of these machines to the work became more or less an after-study. To the credit of the tractor industry, this adaptation has improved year by year and a closer study has been made of farming operations to develop a more universal type of power unit. The big problem before the advocates of power farming to-day is how to put farming on an absolute power basis. Until this can be done, I believe that the advocates of tractors cannot really come into their own.

To get at this problem, therefore, I can see a research farm; not one where tractors alone are tested and not where present-day horse-drawn implements are pulled by tractors, but where there is truly a research into every farming operation to see if certain of them cannot be dispensed with and replaced by others more adaptable to power farming; where certain types of equipment can be changed or redesigned to make them more adaptable; and where different types of power unit can be tested and studied to see what type lends itself best to all farming operations. The final result of such a course of research should be, if necessary, an entire turning-over of present methods into a new and standardized type of power farming which may consist of a new series of operations on the farm, perhaps done by a new type of implement and a new standard form of tractor if that is what this research may develop.

The problem of research must, of necessity, be very complicated. It must start from the bottom, with the agricultural engineer if you please, and not at top with the mechanical engineer. Comparative studies must be made of the economics of different processes and the results obtained. Unit costs must be kept on every detailed operation and these plotted as accurately as the finest cost-study ever devised in an industrial plant. Engineers with inventive ingenuity must be prepared to meet every operation with some solution from the power standpoint; for, to make a success of power farming, it will not do to try to replace horses in all the operations and then have to have horses to help with the threshing or some of the other operations. Records to-day show that the tractor does not actually replace the horse on the farm anywhere near in proportion to its initial cost.

We all ask why the tractor has not become standardized as the automobile has become standardized, but it seems that it cannot become standardized in the same degree because of the varying problems it has to meet. Some day what I have outlined will be done. Shall the day come after years and years of gradual evolution, for there is one correct solution and our individual researches will gradually lead us to it; or shall the tractor builders cooperate in finding the solution and hasten the approach of that day as much as possible?

[1] From an address, before the Minneapolis Sections December 1921 meeting, by Emil F. Norelius, consulting engineer, Minneapolis.

Questions Answered by the Research Department

MANY of the inquiries answered by the Research Department are of much general interest and involve a considerable amount of thought and research. For the information of our members a few of them will be published each month. It is hoped that in this way those members who have not yet availed themselves of the facilities offered by the Research Department may be encouraged to take advantage of them in the future.

PISTON-RING MATERIAL

Question:—As you are well aware, the question of piston-rings and piston-ring design has been the bête noir of engineers and metallurgists. We are anxious to arrive at some definite conclusion, first, as to the best iron to be used and, second, as to the necessity for mechanically improving the physical properties of this iron after it is made. To do this, we would like to know just what each manufacturer of piston-rings is doing at the present time, the sort of iron that he is using, whether made in the electric furnace or a cupola; the analysis to which he is working; whether his rings are pot or individually cast and the means used to increase mechanically the tension of the rings after manufacture. Possibly you may know of someone who has this information available, but if not, do you not think that a committee on piston-rings and their standardization would do considerable good?

Answer:—A letter was sent out to a number of piston-ring manufacturers, quoting the substance of this letter. We had originally planned to group this information under the four heads suggested but found the material we obtained was of such value that we decided to quote the letters as they were received, omitting the names of the manufacturers, believing that a copy of the letter in each case would be of far greater interest and value than an outline. Three of the replies that cover the main points of interest that were brought out are given below substantially as received.

We agree with your correspondent that the question of piston-rings and piston-ring design has been a "bugbear" for engineers and metallurgists, but we are not so sure that he will arrive at a definite conclusion unless he gives the piston-ring manufacturer something definite to work upon.

The problem that has confronted us, and we presume that it is common with other piston-ring manufacturers, is that the automotive engineer in many instances has expected the impossible from a conventional type of piston-ring. Engines are designed to accomplish well-defined results and the piston-ring, which is one of the most important factors in attaining these results, is very often given very little attention. All too frequently, blueprint specifications for piston-rings are determined in the drafting-room, rather than on the experimental floors, and when the size is thus determined the requirements are turned over to the purchasing department and a canvass is made of piston-ring manufacturers to determine where the product may be bought at the most advantageous figure.

Then, there are those among the engineering fraternity who have been sent abroad by their principals to observe European practice, and upon return to their own factory have adopted Continental methods that are, perhaps, in no wise suitable to American road conditions. Desirous of reducing internal friction to the minimum, rings of extremely light tension often have been specified by American engineers, giving little thought to the fact that American automobiles are expected to negotiate roads absolutely unknown to the European car-owner.

We believe that tension should be an inherent part of the ring, and submit as an obvious fact that the automobile engineer would not permit the use of any other bearing in his engine which, in order to fit, must of necessity be "Pittsburghed."

We know of no manufacturer of any importance in the trade now offering pot-cast rings. Some of the multiple-piece rings, because of their construction, must be made of pot castings, but multiple-piece rings, so far as our information goes, are not being used by automobile companies as original installation.

Prompt seating of piston-rings is desired on the part of many manufacturers, and various suggestions have been offered along this line. Some companies produce what is known as a turned ring, but in our opinion a ring that is soft enough to turn will continue to seat indefinitely and thus lose its tension within a very short time. To accomplish quick seating, we have devised what we call our Quick-Seating Ring. This comprehends the grinding of a channel in the central face of the ring approximately one-half its width and about 0.002 in. deep. This throws up a web on each side of the ring which wears down quickly and permits the ring and cylinder to attain harmonious surfaces within a comparatively short time. This channel ground into the central face of the ring does not throw it out of balance. Pressure is exerted equally over the entire ring width, but focused during seating on the raised edges and, as soon as these webs seat, further seating, except for normal wear, ceases automatically.

In our opinion the exact specifications for piston-rings must be determined by tests made by the engineering department of the apparatus in which the ring is to be used. It is, of course, important to determine the type of ring to be used. Some manufacturers favor the diagonally cut ring, but others have a pronounced leaning toward what is known as the step-joint type. Some manufacturers who have used the former contend that with the sort of help usually available for installation, and especially in service-stations, if sufficient tolerance is not allowed between the points of the ring the expansion due to rise in temperature will in such cases cause cylinder scoring, while the use of the step-joint type of ring can produce nothing except perhaps abnormal but uniform wear on the cylinder.

Tension or lateral wall-pressure is another important factor to be determined by the automotive engineer. It is desirable to reduce internal friction to the minimum, but some engineers have gone so far in this direction that oil-pumpers have been developed, and many devices have been employed to overcome oil trouble, whereas a few pounds additional lateral expansive effort in the piston-ring would have prevented the trouble. Tension may be secured in one of three ways; increase in width of ring, increase in thickness of ring or increase of amount to be removed when the ring is split.

There has been a tendency lately among some manufacturers to adopt a very narrow ring. Several 1922 engines are equipped with rings 1/8 in. wide, but these rings, so far as our information goes, will exert approximately the same radial stress as that of the 3/16-in. rings formerly employed. These narrow rings, we understand, are used almost entirely in pistons of aluminum alloy, the theory being that the narrow ring

approximately 1/8 in. in width will not wear the grooves as quickly as the 3/16-in. or wider ring.

Everyone in the piston-ring business has been making an effort to produce the best possible piston-ring material for this purpose. Opinions differ somewhat on some details, but generally all ring manufacturers are agreed that there are two main factors to be considered. They are: ability to wear and elastic properties.

Piston-rings operate against the cylinder-wall with a definite pressure at a high rate of speed. A material that will give the best results under these wearing conditions must be used.

The piston-ring acts like a spring, and for that reason must have a high yield-point and a high elastic-coefficient. Steel would be excellent for piston-ring purposes if we did not have the wear factor to contend with. Cast iron in cylinders, pistons and rings has proved to be the ideal metal for all-round purposes. Cast iron wears less than steel in piston-ring operations because it is made up of a steel matrix imbedded with minute particles of pure graphite; this graphite is a natural lubricant. Cast iron, however, ordinarily has very low physical properties, and our problem is to make a spring material with high elastic properties and yet retain the desirable wearing qualities of gray iron.

The effect of the metalloids on the carbon in gray iron is fairly well understood, in that most foundrymen have witnessed the variation in the percentage of the total carbon in iron separated out into combined carbon or graphitic carbon. Great varieties of structure can be produced from gray iron, providing the control of the melting and refining operation is flexible. Very good grades of gray iron have been made by exceptional cupola operation, but there are factors in the operation of the cupola that give it definite limitations.

For the last two years we have been making all of our ring castings by the electric furnace process. The results have exceeded our highest expectations. Our reasons for a change from the cupola process, which has undeniable price advantage, were as follows: Phosphorus and sulphur should be kept to a minimum in gray iron for piston-rings. The cupola adds from 0.02 to 0.04 per cent of sulphur during the melting operation. This is an amount equal to the total sulphur-content of good pig iron. It is difficult to continually obtain an extremely low phosphorus-content in pig iron, but with the electric furnace sulphur and phosphorus are no longer a problem.

The higher the total carbon in gray iron the higher will be the graphitic carbon, which, as previously stated, is very desirable as a lubricant. The form, however, that this graphitic carbon takes determines the strength or weakness of the casting. A high pouring-temperature, together with a perfect control of the silicon and manganese, is necessary to produce a high graphitic separation in finely divided particles. The term "close-grained gray iron" so often used is descriptive of this condition. The pouring temperature of gray iron melted in the electric furnace can be as high as desired, and duplicated exactly day after day.

The effect of oxygen on cast iron has occupied considerable attention during the last few years. For ordinary castings of the usual size, we believe that oxygen plays a very small role. But in casting individual piston-rings that weigh from 1/2 to 1 1/2 oz. the cooling is so rapid that oxygen in the metal becomes a serious factor. In the electric furnace no blast is used and, if the furnace is properly handled, any oxygen there might be in the charge will be reduced.

The design of piston-rings follows somewhat the general design of springs, and it is necessary to know the physical properties of the material to make the design complete. Standard test-bars of any size such as might be used in steel cannot be used in testing gray iron. The percentage of any total carbon separating into graphite or combined carbon will vary according to the size of the casting and for that reason a true test of the material for calculation purposes must be made on a specimen cast under identical conditions obtaining in a piston-ring casting. For that purpose we use a test-bar 1/4 in. wide and 5/32 in. thick. The bar is cast individually, and has a length of approximately 12 in. This bar is identical with the piston-ring except that it is straight and the piston-ring is curved. A very delicate transverse testing-machine is used to run the test. Small increments of loading are registered, and a dial indicator shows the deflection in thousandths of an inch. Test-bars in a transverse test show 23,000-lb. stress before taking any permanent set. The coefficient of elasticity shows upward of 15,000,000 and the transverse breaking-stress runs between 60,000 and 70,000 lb. per sq. in. In spite of the fact that these figures are based on a transverse test, you can see that they are very unusual. With this information it is very easy to arrive at really definite calculations as to the proper ring-tension, thickness, deflection and width.

We take it that your question regarding increasing the tension of rings after manufacture means operations such as heat-treatment or peening. We have never been able to get very far with any sort of heat-treatment on gray iron, and we do not know of anyone who has been successful. Steel is easily annealed, drawn, forged and hardened, but it has a maximum of 0.20 to 0.80 per cent carbon, all of which remains combined. Cast iron has 3.50 per cent carbon with approximately 3.00 per cent in a graphitic form. Free graphite is the principal cause for weakness in gray iron, and it is not susceptible to changes by heat-treatment except possibly to soften the casting further. Our experiences show that heating gray iron generally weakens the structure. We have in no case found any heat-treating process that strengthens it.

Cast iron cannot be successfully forged, drawn or hammered for the same reason that it cannot be heat-treated. The large percentage of graphite makes it granular, and any effort to peen it causes small hair-cracks where the peen strikes. This is not serious in the case of piston-rings peened on the inside, for the reason that the inside of the ring is in compression, and the outside of the ring is in tension while in operation. The hair-cracks are closed because they are on the inner section of the ring. Nevertheless, the hammering process does not and cannot improve the material.

All piston-rings must have an out-of-round shape when uncompressed, so as to fit properly when compressed in the cylinder. Rings are peened to get this out-of-round shape, and for no other purpose. The same is true of heat-treated rings. They are heat-treated so that they can be bent easily to the desired shape. A better process is to put the desired shape in a pattern and cast the ring with the out-of-round condition in it. Why heat or peen a ring if the same results can be obtained in a solid, unstrained material in the casting? The need is to produce a stronger material for piston-rings, and nothing should be done to the ring after casting that will not preserve the original strength or strengthen it more.

To answer your questions more or less in order, we will state first that the material used in our rings is high-grade Northern machine-cast pig iron of various analyses combined so as to produce in the finished product as near as possible the following analysis: graphite carbon, 3.10; combined carbon, 0.45; silicon, 2.75; sulphur, not to exceed 0.055; phosphorus, 0.55; manganese, 0.55. This composition is for rings under 5 in. in diameter. In larger sizes the silicon will run some-

what lower and the sulphur a trifle higher, the latter due to the fact that we use more of our own returns in the large rings.

We maintain a laboratory to analyze carefully all metal received and all coke, limestone, etc., and also analyze samples from each heat. The rings 5 in. and under in diameter should show a Brinell hardness of from 180 to 210; we endeavor to maintain it at 190. Our method of making this test is to use a ½-in. test-bar and after the scale is removed apply a pressure of 3000 kg. on a 10-mm. ball for 15 sec. We make this reading microscopically as to the width of indentation and provide for a check of this reading by determining the depth of the indentation.

Our metal is melted in a cupola and by carefully and accurately combining the grades of iron it is not necessary for us to doctor it to obtain the proper results. We have used the cupola ever since we have been manufacturing piston-rings and have still to be shown wherein an electric furnace will give any better results than we obtain. We pay a premium for the coke we use to get an extremely low sulphur-content, and buy the high-grade limestone and fluorspar for flux. Great care is exercised in the purchase and preparation of our molding sand and all of our patterns are made to the utmost accuracy, the foundry tools and equipment are maintained on the same basis as our machine-shop equipment.

Our rings are cast individually, whether they are of ⅞ or 40-in. diameter, which is our present range of sizes. We have never made rings from pot castings in the 10 years we have been in business. The castings are made in an out-of-round shape. In other words, a piece is inserted in the pattern that is equal in length to the segment we wish to remove for the slot. As this is done by merely slitting the pattern and swaging this section in, it slightly flattens the pattern at that point and you can readily see that, when we remove the corresponding section from the ring casting, our castings will resume a perfectly round shape when the ends are forced together. This is a method originated by ourselves and by it we obtain a ring casting in which there is whatever degree of spring we may require, and also one that, upon being closed to cylinder diameter, resumes a perfectly round shape. This is particularly important as we do not machine the inside of the casting, merely smoothing it upon a stone. This is a very desirable feature as we retain the skin of the casting in this manner, which as you know is the strongest and springiest part of the cast iron.

We have, therefore, in making our castings, provided an absolutely circular ring with a resiliency that will never fail, and whatever pressure we desire by varying the segment in the pattern that is afterwards removed from the casting. By thus producing a casting inherently containing all of the necessary properties such as pressure, truly circular form, and resiliency, it is absolutely unnecessary for us to use any mechanical means of increasing the tension, and we are unalterably opposed to the theory of disturbing the natural position of the molecules of the casting or the ring to produce a property that we obtain by maintaining the casting in its true and natural condition. Our rings, being perfectly natural in their final structure, can be opened to the degree necessary to put them over the piston and closed a great number of times without in any way interfering with the qualities desired. A ring in which a spring is artificially produced will not, we believe, show this same characteristic, as the particles of metal that have been rudely displaced from their natural position cannot permanently provide the same results as one that functions in its natural form.

Our machining processes are built around the peculiar design and construction of our castings and are of course as accurate as is commercially practical. We employ various tools, jigs, fixtures and automatic machines of our own design and believe that this branch of our work is maintained at a very superior standard. The greatest care is used in their inspection. It is an obsession with us to sacrifice whatever need be to maintain or improve the quality of our product.

GAGE-STEEL INVESTIGATION

In the program for laboratory work to be conducted at the Bureau of Standards the first thing undertaken in the recently projected gage-steel investigation was to determine the reliability of the Amsler wear-test machine. Test discs of S.A.E. No. 1020 steel case-hardened and S.A.E. No. 1090 steel hardened were made up by Pratt & Whitney Co. The case-hardened discs flaked in the machine mentioned and were consequently not suitable for determining its performance. As it is difficult to harden carbon tool-steel uniformly, an oil-hardening steel was made up into discs, hardened and used for preliminary tests of the machine.

As the progress of the wear tests has been rather slow, arrangements were made to get a supply of 1.10-per cent carbon, 1.40-per cent chromium steel for hardening experiments and wear tests and of 0.45-per cent carbon-steel for testing the wear of hard discs against soft. As the chromium-bearing steel is the most universally used gage steel, it is planned to make the most elaborate tests on it.

Some quenching experiments have been made to determine the characteristic curves, cooling power and reproducibility of the common quenching media; this was accomplished by finding calorimetrically the average temperature of a standard nickel cylinder after different times of immersion in the quenching bath. Cooling curves were thus obtained for quenching in water at 30 deg. cent. (86 deg. fahr.) with and without motion of the cylinder, for quenching in oil at 30 deg. cent. (86 deg. fahr.) without motion and with slow and fast motion of the cylinder, for quenching in oil at 10 deg. cent. (50 deg. fahr.), 100 deg. cent. (212 deg. fahr.) and 200 deg. cent. (392 deg. fahr.) without motion of the cylinder, and for cooling in still air.

The heat-treatment of several steels in the form of 4-in. cylinders similar to those recommended by the committee has been varied with the principal object of determining the effect of the rate of heating on the dimensional changes. Some of these cylinders showing large dimensional changes on hardening are being measured for time changes.

The length measurements are being made under the direction of the Gage Section. This section has also prepared an attachment to the millionth comparator to take 4 ± 0.003-in. blocks for measuring the changes on hardening and with time. The attachment includes an oil bath in which the specimens are partially immersed to secure temperature uniformity.

Advantages of Light-Weight Reciprocating Parts

By L. H. POMEROY[1]

BUFFALO SECTION PAPER *Illustrated with* CHART

AFTER pointing out that the general question of weight reduction is no exception to the fallacies that seem to have beset the development of the automobile from its earliest days, the author outlines briefly the problem confronting the automobile designer. The influence of the weight of the reciprocating parts on the chassis in general and the engine in particular is emphasized as being of greater importance than the actual saving in the weight of the parts themselves, it being brought out that the bearing loading due to inertia is really the factor that limits the maximum engine speed. Reference is made to the mathematical investigation by Lanchester in 1907 of the advantages of using materials of high specific-strength and the conclusions arrived at are quoted in full. A tabulation of the specific strengths of various materials used in automotive engineering practice is presented as showing the advantages of aluminum as compared with steel.

The savings in weight that are possible by use of aluminum without any sacrifice of strength are next pointed out. The stiffness of steel and aluminum sheets is compared as one specific instance of weight reduction and this is followed by an extended consideration of aluminum connecting-rods, including an analysis of the loading due to inertia throughout a complete four-stroke cycle, and a comparison of steel and aluminum connecting-rods on a weight basis. The advantages of using aluminum to secure the required stiffness in a connecting-rod because of its low density are emphasized, it being brought out as the result of a mathematical analysis that equal stiffness as compared with steel can be secured in an aluminum connecting-rod with about one-half the weight of the material. An extended comparison of steel and aluminum connecting-rods that have been in service is next presented. The production methods employed for steel connecting-rods are stated as being applicable to aluminum. The advantages of the combination of the aluminum piston and the connecting-rod are pointed out, it being stated that a saving of 15 lb. in this connection as compared with a cast-iron piston and a steel connecting-rod results in an overall saving of about 14 times this amount.

ALTHOUGH automotive engineering science is now arriving at the stage when there are few engineers who claim that weight is advantageous apart from the necessities of strength, the general question of weight reduction is by no means free from the fallacies and false theorizing that seem to have beset the development of the automobile from its earliest days. For example, the average salesman will assert with no little emphasis that it must require more energy to reciprocate a heavy piston than a light one and that the horsepower of the engine is correspondingly affected. This may or may not be true according to the design of the pistons and their relative friction, for it is very easy to conceive of a tight-fitting light-weight piston with many rings offering a greater resistance to motion than a slack-fitting pistion of twice the weight and say one ring. The point is, of course, that if any mass is put into motion and afterward brought to rest as in the case of a piston during its travel, no work is done apart from friction. In other words, the energy put into the piston to start it moving is given up by it during its period of slowing-down to rest. With even greater emphasis it is claimed that the use of light pistons reduces vibration. This again while true as a general proposition is by no means an immediate truism. Most engines run at their best when the inertia-pressure diagram is approximately midway between the compression and expansion lines of the gas-pressure diagram, and it is again easily conceivable that for an engine running at a constant speed increasing the weight of the piston might improve engine smoothness.

The case for light reciprocating parts, however, rests upon grounds which are overwhelmingly more important than the somewhat hair-splitting considerations mentioned above. Before presenting this in detail it is of interest to review briefly the problem before the automobile designer. With a Pierce-Arrow at one end of the scale and a Ford at the other it is only possible to generalize vaguely but there are certain things in common: first, an approximately equal passenger carrying capacity, although the seven-passenger Pierce-Arrow is usually occupied by three or four persons in luxury, while the five-passenger Ford is usually occupied by seven or more in acute discomfort; second, the capacity to traverse any road upon which the wheels can hold; and third, the maintenance with safety of at least the legal limit of speed. These three items can, of course, be supplemented but they cover more or less those chiefly related to road performance.

We have then in an automobile a passenger load supported upon a frame, axles and wheels, together with an engine for propulsive purposes. From the viewpoint of the total weight involved it is obvious that the passenger weight is in accordance with the dictates of birth and diet and not under the control of the designer. The remaining weight is determined chiefly by the selling price of the car, and the problem resolves itself into giving the public the maximum aggregation of virtues that will result in that combination of sales on the one hand and profit on the other, essential to commercial success and stability. This may be differently expressed by saying that the cost of material is the largest single item of the three factors of cost, namely, labor, overhead and material, and that any reduction of the material cost, that is any reduction of the weight and the dimensions, goes hand-in-hand with the reduction of labor and consequently of overhead charges.

As in everything else the more one pays the more one gets or at least expects to get. The man who buys a typical heavy car does so because he associates with such

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a car the cardinal virtues of reliability, comfort and first-class road performance and, as it is difficult to obtain these in any other way, he does not resent the relatively high running-costs involved.

It is. therefore, the business of the engineer to fulfill these conditions by applying his knowledge of engineering science and research to the reduction of weight and thereby of running costs without sacrificing one iota of reliability, luxury or longevity, and at the same time to reduce the cost of production so that engineering and economic ideals will progress together.

An examination of the components of an automobile indicates that the bearing surfaces and the weight are closely interrelated and that in orthodox construction at any rate it is possible to build down to a given weight only by reducing the bearing surfaces to the minimum. Throughout an automobile we find this holding true; a few examples are taken at random for purposes of illustration. In the engine the crankshaft dimensions control its own weight and the weight of the bearings supporting it; similarly with the camshaft, pistons, wristpin, tappets and other parts. The weight of the clutch is determined by the area of the friction contact; and that of the transmission by the dimensions of the gear teeth and centers, the desired stiffness of the gear shafts and the permissible load upon their bearings, which items in turn decide the dimensions of the case that contains them. The same remarks apply to the universal-joints, the rear-axle bevel and differential gears, the front and rear hubs, the brakes and the steering-gear. The essential difference between the heavy and the light car is in the difference between the factors of bearing wear and the rigidity of construction appropriate to the problem in hand.

The load upon any bearing is due in part to the weight arising from its own dimensions in order that it may be adequately supported and in part to the passenger and body load supported by the chassis as a whole and the power requirements thereof. For approximately the same passenger carrying capacity and body accommodation it is easily possible to have a variation in car weight of from 2400 to 5000 lb. Allowing that the weight of the body fitted to the lighter car is say 700 lb. and that on the heavy car it is say 1100 lb., the chassis weight of the light car becomes 1700 lb. and that of the heavy car 3900 lb., a striking example of how ideals differ in producing two articles professing to do approximately the same job. Actually, of course, they do not do the same job, if the average light car were put to do the work possible with the heavy car, it would give up the ghost very early. The larger shaft dimensions, bearing surfaces and supports in the heavy car have been shown by high-duty experience to be necessary.

While admitting and even claiming this, there have been developments during the past few years in the field of light alloys which profoundly modify the whole problem of automobile design and make it perfectly demonstrable that a wholesale reduction in the weight can be obtained with the same or even higher factors of safety and wear in the bearings, and without the slightest sacrifice of the stiffness of shafts and general chassis construction so essential to a car that is required to meet the most exacting conditions. The object of this paper is to show more particularly the advantages to be derived by attacking one part of this problem only, namely the reduction in the weight of the reciprocating parts and to leave to the imagination the possibilities that can be achieved when this is coupled with the general weight reduction referred to. As will be seen the weight of the reciprocating parts has an influence upon the weight of the chassis in general and the engine in particular, which is vastly more important than the weight saved in the parts themselves.

The extent to which engine dimensions are a function of inertia rather than of gaseous pressures is often overlooked. At very high speeds and part throttle, as when driving downhill, the inertia pressures can easily be much greater than those due to the explosion. On the other hand, when at full throttle and low speed, as when pulling on high gear uphill, the situation is reversed. It becomes of interest and importance, therefore, to obtain some approximate idea of the conditions under which the inertia forces are greater than those due to the explosion.

The capacity of a bearing to withstand wear between the limits of the oil being crushed out of the bearing due on the one hand to heavy pressure and on the other to being evaporated out by the heat generated at high speed, is measured by the product of its mean loading in pounds per square inch and its peripheral velocity in feet per second. This value in good automobile practice should not exceed 16,000. At very low engine speeds the inertia forces are negligible compared to the gaseous pressures, so that for a crankshaft say 2¼ in. in. diameter running at 400 r.p.m., or a peripheral speed of 4.7 ft. per sec., the permissible limit pressure would for a load factor of 16,000 be some 3400 lb. per sq. in., which is very much greater than that actually arising under such conditions. At high speeds, say 2800 r.p.m., with a peripheral speed of 32.9 ft. per sec. the allowable unit pressure would be some 485 lb. per sq. in., a value frequently attained and even exceeded in existing automobile engines.

It may be said that if the bearings of an automobile engine are designed to take care of maximum-speed conditions, low-speed conditions will take care of themselves. As suggested, many automobile engines are now running at speeds that are up to and in some cases above those permissible for bearing reliability and it is not too much to say that bearing loading due to inertia constitutes the real upper limit to commercially possible engine speeds.

The advantages of the use of material of high specific strength, or strength per unit weight, for reciprocating parts were mathematically investigated by Lanchester[2] in 1907, but like many investigations this was somewhat ahead of its time. His generalizations are much more important and applicable to engine design to-day than they were when they were written 14 years ago, and constitute a good example of how pure mathematical reasoning from fundamentals finds a definite application when empirical developments have cleared the way therefor. He pointed out that the limiting speed of engines is determined by the strength of materials and that if similar engines be compared it is possible to predict the relative safe speeds at which they can be run.

The gist of these conclusions is quoted below and the engineer is recommended to study the paper in full.

INFLUENCE OF CHANGES IN THE DENSITY AND STRESS ON THE HORSEPOWER DEVELOPED

We will now revert to the general expression

$$Hp = (\sigma^{1.5}/\rho^{0.5})\ l^2 \times \text{a constant}$$

and discuss the influence of changes in the physical attributes of the materials employed, i.e. variations of σ and ρ (stress and density).

Translated into ordinary language the expression shows that in similarly designed engines the horsepower varies as the 1.5th power, that is, as the cube

[2] See Proceedings of the Institution of Automobile Engineers, 1907, p 155.

of the square root of the stress, and as the square root of the density of the materials employed.

Now it is evident that the weight of the engine also will depend upon the variables and l, and for the conditions of geometrical similarity the form of this expression is

$$W = \rho l^3 \times \text{a constant}$$

so that the horsepower per unit weight, which is the quantity of most interest to us, will be

$$\text{Hp}/W = [\sigma^{1.5}/(\rho \times \rho^{0.5})] \times (l^2/l^3)$$
$$= [(\sigma/\rho)^{1.5} \div l] \times \text{a constant}$$

Let us denote the quantity σ/ρ by the symbol Φ, and term it the "specific strength" of the material; then we have

$$\text{Hp}/W = \Phi^{1.5}/l$$

We have now the question of weight saving in a nut shell. The above expression shows that to which I have already drawn your attention, the importance of subdividing the power unit by employing a multiplicity of cylinders of individually small size, for we have the horsepower per unit weight inversely as the linear dimension, the latter, l, being the denominator in the above expression. We can also see at once the importance of employing materials of high specific strength; the form of the expression shows that if we can, by employing all-round a higher grade of material, say of 10-per cent greater specific strength, we shall effect a saving of weight of approximately 15 per cent.

Of course, it is not always possible to effect an improvement in the quality of the material in every part of a machine, and it is of considerable interest to us to ascertain where and how the saving in weight is most usefully effected.

WEIGHT SAVING CONSIDERED IN DETAIL

Let us, to fix our ideas, suppose that we have at our command two kinds of material, one of which has just four times the specific strength of the other; and let two carefully designed engines be built to the same specification, one from each kind of material. Now it is evident that, part for part, the one engine can be built one-fourth the weight of the other. There may be some slight difficulties in design, owing to the slenderness of some of the parts, but we can brush this difficulty to one side by supposing the difference of specific strength to be wholly due to a 4 to 1 difference of density, that is, σ remains constant.

So far we have accounted for the Hp/W varying in the direct ratio of Φ only, but the one engine will not only be lighter than the other but it will develop more power, for its reciprocating parts will give rise to less inertia and the revolution speed can be increased. The extent to which the revolution speed can be increased is in the inverse ratio of the square root of the weight of the parts, or in the case in point the revolution speed can be doubled. Thus the horsepower of the lighter engine will become twice as great as that of the heavier one, or its Hp/W will be 4×2; that is, eight times as great, which is $4^{1.5}$ in accordance with the equation.

We thus see that on the former supposition of a 10-per cent improvement in the material, producing approximately a 15-per cent improvement in the power weight factor, 10 per cent of this improvement is due to the direct lightening of the engine and 5 per cent to the increased power derived from the higher revolution speed rendered possible.

It is thus evident that by far the greater importance attaches, relatively speaking, to the quality of the material employed in the pistons and connecting-rods, for these reciprocating parts do not usually exceed 10 per cent of the total weight of the engine, and attention given to this 10 per cent is of as much effect as similar attention devoted to any other 50 per cent of the engine. It is thus found advantageous to adopt the very highest class of material for pistons and connecting-rods. For some years past I have employed a high grade of nickel steel both for the connecting-rod stampings and for the blanks from which the pistons are turned and I believe that the results would justify even more attention still being paid to the reduction of weight in these organs.

A SECONDARY EFFECT

A secondary effect, which must not be lost sight of, results in a saving of weight which is not obvious from a mere inspection of equation for Hp/W.

We have seen that the change in the power-weight factor as due to $\Phi^{1.5}$ takes the form of a saving of weight in the direct ratio of Φ, and in an increase of power in the relation $\Phi^{0.5}$. But we may not want increased power; it is usually some stated power that is required, so that l^2 will require to vary inversely as $\Phi^{0.5}$, that is, l varies as $l/\Phi^{0.25}$. Substituting, we have

$$\text{Hp}/W \alpha\, \Phi^{1.75}$$

under the conditions of stated horsepower, that is to say, hp. = a constant. This may be expressed alternatively by saying that for a given horsepower, for an engine of given number of cylinders, the weight varies inversely as $\Phi^{1.75}$.

The first equation may be written in the form

$$\text{Hp} = \sigma \times \Phi^{0.5} \times l^2 \times \text{a constant}$$

In this form the σ relates to the stress in the working fluid that is the cylinder pressure; taking this as constant we have

$$\text{Hp}\ \alpha\ \Phi^{0.5}\, l^2$$

and when Hp is constant we have

$$\Phi^{0.5} \times l^2 = \text{a constant}$$

or

$$l\ \alpha\ (1/\Phi^{0.25})$$

which gives the same result as before,

$$W\ \alpha\ (1/\Phi^{1.75})$$

We thus see that the saving of weight to be effected by employing high-grade material is even more than we had hitherto concluded, so that a 10 per cent higher specific strength would give about 17.5 per cent, instead of 15 per cent as previously concluded. The earlier figure was perfectly correct so long as the linear dimension of the engine was the constant, instead of the horsepower.

USE OF ALUMINUM

In the searching that has occurred since the facts were recognized for materials in which the tensile-strength was high per unit weight, the manifest advantages of aluminum as compared to steel have been overlooked or not taken seriously.

The specific strength of the various materials commonly used in automotive engineering practice rank as given in Table 1.

From Table 1 it will be seen that the specific strength

TABLE 1—SPECIFIC STRENGTH OF AUTOMOTIVE MATERIALS

	Tensile-Strength, Lb. per Sq. In.	Weight per Cubic Foot, Lb.	Specific Strength
Forged Aluminum	60,000	180	332
0.20-Per Cent Carbon-Steel	80,000	490	163
0.35-Per Cent Carbon-Steel	105,000	490	212
3-Per Cent Nickel-Steel	170,000	490	348
Sand-Cast Heat-Treated Aluminum	30,000	180	161
Chill-Cast Heat-Treated Aluminum	40,000	180	212
Malleable Iron	45,000	480	94
Steel Castings	60,000	480	125
Hard Cast Bronze	35,000	540	65
Cast Manganese	60,000	540	112

of forged aluminum closely approaches that of a 3-per cent heat-treated nickel-steel, while cast aluminum is greatly superior in specific strength to any other cast material in common use. It follows, therefore, that by suitably increasing the dimensions of a part previously made in any of the other materials mentioned, except 3-per cent nickel-steel, it can be made in aluminum to give the same strength but to weigh considerably less. This argument applies directly to cases of pure stress, tension, compression and shearing. There is, however. another very important aspect of the case, which in practice annihilates the superior specific strength of the high-alloy steel, arising from the fact that where compound stresses are involved dimensions per se confer strength.

It is well known that in engineering design generally examples of pure stress are conspicuous by their absence. Even in the simple case of a nut and bolt nominally in tension, it is very doubtful if the nut can be tightened without involving some degree of bending due to the thread being non-axial.

A simple example of compound stress is that of a rectangular beam in which doubling the depth quadruples the load carrying capacity. It may be urged that this applies to all materials, as in truth it does, but a very large portion of the section in the region of the neutral axis of a beam is only lightly stressed compared to that at the top and bottom. This constitutes useless material and it is obvious that if the necessities of design compel useless material to be carried, as they often do, the lower its specific gravity the greater is the weight saving effected. It can, of course, be argued against this that it is possible to design the section of a beam so that this useless material is removed by the processes of manufacture, as in the case of rolled steel sections. Unfortunately, the processes of manufacture to attain the end of eliminating the useless material are limited in scope and expensive in application and this fact constitutes an exceedingly important claim for the engineering and economic advantages of the broadcast use of aluminum in its various forms.

The essence of the technique of weight saving is a study of stress distribution and the design of parts so that the material used is proportional to the load at any point. While it is practically impossible to attain this end, at any rate in most automobile parts, due to the limitations of fabricating processes, the best alternative is to use a material in which the useless portion is of minimum weight.

In automobile design, however, the question of the strength of the various parts is but one aspect of the problem. The majority of parts need consideration from the point of view of stiffness rather than strength.

The history of automobile design is that of the increase in the dimensions of important details to overcome vibration and whippiness. For example, automobile frames are now girders of very great strength, but the strength is entirely secondary to the fact that stiffness is the dominant consideration in the design of a frame so that a closed body can be mounted with a reasonable chance of being able to open or close the doors after 6 months of use. Similarly with axles, transmission cases, crankcases, crankshafts, and other parts, while these were found strong enough in ancient designs they were not stiff enough and their dimensions have been increased.

The strength of a beam (nearly every part of an automobile is a beam in one sense or another) is a function of the square of its depth and its safe tensile-stress, while its stiffness is a function of the cube of its depth and its modulus of elasticity. As this last expression is so intimately connected with the application of aluminum to engineering design it may be worthwhile to explain that the modulus of elasticity of any substance is the ratio of stress to strain within the elastic-limit, the stress being the load per square inch and the strain the extension or compression in inches caused thereby divided by the original length of the piece. In other words, the modulus of elasticity of a material is the load that would double the length of a bar of the material 1 sq. in. in section if the section remained constant. For the common engineering materials, such as steel, cast iron and bronze, the modulii of elasticity are approximately 30,000,000, 17,000,000 and 14,000,000 lb. per sq. in. respectively, while for aluminum it is about 10,500,000 lb. per sq. in. It is of interest to note that the modulus of elasticity is a function of the character of the material rather than of its precise analysis or tensile-strength. Thus low-carbon steel and the nickel-chrome steel-alloys that may vary in tensile-strength by hundreds of per cent do not vary 10 per cent in the modulus of elasticity. Similarly the bronzes and aluminum alloys take their modulus from their basic material and are relatively little affected by the materials that compose the various alloys.

Steel and Aluminum Sheets Compared

With the above in mind it is of interest to compare the stiffness of a sheet of steel with that of ordinary rolled aluminum. For similar supporting means, as in the panel of a door, for example, the deflection due to a load applied at any similarly situated point in each sheet is inversely proportional to the modulus of elasticity and the cube of the thickness. If the thickness of the steel sheet is say 0.04 in. and that of the aluminum 0.06 in., the relative deflections will be as

$$1/[(0.04)^3 \times 30{,}000{,}000] : 1/[(0.06)^3 \times 10{,}500{,}000]$$

or as

$$[(0.06)^3 \times 10{,}500{,}000] \div [(0.04)^3 \times 30{,}000{,}000] = 1.18 : 1$$

The aluminum sheet is, therefore, 18 per cent stiffer than the steel sheet and 50 per cent thicker. As the weight of a steel sheet 0.04 in. thick is approximately 1.6 lb. per sq. ft., while that of an aluminum sheet 0.06 in. thick is approximately 0.9 lb. per sq. ft., it will be seen that this *increase* in stiffness of 18 per cent is accompanied by a decrease in weight of 0.7 lb. per sq. ft., or 43 per cent.

The relative strengths in the above example depend upon the square of the thickness and the ultimate tensile-strength, so that if the steel sheet has an ultimate tensile-strength of 55,000 lb. per sq. in. and the aluminum 25,000 lb. per sq. in., the relative strengths are as $55{,}000 \times 0.04^2$ to $25{,}000 \times 0.06^2$, or as 90 to 88 in favor of the aluminum. With heat-treated aluminum sheet the advantage of aluminum in respect of strength compared to steel would be in the order of 2.4 to 1. This is a perfectly legitimate example of the weight saving that can be obtained by using aluminum in one of its simplest applications to the automotive industry. A survey of fabricating methods and of other metals in general use fails to suggest any other way of obtaining a given desired strength and stiffness with such reduction in weight.

It must be emphasized that by no means the least important aspect of the case for aluminum as a means of obtaining light-weight construction without any sacrifice of strength or stiffness is that practically all engineering design tolerates a vast waste of material in the interest of economical fabrication. For example, a brake-rod that

is threaded at each end has only the strength of the metal at the base of the thread, so that all areas in excess of this are useless. Similarly, a rolled steel beam supporting a floor has a constant cross-section instead of one proportional to the bending-moment applied.

With light-weight alloys the same conditions arise and there is of course in similar designs the same percentage of waste materials. The point, however, is that the absolute dead-weight of useless material in the region of neutral axis is of far less consequence.

The Aluminum Connecting-Rod

The enunciation of these general principles leads to the particular consideration of the aluminum connecting-rod. From the point of view of specific strength, it has been shown that forged aluminum is greatly superior to the steels used in the vast bulk of automotive engine connecting-rods where from the *strength* point of view a steel giving from 80,000 to 90,000 lb. ultimate tensile-strength has proved perfectly satisfactory. There is no doubt that the development of the forged aluminum connecting-rod follows logically from the aluminum piston, to which many makers have been forced to return, and for the same reason, the reduction of internal wear-and-tear in engines by 50 per cent, and the vastly improved performance obtained thereby.

The primary advantage of forged aluminum as a material for high-duty connecting-rods in high-speed internal-combustion engines arises from the fact that the chief cause of bearing wear and failure is the loading imposed upon the crankshaft and connecting-rod bearings due to the inertia of the moving parts.

The loading on the connecting-rod big-end bearings is composed of two parts; (*a*) that due to the fluid, the gaseous mixture in the cylinder, pressures during compression and expansion, and (*b*) that due to the inertia forces arising solely from the mass of the moving parts, the piston and the connecting-rod.

If the four strokes of the conventional four-stroke cycle are examined the following will be apparent:

(1) At the beginning of the induction stroke, the loading of the big-end and adjacent crankshaft main bearings is due to piston and connecting-rod inertia only and acts on the inner side of the crankpin, or that nearest the axis of the crankshaft

(2) Slightly before the middle of the induction stroke the piston inertia forces vanish but the full inertia of the connecting-rod big-end is exerted on the inner side of the crankpin

(3) At the end of the induction stroke the full effect of piston and connecting-rod inertia still is exerted on the inner side of the crankpin

(4) At the beginning of the induction stroke the full effects of piston and connecting-rod inertia still are exerted on the inner side of the crankpin

(5) At the middle of the compression stroke the piston inertia vanishes and the loading on crankpin is due to connecting-rod big-end inertia, the direction still being toward the inner side of the crankpin but slightly modified by the fluid pressure

(6) At the end of the compression stroke the loading on the crankpin is the difference between the fluid loading due to compression and the inertia pressure of piston and connecting-rod. Hence at the moment before ignition the crankpin is subject to its smallest loading at least at speeds of 1000 r.p.m.

(7) At the beginning of the explosion stroke the loading on the crankpin is due to the difference between the explosion pressure and the inertia pressure of the piston and the connecting-rod. At high speeds and part throttle these may also neutralize each other

(8) At the middle of the explosion stroke the preponderating load is due to the fluid pressure but the centrifugal loading due to big-end inertia remains

(9) At the end of the explosion stroke the loading of the crankpin is due to the sum of the inertia pressures due to the piston and the connecting-rod plus that due to the fluid pressure and is exerted also on the inner side of the crankpin

(10) At the beginning of the exhaust stroke the loading is on the inner side of the crankpin and due to piston and connecting-rod inertia

(11) At the middle of the exhaust stroke the loading is on the inner side of crankpin and due to the big end of the connecting-rod inertia

(12) At the end of the exhaust stroke the loading is on the inner side of the crankpin due to piston and connecting-rod inertia

With regard to the 12 positions of the crankpin mentioned, it will be seen that in three only, namely, the end of the compression stroke and the beginning and the middle of the explosion stroke, do the fluid pressures in any appreciable way counteract the effect of the inertia pressures. It will be seen also that the pressure on the inside of the crankpin bearing due to the centrifugal action of the big end of the connecting-rod is always present. This action is approximately that due to the weight of the big end of the connecting-rod, the weight obtained by placing the big end of the rod on a scale-pan, while the small end is freely supported in space. It should be noted also that at the top and the bottom of the stroke, the inertia pressure on the inner side of the crankpin is that arising from the whole mass concerned, or the complete piston *plus* the whole connecting-rod.

In general the inertia effect of the connecting-rod may be considered as if the weight of the big end as previously described is concentrated to the crankpin while the difference between this weight and that of the whole connecting-rod, or the weight of the small end, is concentrated at the wrist-pin. For example, a steel connecting-rod for a 3½-in. bore engine made as light as practicable weighs about 3.50 lb., of which 2.75 lb. can be reckoned as rotating mass and the remaining 0.75 lb. as reciprocating mass. In aluminum this could be reduced to 2 lb., of which about 1.5 lb. would be rotating and 0.5 lb. reciprocating mass.

A cast-iron piston for a 3.5-in. cylinder bore weighs about 2.25 lb. complete.

The effect of using an aluminum connecting-rod would in such a case reduce the big-end loading due to inertia, neglecting connecting-rod angularity, in the ratio of 2.75 to 1.50 at the middle of the stroke, where the piston inertia forces vanish, and in the ratio of 5.75 to 4.25 at the ends of the stroke, an advantage in the reduction of the load factor varying from 40 to 26 per cent, or a mean reduction of some 33 per cent.

The advantage thus obtained can be utilized by

(1) Reducing the width of the bearings by 33 per cent, which in turn reduces the overall length and the total weight of the engine

(2) Reducing the gear-ratio and running the engine at a higher speed, thus obtaining a better performance with the same factor of bearing safety

(3) Improving this factor in engines that are unduly supplied with bearing surface

In brief, the use of aluminum for connecting-rods affords a ready means of making great overall economies

in a new design and of allowing a considerable development of existing designs.

Connecting-Rod Design

The greatest difficulty in connecting-rod design is to give adequate support to the babbitt or other material in direct contact with the crankpin without an undue increase of the weight. This is not a matter of strength of the supporting means, either steel or aluminum, but of securing stiffness. In this connection aluminum is particularly valuable owing to its low density. For example, a connecting-rod to suit a 2-in. diameter crankpin must be bored out to at least a 2⅛ in. diameter, leaving 1/16 in. for babbitt. The mean thickness of the metal surrounding the babbitt in steel should be at least ¼ in., neglecting big-end bolt bosses.

The mass of such a big end in steel is, therefore, proportional, the difference between the squares of the outside and the inside diameters multiplied by the density, or

$$[(2.625)^2 - (2.125)^2] \times 0.28 = 0.7$$

The stiffness for the same internal diameter is approximately proportional to the cube of the mean thickness times the modulus of elasticity or

$$0.25^3 \times 30{,}000{,}000 = 468{,}750$$

Since the stiffness of a part is proportional to its modulus of elasticity, other things being equal, and as the modulus of elasticity of aluminum is 10,000,000, while that of steel is 30,000,000, it follows that to obtain the same stiffness of the big end in aluminum as in steel the thickness must be increased accordingly.

$$\sqrt[3]{(30{,}000{,}000 \div 10{,}000{,}000)} = 1.44$$

The equivalent thickness, therefore, in the above example becomes

$$0.25 \times 1.44 = 0.36$$

From this the relative weight of the aluminum big-end is as before proportional to the difference of the squares of the outside and the inside diameters multiplied by the density, or

$$[(2.845)^2 - (2.125)^2] \times 0.1 = 0.36$$

Thus it will be seen that the same stiffness can be obtained in an aluminum rod for about one-half the weight of material required in a steel construction.

Apart from the amount of metal required to give the desired stiffness of bearing support, the question arises as to the number of bolts for securing the big-end bearing-cap. As a rule four bolts make a much better job than two and incidentally need be no heavier.

It is important to provide plenty of bearing surface between the cap and its abutment on the connecting-rod. The proportions of the small end of the connecting-rod are dependent upon the vagaries of the designer of the wrist-pin, but it should be kept in mind that the weakest part of the wrist-pin in a vertical engine is at the top of the rod, through which an oil-hole is generally drilled. The metal at this point should be 2½ to 3 times as thick as at the sides of the wrist-pin.

Coming now to the proportions of the shank of the connecting-rod, namely that part between the big and small ends, the region is entered in which much high-class mathematics may be applied. This would be justifiable if engineers had any complete knowledge of the distribution of the stress in a connecting-rod; as such knowledge is only now being acquired in respect of the very simplest structures, experience is still the best guide. In the nature of things it is highly probable that stress distribution is more uniform in a ductile metal such as forged aluminum than in less ductile metals such as nickel-chrome steel.

In the present state of knowledge it appears to be perfectly safe to make the section of an aluminum rod that replaces a reliable steel rod such that the section in the aluminum rod is similar to that in the steel rod and of twice the area. All designers will realize the number of exceptions there may be to this rule and the desirability of consultation with the prospective suppliers of the aluminum connecting-rod forgings.

The effect of aluminum connecting-rods on crankshaft bearing design is important. Many engine builders use counterbalanced crankshafts to reduce main-bearing wear, particularly on the middle main-bearing. There are, however, certain distinct objections to this practice. In the first place, a counterbalance crankshaft is heavy and expensive, and in the second place it introduces a distinct liability to torsional periodicity, particularly in six-cylinder engines.

The necessity for such crankshafts arose from the bearing wear that was due in turn to heavy pistons and connecting-rods. The use of aluminum rods reduces very considerably the crankshaft skipping-rope action which causes bearing wear. It is safe to say that in combination with aluminum pistons as well, counterbalanced crankshafts are unwarranted and disadvantageous.

Comparison of Steel and Aluminum Connecting-Rods in Practice

An interesting example of the truth of the foregoing remarks is obtained by comparing the characteristics of a steel connecting-rod taken from one of the best four-cylinder engines produced and of an aluminum rod that has done many thousands of miles under most exacting conditions without a suspicion of failure. The only criticism that can be urged against the steel connecting-rod under discussion is that it is too light, particularly in respect of the metal supporting the babbitted shell in the big end. The steel connecting-rod is from an engine of 3⅜-in. bore and 5-in. stroke; the aluminum rod is from an engine of 4⅛-in. bore and 4¼-in. stroke.

The detail dimensions and weights are as follows:

	Steel	*Aluminum*
Length betweeen Centers, in.	12⅛	11⅝
Diameter of Big-End Bearing, in.	1⅞	2
Length of Big-End Bearing, in.	2¼	2¼
Diameter of Wrist-Pin Bearing, in.	⅞	1⅛
Length of Wrist-Pin Bearing, in.	1¼	1⅝
Total Weight of Rod, lb.	2.900	2.140
Weight of Big End, lb.	2.340	1.690
Weight of Small End, lb.	0.560	0.450
Weight of Piston[3], including Wrist-Pin and Piston-Rings[4], lb.	1,206	2.040
Total Reciprocating Weight, lb.	1.766	2.490
Total Reciprocating Mass, lb.	2.340	1.690

[3]Piston is made of aluminum in each case.
[4]The piston used with the steel connecting-rod has three piston-rings, while that used with the aluminum connecting-rod has four.

Now as previously stated the loading of the crankpin bearing due to the inertia forces is ascribable in part to the centrifugal effects of the rotating mass of the big end of the connecting-rod itself and in part to the reciprocating inertia-effects of the piston and the connecting-rod small-end. These latter apply at the top and the bottom of the stroke only, while the former acts continuously. The mean loading, therefore, is that due to the rotating mass of the big end plus *half* that due to the reciprocating masses, as these are fully manifested only at each end of the stroke and vanish about the middle thereof.

The actual pressures manifested are directly proportional to the weight of the parts in question, the square of the number of revolutions per minute and the stroke. In comparing the two connecting-rods, however, the speed of the engine may be neglected, as the comparison be-

tween the two rods at any engine speed will hold at any other. We have then the average loading of crankpin proportional to the stroke multiplied by the sum of one-half the weight of the reciprocating parts and the weight of the rotating parts.

For the steel rod this becomes

$$5\,[(1.766 \div 2) + 2.340] = 16.100$$

while for the aluminum rod the loading is proportional to

$$4.25\,[(2.49 \div 2) + 1.69] = 12.50$$

The advantage obtained in reduction of big-end bearing pressures by the use of aluminum connecting-rods is thus clearly manifest, especially when it is noted that in each case aluminum pistons of similar design were used. The argument, however, goes much further than this, as the engine in which the steel rods are fitted is of 3⅜-in. bore and 5-in. stroke, while that with the aluminum rods is of 4⅛-in. bore and 4¼-in. stroke.

The final result may be computed in terms of cylinder capacity or of piston area. In the former the inertia effects of the steel rod are proportional to $16.100 \div 45 = 0.368$, while for the aluminum rod this "figure of merit" becomes $12.500 \div 55 = 0.228$. Figured on the basis of piston area, the comparison becomes $16.10 \div 8.90 = 1.81$ for the steel rod and $12.50 \div 13.30 = 0.94$ for the aluminum rod.

These figures are sufficiently striking to justify attention, and would be even more remarkable if the engine with steel connecting-rods were designed to reduce the secondary unbalanced forces to the same extent by having the same ratio of connecting-rod length to crank-throw as that in which the aluminum rods are used. This inherent advantage of the short-stroke engine can, however, be thrown in and still leave the argument for the aluminum connecting-rod in its above convincing state.

Summing up, as between two engines, one 3⅜ x 5 in. and the other 4⅛ x 4¼ in., each using aluminum pistons and both of really modern design, the inertia effects that determine the capacity of the engine to resist wear-and-tear of the bearings are reduced by 38 or 48 per cent by the use of aluminum connecting-rods, depending upon whether the cylinder capacity or the piston area is used as a basis for comparison. If it is argued that in the case of the engine with steel rods the wear-and-tear is satisfactory from the user's point of view, the figures then show clearly the possibilities of reducing the size of the bearings and the overall length of the engine, and of the manufacturing economies in respect to the total weight of material required for a given result.

In addition to the above analysis of bearing loading it may be of interest to compare the strength of the aluminum and the steel rods. Considered as a strut, the strength of a connecting-rod is directly proportional to the moment of inertia of its cross-section at the point of maximum stress (which is approximately midway between the ends) and the modulus of elasticity of the material, and inversely proportional to the square of its length. The student will recognize the above as the basis of Euler's formula, which is used as a ready means of comparison for the reason that the relation of the length of automobile connecting-rods to their cross-section does not vary greatly in practice. The moments of inertia of the cross-sections of the connecting-rods are 0.024 and 0.090 for the steel and the aluminum rods respectively.

The relative load-carrying capacity is then $(0.024 \times 30{,}000{,}000) \div (12.125)^2 = 4900$ for the steel connecting-rod and $(0.090 \times 10{,}000{,}000) \div (11.625)^2 = 6650$ for the aluminum connecting-rod, or the relative strengths of the steel and aluminum rods are as 1 to 1.36, the relative piston areas and total explosion pressures being as 1 to 1.5. On this reckoning it will be seen that the aluminum rod is not proportionally so strong as the steel rod. On the other hand, the aluminum rod in question has been subject to most drastic running without a suspicion of failure and from all practical points of view is well up to its job.

The truth is that the loading of a connecting-rod is so complex that it is difficult to reduce it to calculation. If engines were run at full throttle continuously at low speeds, the explosion pressure would be the determinant of the design. Just as the speed increases to that at which engines normally run, so do the inertia effects cancel out those of the fluid pressure, while at the very partial throttle required to run an automobile at say 30 m. p. h. the inertia effects completely overwhelm those of the explosion. Further, the compressive stresses set up by the explosion are not nearly so harmful as the alternating stress induced by inertia, so that the above discussion in respect of the explosion pressure is of not much more than academic interest.

In practice it is difficult or impossible to stamp steel rods of sufficiently light section, and subsequent machining is necessary to obtain the best results in respect to strength and lightness. The steel rod in fact suffers from excessive strength and insufficient stiffness. Even if machining is resorted to in order to reduce the weight of the rod, the extent to which this can be done is practically limited to the shank of the rod and the resultant effect is small. For example, on the steel rod in question the reduction of the average section of the rod to 1/16-in. thickness instead of the average 7/64 in. aimed at in forging would reduce the weight by some 3 oz. only, about 6 per cent, an insignificant result compared to that easily attainable by the use of forged aluminum.

Manufacturing Consideration

The general methods of manufacture applied to steel connecting-rods are equally applicable to aluminum. In aluminum rods machining the shank to reduce the weight may be dispensed with, as the consequent reduction of the weight is negligible. Further, there is no necessity for bushing the small end of the rod, although such bushing may be desirable in the case of small-bore engines where the wrist-pin is necessarily short.

Similarly, with aluminum rods the same babbitted shells may be used as with steel rods, although this practice is regarded as mechanically deficient with either steel or aluminum rods. The use of the babbitted shell necessitates increasing the total weight of the big end to compensate for the additional diameter of the big-end bore necessitated by the shell. More important still, the heat generated by big-end friction *and* that conducted down the rod from the hot region near the piston, has to be dissipated through two oil-films, that between the babbitted shell and the crankpin itself and that between the outside of the babbitted shell and the connecting-rod big-end proper. As the whole object of a bearing is to dissipate readily the heat generated by friction, it is difficult to see why its conductivity should be reduced by 50 per cent.

It may be argued that the advantage of the babbitted shell lies in its capacity for ready replacement. While this may be true, the percentage of bearing failures when babbitted shells are not used is so small as to make it more satisfactory and economical to replace the whole connecting-rod. As in many other instances in automobile design, the provision made for replacement makes the replacement necessary.

On the above grounds the use of direct-babbitted con-

necting-rods is strongly urged. From the production point of view there is no more difficulty than in babbitting a bronze shell and there is the economy obtained by dispensing with the shell. Successful methods of babbitting aluminum connecting-rods have been developed from the points of view of a complete technical solution of the problem and of rapid economical production. The results of these methods are such that the babbitt in a connecting-rod can be removed only by melting or laborious chipping. There is a definite metallic fusion between the aluminum and the babbitt that it is practically impossible to obtain with steel or bronze.

In the foregoing it has been taken for granted that the virtues of the aluminum piston are generally recognized by engineers; however much they may differ as to whether these virtues are offset by disadvantages. The renaissance of the aluminum piston is beyond doubt, so that it is fair to assume that the aluminum piston has been found to possess a number of advantages. It may not be out of place to state that this is due to the following:

(1) The elimination of piston slap by the use of pistons capable of distorting under high temperature

(2) The reduction of wear by carefully finishing the surface of the cylinder bore and the development of piston alloys of a hardness comparable to that of cast iron

The combination of aluminum piston and connecting-rod allows for a weight reduction of at least 40 per cent in these parts compared with ferrous metals as now employed. The consequences are

(1) That the engine may be speeded up with safety in the ratio of $\sqrt{(100 \div 60)}$, or 30 per cent with the same bearing areas

(2) The bearing areas may be reduced by from 30 to 40 per cent and the engine run at the same speed

(3) Combinations of (1) and (2)

Working along these lines, the author has recently designed an engine with 3¼ x 5-in. cylinders in which the bearing load factor at 3200 r.p.m. does not exceed 14,000 with big-end bearings 1½ in. between the crank webs and 1⅛ in. in net width. With cast-iron pistons and steel rods the corresponding net width of big-end bearing would be about 1⅝ in. The saving in the overall length on a six-cylinder engine is, therefore, some 3 in. in respect of big ends alone, together with further saving of the same amount in the main bearings.

The saving in engine weight arising from this reduction of bearing surface is about 15 per cent, while the available engine speed and torque are all that is required for a car of the medium large type. In other words, a 240-cu. in. engine thus designed with a 4¼ to 1 gear-ratio is capable of doing the work of the average 300-cu. in. engine with a 4 to 1 gear-ratio.

The disposition of weight in an automobile chassis is roughly as follows:

	Per Cent
Engine	25
Frame	10
Wheels and Tires	12
Clutch and Transmission	10
Torque Member, Universal-Joints, etc.	2
Rear Axle	12
Front Axle	3
Radiator and Hood	4
Springs	8
Electric Equipment	6
Steering-Gear	2
Gasoline Tank	1
Miscellaneous	5

Of these approximately 25 per cent, notably the frame, wheels and tires, is substantially independent of the chassis weight in that they are dominated in design by body considerations. Treating the engine as a separate unit weighing 25 per cent of the total chassis, we are left with 50 per cent of the chassis weight varying in some degree with the engine torque. The multiplicity of considerations underlying the design of these parts precludes any definite statement of the extent of this variation, but it is probably safe to say that for equal rigidity of construction the net saving is proportional to the square root of the ratio of the torque under consideration.

Thus the weight of the transmission of a car with a 240-cu. in. engine compared to that of a 300-cu. in. engine, both developing the same brake mean effective pressure, would be as $240/300 = 0.89$, indicating an 11-per cent saving in this respect. Summing up, we have a weight reduction of 15 per cent in the engine itself, or some 3.75 per cent of the whole chassis, together with say an 11-per cent reduction in the weight of 50 per cent of the chassis, or 5.5 per cent of the whole; in all some 9 per cent arising indirectly from the use of aluminum connecting-rods and pistons. In the case of a chassis weighing 2400 lb. the net result is, on the above reasoning, a saving of 216 lb. The weight of the cast-iron piston and steel connecting-rod in a six-cylinder 300-cu. in. engine would approximate 36 lb., while their aluminum counterparts would weigh some 21 lb. Thus a saving of 15 lb. in pistons and rods results in an overall saving of about 14 times this amount.

To forestall criticism, no one is more aware than the author is of the vagueness of this estimate, due to the vast number of factors that enter into the problem. On the other hand, there is no doubt of the truth of the general proposition as to the enormous benefits to be derived by reducing the weight of reciprocating parts by the use of light-weight alloys. The limitations of this paper unfortunately preclude any discussion of the potentialities that lie in the application of these materials throughout the chassis. The fact that extended experience has shown them structurally suitable for the hardest worked parts of the whole chassis indicates the confidence with which they may be applied elsewhere.

THE DISCUSSION

DAVID FERGUSSON:—Mr. Pomeroy's paper is so convincing that it is somewhat difficult to give good reasons for not following his advice. However, there are a few points that I would like to call attention to. In the table giving the specific strength of various automobile materials, the figures of most consequence are those of the elastic-limits of these materials, rather than those of the ultimate tensile-strengths. If this be conceded, it changes the position of the forged aluminum materially. I believe I am right in stating that the elastic-limit of forged aluminum is about 25,000 lb. per sq. in. Its specific strength in relation to its elastic-limit is therefore 133. The elastic-limit of 0.35-per cent carbon-steel heat-treated to give 105,000-lb. tensile-strength will be about 80,000 lb. per sq. in., giving a specific strength of 163, or nearly 25 per cent greater than that of forged aluminum. The elastic-limit of 3-per cent nickel steel heat-treated to give a tensile-strength of 170,000 lb. per sq. in. will be about 130,000 lb. per sq. in., giving a specific strength of 263 or double that of forged aluminum.

A doubtful point in connection with this comparatively new material is its life or endurance. Mr. Pomeroy has had personal experience with this in service, which is the

real test, yet a test I had made about 3 years ago in a Stanton fatigue-testing machine gave very poor results, the specimen failing after 1092 blows of a weight falling 1 in. Common screw-stock of about 0.20-per cent carbon-steel stood 6000 blows. Forged aluminum, no doubt, has been improved since this test was conducted.

The hardness of forged aluminum is I believe only about 100 Brinell, compared with 230 for heat-treated 0.40-per cent carbon-steel. If this is so, is there not trouble due to the metal peening-out? This would result in the small end of the connecting-rod enlarging on the piston-pin if no bearing bushing were used, or in the bushing becoming loose, if one were used. Is there any trouble due to the big-end bolt-heads peening their way into the softer aluminum? Is it necessary to use a large steel washer between the connecting-rod cap and the nuts on the big-end bolts?

Is there not some trouble from the greater expansion of the small and big ends of the aluminum rod, giving these a greater clearance than desirable and so causing a knock? This is one of the troubles I have had with aluminum pistons in which the piston-pin floated in the piston. Unless these were assembled very tight when cold, there would be a knock when the piston warmed-up.

Is not the cost of the aluminum forging somewhat excessive? When I looked into this matter three years ago I found that the cost was so great that it would pay to use steel and machine the rod all over, as the saving in weight would then be very little, the only large saving being in the big end, as I considered, perhaps wrongly, that the cross-section of the rod should be three times that of the 0.35-per cent carbon-steel that is the material I have used on all medium-speed engines, the stiffness being as great as that of the higher-priced alloy-steels. The elastic-limit of this steel is three times that of the aluminum and the modulus of elasticity is about three times that of aluminum, while the specific weight of the aluminum is less than one-third that of steel. In making the comparison, I, of course, figured on a bronze bushing in both the large and small ends of the aluminum connecting-rod.

The saving in the length of Mr. Pomeroy's engine, due to the use of aluminum pistons and connecting-rods, is certainly very interesting. However, I have found that the length of a six-cylinder engine is largely controlled by the diameter of the cylinder bore; the wall thickness; and the space for water between the cylinder barrels, which I consider a necessity. The last named cannot be less than ¼ in. and should be more to satisfy foundry requirements. The diameter of the crankshaft must be so large to avoid excessive torsional vibration that the length of the bearings that the above conditions admit of are usually ample for all medium-speed engines.

I would like to hear more of the type of aluminum piston Mr. Pomeroy has had most success with, including how he avoids piston slap when cold and scoring when hot. I believe that there is a great future for the use of forged aluminum in automobile construction. Mr. Pomeroy has done much to show the way.

L. H. POMEROY:—Mr. Fergusson, with characteristic thoroughness, puts up the contra side of the aluminum versus steel argument. I cannot, however, let his figures on specific strength in terms of elastic-limit pass without comment. In the first place, the elastic-limit of any material is most difficult to ascertain. In fact, the only physical properties that can be determined with anything like accuracy are the ultimate stress and the elongation. It is upon the former of these that the vast bulk of safety factors are based. Without going into this very vexed question, let us see what happens if, instead of taking the elastic-limit as a basis for determining the specific strength, we take another quantity that is related thereto but more easily measured, namely the yield-point.

It is well known that by suitable heat-treatment carbon and alloy-steels can be made to give a very high ratio of yield-point to ultimate tensile-strength at the expense of elongation, but such high ratio and consequent brittleness by no means make the material more suitable for practical purposes. In other words, experience shows that elongation is a necessary characteristic of most materials of construction and that it must be obtained even at the sacrifice of a high yield-point. This in itself to a large extent invalidates Mr. Fergusson's basis of comparison unless such basis predicates the same elongation as may be obtained with wrought aluminum. The yield-point of a good wrought aluminum alloy with an elongation of 20 per cent is approximately 35,000 lb. per sq. in. with an ultimate stress of 60,000 lb. The specific strength in terms of the yield-point becomes therefore the yield stress divided by the weight per cubic foot, or $35,000 \div 189 = 185$. The physical characteristics of S.A.E. No. 1035 steel, having a carbon-content of 0.35 per cent, as given in the S.A.E. HANDBOOK show that when this material is treated to give a 20-per cent elongation it has an ultimate stress of 95,500 lb. per sq. in. and a yield-point of 64,500 lb., the specific strength in terms of the yield-point being $64,500 \div 490 = 132$.

Similarly, S.A.E. No. 2330 steel, which is a 3-per cent nickel 0.30-per cent carbon-steel susceptible of being heat-treated to give 170,000-lb. ultimate stress, possesses only an elongation of some 11.5 per cent in this condition. When treated to possess an elongation of 20 per cent the ultimate stress becomes 104,000 lb. per sq. in. and the yield-point 77,000 lb., the specific strength in terms of the yield-point being $77,000 \div 490 = 157$. These figures show then that the specific strengths of wrought aluminum, 0.35-per cent carbon-steel and 3-per cent nickel 0.30-per cent carbon-steel, all with an elongation of 20 per cent, are 185, 132 and 157 respectively, instead of having values of 133, 163 and 263 as given by Mr. Fergusson.

With respect to endurance or resistance to repeated stress, one cannot of course expect all the poor results to be found with steel only. I would only remark that there are few tests that seem to have provoked more argument as to their value than fatigue tests in general. In actual practice automobile engine connecting-rods can be made in aluminum with from 55 to 60 per cent of the weight of a steel connecting-rod, including bolts, babbitt, etc., and have stood up under the most strenuous conditions. In the aluminum car I have designed we run the engine at speeds of over 3000 r.p.m. with impunity and in a collective mileage on four cars of nearly 80,000 miles and on one car of some 40,000 miles there has not been a symptom of failure. Forty thousand miles with a gear-ratio of 4.25 to 1 is approximately 1780 hr. running at 1000 r.p.m., or $1780 \times 60 \times 1000 = 107,000,000$ revolutions, or 214,000,000 strokes. Allowing three stress reversals per cycle, this becomes over 150,000,000 stress reversals.

Some fatigue experiments made in a fatigue-testing machine consisting of a motor-driven crank and weighted cross-head may be of interest. At 1500 r.p.m. the wrought aluminum rod gave a life of 353 hr. when the crosshead itself broke, whereas the steel rod failed in one case after 25 hr. and in another after 45 hr. of running under identical conditions. The steel rod was of

smaller section but about double the weight of the aluminum rod.

So far as peening-out is concerned no trouble has been experienced, nor should there be any if sufficient metal is used around the small-end bearing. With steel rods, of course, a bronze bushing that does not peen-out is universally used, so that the comparison between the Brinell hardness of wrought aluminum and of 0.40-per cent carbon-steel is hardly appropriate.

Steel washers are certainly desirable and in fact necessary to avoid trouble arising from the nut of the connecting-rod bolt or lock-washer cutting the aluminum when being tightened.

The difference in expansion of aluminum in contact with steel is about 0.001 in. per in. per 100 deg. fahr. Given a good initial fitting of the wrist-pin and adequate lubrication, no trouble is experienced, although doubtless these conditions are more important than with a steel rod unless the aluminum rod is bushed, in which case the requirements are identical. It is not the least of the advantages of the aluminum rod that the usual bronze bushing in the small end may be eliminated.

The cost of aluminum rods nowadays competes with an unmachined steel rod if the big end of the aluminum rod is direct-babbitted, and is much less than that of a steel rod machined all over.

I think that Mr. Fergusson's remarks on overall engine length apply primarily to a T-head engine. Admitting his premises, there is no reason why the bore cannot be reduced, which shortens the engine and reduces weight to a greater extent than it is increased by the longer stroke required for a given cylinder capacity.

The aluminum pistons to which Mr. Fergusson refers are of the split-skirt type made by the United States Aluminum Co. These pistons are characterized by the nature of the piston skirt, which is split to allow for expansion, and by the section of the skirt which is designed to allow deformation to take place without causing the portion of the skirt that is in contact with the cylinder to go out-of-round.

A further important feature is the discovery of a process for making these pistons up to 150 Brinell hardness, which has a marked effect upon their resistance to wear. In the engine previously mentioned these pistons are put up with 0.004-in. clearance for a $4\frac{1}{8}$-in. bore and no trouble has been experienced from seizure or slap.

The successful aluminum piston, like the successful aluminum connecting-rod, cannot, however, be designed blindly. There is a definite technique of construction that has to be observed and those who imagine they can substitute aluminum for steel or cast iron without modification had better not consider the matter further.

MR. FERGUSSON:—Have you any trouble with scoring of the cylinder?

MR. POMEROY:—Not that I know of.

E. H. SHERBONDY:—Is the constant you give for loads of 16,000 Ricardo's or your own? And how was it arrived at?

MR. POMEROY:—The constant was arrived at empirically by Mr. Ricardo, and is in conformity with my own experience. It is the average loading or pressure taken on the crankpin bearing.

OTTO M. BURKHARDT:—I notice that there was rather a sharp line of demarkation between Mr. Fergusson's and Mr. Pomeroy's figures. They represent, so to speak, two different schools of design. Mr. Pomeroy is basing his calculations entirely on the tensile-strength and Mr. Fergusson is basing his calculations on the elastic-limit. As Mr. Pomeroy says, the elastic-limit is rather an indefinite figure, whereas the tensile-strength is well defined and can be obtained, even with crude testing-machines. All fatigue tests that I know of have invariably been formulated on the basis of tensile-strength. During some very interesting tests made at the University of Illinois, it was found that the formulation of the results can be made to better advantage on the basis of the Brinell hardness. There is a fairly definite relation between the Brinell hardness and the tensile-strength of heat-treated steel. This relation has been established by John Miller, the metallurgist of the Pierce-Arrow Motor Car Co., and is represented approximately by

$$\text{Tensile-Strength} = 500 \times \text{Brinell Hardness}$$

No similar relation can be given for the elastic-limit but a very similar relation can be given for the yield-point. It is, according to Mr. Miller,

$$\text{Yield-Point} = 550 \times (\text{Brinell Hardness} - 75)$$

From this it follows that, when factors of safety are based on the yield-point, a happy compromise can be obtained between the two schools here represented. Mr. Pomeroy has chosen the tensile-strength for the simple reason that there is no well-defined elastic-limit in the case of aluminum. Mr. Fergusson has chosen the elastic-limit because this can, through patient research, be found for steels, and Mr. Fergusson, I take it, is a sponsor for the use of steel. In the calculations that I have carried out I have found it most satisfactory to base the factors of safety on the yield-point where infrequent shocks are under consideration. Where fatigue is under consideration, it is advantageous to deal with the tensile-strength. I have analyzed somewhat further the relative merits of the metals here under consideration. If we denote the tensile-strength of ferrous metals by T_f and the tensile-strength of aluminum by T_a, the ratio between the two is

$$T_f/T_a = K$$

This relation indicates that ferrous metals are K times as strong as aluminum, although K may well be smaller than unity.

If we further take into consideration the specific gravity of the two metals, we have another factor that rather expresses the inverse of the previous factor, namely the density of aluminum relative to ferrous

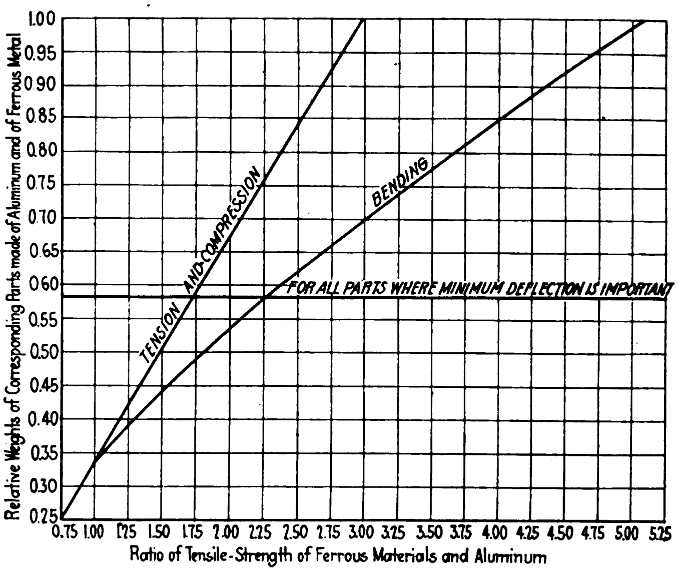

FIG. 1—CHART SHOWING THE RELATION BETWEEN THE RELATIVE WEIGHT OF CORRESPONDING PARTS MADE OF ALUMINUM AND FERROUS METALS AND THE RATIO OF TENSILE-STRENGTH OF FERROUS METALS AND ALUMINUM

metals. This factor is rather constant and may easily be taken as

$$C = 2.6/7.7 = 0.338.$$

In case the structural part be subject to either pure tension or compression, it is obvious that the product of the two factors, namely K times C, would represent the necessary weight to be employed for an aluminum or steel rod or bar respectively in order that in either case the same factor of safety for a steady load may be obtained. I have made special mention of a steady load, because the factor of safety would be an altogether different one in the case of a rapidly fluctuating or reversing load, as in such a case fatigue would have to be considered and aluminum is considerably inferior to steel as far as fatigue is concerned. The product $K \times C$ is a direct function of K only and it is an easy form to represent graphically. For instance, if we plot different values of K as abscissas and as ordinates we plot the product $K \times C$, as in Fig. 1, we can determine at a glance from the ordinates the weight of an aluminum part corresponding to an equally strong ferrous metal part.

In case of bending, we have a slightly different problem, as we then have to take into consideration the sectional modulus. We may agree on a section of let us say a width equal to three-eighths of its height and inasmuch as the sectional modulus is determined by the width and the square of the height divided by 6, we have in case of ferrous metals

$$3/8\ h^3 \div 6 \text{ or } 3\ h^3 \div 48$$

Denoting with h_a the corresponding dimension for an aluminum section, it is obvious that the section should be K times as strong as the ferrous metal section and consequently have the following relation:

$$h_a^3 \div 48\ K = h^3 \div 48$$

From this it follows that

$$h_a = h \sqrt[3]{K}$$

Inasmuch as the weight is proportional to the area of the section, we would have a relation between the weight of a ferrous metal lever and that of an aluminum lever,

$$(3\ h_a^2/8) \div (3\ h^2/8) = h_a^2/h^2$$

Substituting for h_a, we obtain

$$(h^2 \sqrt[3]{K^2}) \div h^2 = \sqrt[3]{K^2}$$

If we multiply this weight ratio by our previously obtained constant, $C = 0.338$, we have a direct relation between the weight of an aluminum lever and that of a ferrous metal lever, both being designed to give the same factor of safety.

In cases where the deflection is of the greatest importance we must bear in mind that the moment of inertia is the determining factor, and this is determined by the fourth power of the sectional dimensions. For instance, the moment of inertia of a section similar to that previously considered for bending would be expressed by

$$3/8\ h^4 \div 12 = 3\ h^4 \div 96$$

Inasmuch as the modulus of elasticity of steel is approximately three times as large as that for aluminum, it is obvious that the aluminum section should be such that the moment of inertia is three times as large as that pertaining to the steel section. This insures equal rigidity and may be mathematically expressed by

$$(3\ h^4 \div 96) \times 3 = 3\ h_a^4 \div 96$$

From this it follows that

$$h_a = h \sqrt[4]{3} = 1.3161\ h$$

For a comparison for weights, we have to consider again the areas and similarly as before we have

$$3/8\ h_a^2 \div 3/8\ h^2$$

After substituting for h_a, we obtain

$$[3/8 \times (1.3161)^2 \times h^2] \div 3/8\ h^2 = 1.732$$

In other words, 73 per cent more area is required for aluminum than for steel in order that both sections may be of equal rigidity.

If we now multiply this constant factor by our factor C, we obtain

$$1.732 \times 0.338 = 0.585$$

Or in a case where deflection is to be held to a minimum, an aluminum part of only 58½ per cent the weight of a steel part can be substituted with equal satisfaction.

In conclusion, I would say that with steels we have reached some sort of an obstacle between what can be had out of the steel in the laboratory and what the factory can handle successfully. We can heat-treat alloy-steels easily to give a yield-point of over 200,000 lb. per sq. in. However, it would be utterly impossible with our existing cutting-tools to handle a steel thus treated successfully in the factory. It is, therefore, necessary to machine steel parts while yet annealed and heat-treat them after machining. This, as we well know, involves scaling and distortion and requires grinding after heat-treating. The only alternative is to sacrifice the best that can be had from steel and be satisfied with the heat-treatment giving a yield-point of only half of what the steel is perhaps capable of, and steel so heat-treated can be handled successfully in the factory. No such limitation is encountered in the use of aluminum. In fact, we are far from getting aluminum hard enough. It is necessary to look forward to a new development of cutting tools to give us greater speeds so as to utilize thoroughly this outstanding property of aluminum that we know is easy cutting.

MR. POMEROY:—Mr. Burkhardt's contribution is an important supplement to my paper. I may say that in general the substitution of aluminum for steel is most easily achieved when the steel part is bounded by the atmosphere. In the case of a crankshaft, for instance, although it might be possible to make this in aluminum and save weight in itself, the necessary increase in the diameter to obtain strength and stiffness and the further increase in the weight of bearings, due to the increased size, would practically balance the initial weight-saving on the crankshaft itself. The case of a connecting-rod is, however, very different and there are usually no pronounced limitations in the space available for the increased section required. The case is similar with an automobile frame. The car to which I have referred has a cast aluminum frame that has stood up perfectly under the most arduous conditions of road use. Its weight is about 60 per cent of that of a corresponding steel frame. In this particular instance the strength is conferred by the dimensions, while the material is of relatively low tensile-strength.

E. O. SPILLMAN:—We have been experimenting recently with a piston with slots, the piston having the ordinary clearances. Some of the test pistons developed piston slap. I have not taken them down to find out what the trouble is, but I think these pistons have collapsed on the off-pressure side. Should we increase the weight of this piston or increase the Brinell hardness? Does Mr. Pomeroy use aluminum shims on the big end?

MR. POMEROY:—If the piston has a slap as you describe, I think that it has collapsed. If this is the case, more metal is needed. We certainly do not recommend shims in the large end of connecting-rods.

MR. SHERBONDY:—Mr. Pomeroy compared the piston and connecting-rod weights in the Essex engine with his

own on a basis of cubic-inch capacity. I believe that these should be based on the horsepower output at any given speed. What does Mr. Pomeroy consider a fair stress for connecting-rods? And what does he consider a safe deflection for them?

MR. POMEROY:—I agree with Mr. Sherbondy that the basis for comparison of the weights of connecting-rods in various engines should be the horsepower developed at any given speed, since this is in terms of cylinder capacity if the brake mean effective pressure is the same in the two engines. In the case in question, the brake mean effective pressure of the Essex engine is about 10 per cent higher than that of my own engine and this correction though small should be allowed for.

The safe stress for an aluminum connecting-rod is rather difficult to state as the loading of a connecting-rod at high speeds is very complex. Using more or less accepted methods of calculation, I try to keep the combined stress in the shank of the rod down to about 5000 or 6000 lb. per sq. in. at 3200 r.p.m. This can usually be done if the section of the steel rod is increased by 20 per cent. Each case, however, demands individual consideration. In many cases the steel rods in use are stiffer than they need be from forging considerations. In other words, most forged connecting-rods would be considerably improved if the shank section were reduced by machining.

A MEMBER:—What does Mr. Pomeroy think of the possibility of using metallic magnesium for the same purpose as aluminum?

MR. POMEROY:—The use of magnesium is now being developed for a considerable number of automotive parts. Its application to connecting-rods is, however, a matter upon which nothing can be said at the moment.

A. J. FITZGIBBONS:—Has a successful universal-joint been made of aluminum?

MR. POMEROY:—In the case of a universal-joint the difficulty which arises is that of fixing the aluminum forging to the shafts themselves. There is no real reason why this cannot be done but so far the circumstances have not arisen to cause this to be done.

MR. BURKHARDT:—How about the ring type of universal-joint?

MR. POMEROY:—I do not see why aluminum could not be used for that.

MR. SHERBONDY:—Another point is the question of expansion, where the connecting-rods and the diameters of the pistons are small. Here we have to deal with 3 or 4-in. diameters and run at from 0 to 160 deg. fahr. temperature, so that the change in size becomes a very serious factor. In some cases it may cause failure. In fitting pistons as tightly as Mr. Pomeroy recommends, the only way to fit them is to heat them before putting them into the cylinders.

MR. POMEROY:—The expansion of aluminum is twice that of steel. For a 2-in. diameter shaft, a 100-deg. temperature-difference between the steel and the aluminum would mean a difference in the diameter of 0.001 in. and it is difficult for me to believe that this would make any great difference.

OIL CONSUMPTION

(*Concluded from p.* 494)

although I still argue it is impractical to put anything in a piston-ring that is likely to lose its efficiency. If you make, as Mr. Lítle says, a more or less feathered edge, which you could do by putting a groove in there, it is probable that it will retain its sharp edge if you make that angle acute enough. But, judging from the condition of the rings as I have seen them, it is possible that the wear would gradually reduce the efficiency of that edge to a point where it would not exercise the control that it did when it was first installed.

Regarding carbon deposit, it is a peculiar thing that in some territory I have been visiting recently, despite the fact that they pump a large quantity of oil and use much oil, I did not find excessive carbon-deposit. With so much oil on it does the piston not become hot enough to form carbon? Or, is the effect due to some fuel-condition that happens to exist in that particular territory? I think the latter is the case, because in Detroit, with the same amount of oil, the carbon deposit has been excessive.

I was asked if compression is any better with multiple-piece rings than with plain rings. That is so largely a matter of the circumferential ring-fit and the trueness of the bore, that I think I cannot say authoritatively that there is any benefit to be obtained with the multiple ring at this time. I do believe, however, that a multiple-piece ring that expands vertically and fills the groove up and down is likely to retain compression after long use better than the plain ring.

WAGES AND PRODUCTION

MONEY, the medium of exchange, is a necessity to modern civilization, but, unfortunately, its use sometimes obscures or distorts industrial facts. Producers, especially wage earners, are apt to think their comforts would be doubled if their pay were doubled. To analyze this, let us suppose that at a given time the pay of all persons engaged in any gainful occupation were doubled, while production remains the same. Is it not clear that the cost of everything would be doubled, and that each one, with his double pay, would be able to buy only as much as he did before? On the other hand, let us suppose that the pay remains the same to each, and that at a given time by improved machinery or otherwise, the productive force of each worker is doubled. Is it not plain that the cost of everything would be reduced one-half, and that each one, on the same pay, would be able to buy twice as much as before?

The only practical way to double the reward to workers is to double their products. Larger production per man, through machinery and improved methods accounts for the fact that workers are to-day able to enjoy comforts that 50 years ago would have been impossible. If money were eliminated, many of the popular delusions would not exist. All would understand that if every worker turned out twice as many products as formerly, deposited them in a public receptacle, and then each carried away what he desired, in proportion to what he had deposited, each would carry away twice as much, and thus have twice as much to enjoy.—George H. Hull.

Selection of Machine-Tools

By A. J. Baker[1]

Detroit Production Meeting Paper

The problem of determining when to make a change of equipment by substituting new machine-tools for old, or special machines for standard, is carefully investigated. The fact that most manufacturers already have a surplus of machine-tools on hand on account of the demand for excessive production caused by the war makes the problem one not of providing for increased production but of decreasing its cost. The advantages and disadvantages of both special and standard machine-tools are weighed and the conclusion is reached that, although the ability of a special machine to produce pieces in fewer seconds is usually greeted with enthusiasm, other considerations such as the possible changes of the design of the pieces to be made, the inability to secure repair parts quickly, the dearth of skilled labor and the waste caused by employing inefficient help may make the change inadvisable. A method of analysis is given, by which an executive can determine how many cars of a particular model must be produced before a change of equipment can be justified.

I PROPOSE to lay down some general principles by which equipment can be scrutinized and the desirability of installing it determined. The title of this paper indicates machine-tools only, and since the application of these principles will be found to be greater with machine-tools than with any other type of equipment, we may let the title stand. I shall make a difference, however, because an executive, when equipping a plant, must select some items of equipment, not necessarily because he can effect a saving by them, but because he cannot produce a commercial success without them.

It is just as important for an automobile to have a body as to have a differential gear, but since the differential gear can be produced in a variety of ways and the sheet metal of the body in practically only one, the question of proper selection becomes much more important on the smaller and less expensive equipment required for the differential than for the heavy and expensive presses required for the body. Generally speaking, these principles apply to the selection of such machinery as lathes and vertical drilling, grinding, broaching, shaping, gear-cutting and milling machines, standard lines of wood-working machinery, hammers and the smaller sheet-metal presses. And in outlining those machines we must consider also those that were specially developed. Although they are described under other and special trade names, yet in view of the work produced these machines still come under the same general classification that is applied to the simpler standard machines.

A primary consideration that an executive must give to any purchase, be it design, material or equipment, must of course be its suitability for the purpose intended; another is the availability of a source of supply. Touching for a moment on this second point, we may look into the source of supply of the machine-tool industry during the last 10 years.

One extremely favorable aspect of the matter is that there is no apparent tendency of the machine-tool industry to become monopolistic in character. It is true that an association exists and it is also generally true that such associations ultimately must be paid for by the consumer. Many examples of this sort no doubt present themselves to you. However, since the manufacture of machine-tools apparently has always attracted a number of new devotees each year and since the various establishments range in size from those employing 50 men to those employing from 3000 to 4000, we can feel reasonably well assured of a diversity of interest and of sufficient competition to make it appear unlikely that any association can dictate to us as to the equipment we shall buy or the prices we must pay. In addition to this, the very remarkable growth of the machine-tool industry must not be forgotten. This Country is, without doubt, a greater producer of small and medium-size machine-tools than is any other. It does not stand proportionately so high in production of the heavier types of machinery, since much of this kind of equipment is produced in quantities so small that it does not lend itself to American methods of production and calls rather for the individual skill that is found more highly developed in the principal European countries. Nevertheless, as a whole, we have at hand all that is best in design and in workmanship of that class of machinery that is particularly applicable to the automobile trades, and which may be covered by lathes up to 36 in., planing machines up to 56 in., radial drilling machines up to 5 ft., milling machines up to No. 4 and gear-cutting equipment up to 48 in.

Besides we have an unquestioned superiority in the matter of those special highly productive machines that are developments of the standard equipment mentioned above and owe their inception so largely to the mass production of the sewing-machine, typewriter and automobile industries.

The Machine-Tool Industry

Diverting for the moment to the development of the machine-tool industry, prior to the war the number of men employed in the United States in the construction of machine-tools was approximately 33,000 and the output was valued at approximately $45,000,000 per year. At the peak of production during the war over 80,000 men were employed and the output was estimated as somewhere between $400,000,000 and $500,000,000 per year. These valuations, of course, do not express accurately the number of machines produced because the cost was increased very materially during the war. But, making due allowance for the non-employment prior to the war, the overtime work during the war, and the 100-per cent addition to the price of the machinery, it is reasonable to estimate that our machine-tool productivity of to-day, if stressed to its maximum, would be at least two and one-half times that of 1913; and it is further to be noted that the larger part of this increase is in the field of the small and medium-size machine-tools that I have already specified. Of course, a certain amount of increase and of development of production would have come in any case, through the normal processes of time and evolution. But no one I think will argue that the demand has as yet caught up with the unusual jump in machine-tool productivity that the war caused, nor will anyone be disposed to doubt that the vast majority of our factories

[1] Production department, Willys-Overland Co., Toledo.

during that period so added to their equipment that their normal demands, with a requisite allowance for the increase in equipment needed to meet their increasing trade, have for some time past been discounted. The great majority of the larger automobile factories possess surplus equipment, the full utilization of which is not likely to occur for some time to come. Some of this equipment has been so strained and injured that it must be replaced within a much shorter period than would be the case had it been operated under peace conditions. But, even allowing for this, I think you all will find that the factories you represent possess far more equipment of the standard types than can be utilized, particularly if the peak points in the production of automobiles could be ironed out. Consequently, the machine-tool builder, who looks toward a full utilization of his plant, will use all his engineering ability to develop some new machine, the output of which shall be so great that it will relegate to the discard all the machines previously produced by him, even though they may have been so well constructed and so well used that their productive life is still a matter of several years. He will do this on the theory, of the accuracy of which his sales department will endeavor to convince you, that you cannot afford to be without the newer machine because of the marked increase in production of the newer tool. If we could buy machines solely on the increase in production, the road would be easy, but this we should not do.

In the great majority of cases, assuming that a condition of a surplus of equipment does prevail, then the measuring stick by which we shall consider these offers is not increased production but decreased cost; and the two do not always go hand-in-hand. I am not dealing with a condition in which increased production is the essential thing from the viewpoint of the factory, because I do not believe that to be the case in most factories. My whole argument is built upon a belief that most of us have carried over from the war more machine-tools than we would by this time have acquired in normal times and under normal conditions, and that our problem is to determine whether we can afford to keep these machines or can dispense with them. Of course, if we are faced with an addition to our equipment that will permit us to produce more cars per day, our problem is greatly simplified since we would have only to select the machines that show the highest productive ability and apply to them the same general rules that will be laid down for the other case.

I think the foregoing should convince us that we have an ample source of supply; that it cannot become monopolistic; that the increased facilities at the disposal of machine-tool builders and their desire to utilize those facilities will lead them to the development of newer and better machines; and that, if we can exercise some influence, these machines may in the truest sense of the word be economical from the standpoint of the user. So much then for the market and the source of supply.

Cost of Labor Per Car

The third principle to be considered is the importance of a reduction in the cost of labor per car; you will please note that I do not say reduction in the price of labor. Our industry is so unfortunate as to be one in which the cost of labor is by no means equal to the cost of material. This fact makes it difficult to iron out our production schedules so that the same number of cars shall pass through our factories day after day. The demand for cars is more or less seasonal; that demand, reflected back to the factories, gives us our dull and prosperous periods, which we can not guard against by building up a stock of cars during the dull period, because of the tremendous inventory that we would accumulate by so doing.

Consequently, our industry offers its employes a relatively intermittent employment. To keep approximately the same number of employes throughout the year is given only to a very few of the larger shops and to a greater proportion of the smaller shops. Therefore, at certain periods, the employment department is called on to supply machine operators at a time when all other automobile manufacturers are clamoring for them. The result is that skilled operators cannot be secured and we must entrust our work to help of no skill or training in the manipulation of the machine. The automobile industry has never tackled in a large way the problem of instructing help, so that an adequate supply shall always be available. It has taken its skilled help from the other machine-tool-using industries, usually paying higher wages than most other industries could afford, and has never erected the machinery to replace the natural decrease in the available number of skilled men, or reciprocated by turning over to other industries trained men to take the places of those that have been taken.

As to the wisdom of this course there can be no question, but we are facing a condition, and those of us who select the machinery must bear in mind the type of help that may operate it. Machines that call for adjusting by hand during their operation, for accurate reading of dials or indicators, for careful setting up of the work in the machine, for a complex cycle of operations involving a developed mentality, all are to be decried, for not only do such machines limit the number of operators available, but under the stress of production the amount of scrap that the machines will produce is always entirely out of proportion to that produced by simpler equipment.

The Special Machine

A natural development of the above line of thought leads us to the special machine. By this I do not mean the single-purpose machine or, better still, the single-piece machine. There is a marked difference here that must not be lost sight of; and our failure as an industry to keep this difference clearly before us has led to the adoption and use of some machines that cannot be regarded as wholly satisfactory from an economic standpoint. In an enthusiastic endeavor to reduce time and to simplify operations, a number of machines have been developed that are useful for one piece only. They act as a deterrent from change in design and, generally speaking, are open to these objections.

(1) Their original cost must be great because the engineering and designing must be absorbed by the few machines that can be made on those models

(2) There is always considerable delay in producing them, so that the loss on account of the continued use of the older machine until the single-piece machine has been developed and tested out goes far toward overcoming the difference of the cost of labor between the single-piece machine and one of more general application that could be purchased as standard

(3) Since most machine-tools have been through a long process of development, it is certain that most special machines must pass through a long experimental period before they can reach the ideal set up by their designers, this adds further to the delay in obtaining full production

(4) The risk of break-down is much greater

(5) The delay in securing parts for replacement will be greater since all such parts are likely to be special
(6) The retention of an additional machine as assurance against break-down will often run the investment into large figures
(7) The difficulty of instantly replacing an operator
(8) The likelihood that special tools and fixtures must be designed and maintained
(9) The tendency of designers to incorporate elaborate tooling set-ups into such machines cannot be overlooked

All these are points of general application which are apt to be overlooked in the enthusiasm with which one views the statement that such a machine will turn out a given piece in so many seconds less than will a machine of a standard type. Often after a single-piece machine has been installed and satisfactorily operated, after its peculiarities of operation and tools have been fully understood and an organization has been trained that is able to maintain it in a state of efficiency, there is still the ever-present danger that a change of design may render the machine of no value whatever. There are to-day in the second-hand salesrooms so many of these machines that are without adjustments and are made so that they can produce only one piece that we need not go farther to see that we should step with caution.

Such machines have no value when divorced from the original purpose for which they were designed. A standard machine-tool, on the contrary, has a fixed market-value that depends upon its age and condition; and this value, carried on the books, can always be regarded as an asset. A special machine is apt to be carried on the books and to be depreciated by a nominal sum each year until a time comes when it is desired to turn the machine into dollars. A marked reduction in the inventory value must then be made through the inexorable law of supply and demand. A standard machine, on the other hand, can be transferred from one department to another and from one piece to another; its operators form a class and may be advertised for and hired under a classification, after the rates have been determined according to the location; the setting-up of the machine becomes a standard operation; the design, purchase and maintenance of the tools are all matters of routine. In the event of a break-down, though only one machine may be in use, it is possible to secure repair parts almost immediately from the builder and with a reasonable guarantee of interchangeability.

Between the single-piece machine and the standard machine-tool is the safe position. Some machine-tool builders already have recognized, and there is no doubt that others will recognize, the special needs of the automobile business. They have produced machine-tools in which the feeds and speeds cannot be changed at the will of the operator but can be changed at the will of the executive by the transposition of gears. These machines permit adjustments but only by the set-up man. They are constructed liberally along the lines of spindles, slides, gearing, pulleys and the like and preferably are over-designed for the power that they will consume. They are lubricated fully and automatically and do not require the use of the oil-can. In the hands of the operator they are only single-piece machines and as such may be designed with a reserve of power and a rigidity much in excess of the more universal type of machine because their application is not so constrained, and they can be regarded as a perpetual asset even though the model, or the detail of a model, were discarded and another took its place.

The same general line of reasoning will apply to tools and fixtures. Immense sums of money are spent in providing new tools when models are changed or improved. These sums may be and frequently are calculated, and the money is set aside to meet the expenditure. The maintenance and upkeep of the tools depend largely on their standardization, which is more difficult if single-piece machines are used, since the designer is apt to build his tools, as well as his machine, to suit the piece. If we deplore the reduction in the number of skilled machine operators, how much more should we deplore, and at the same time censure ourselves for, the reduction in the number of skilled tool and die makers.

It is true that an attempt has been made, and in some shops is well under way, to split up the tool-making and the die-making departments into various groups. But this, of course, is not applicable to the smaller shops, and at its best can only reduce the requirement and not abolish it. The tool designer is another of our operating units that each year is becoming more rare. I do not mean that we cannot get enough applications from tool designers, but I do say that a much lower percentage of capable men is to be found. The vision and the administrative capacity may be there, but the instruction or apprenticeship course that develops a high-grade machinist, a tool and die maker or a tool designer is very sadly lacking. Many of the fixtures that we apply to-day either to standard or to special machines bear evidence of having been made by a novice. There is a glorification of the complicated. The injunction to make two ears of corn grow where one grew before evidently has been taken literally.

If any of you have analyzed the tools and fixtures in your own shops and have compared them with simpler fixtures, not from the point of view of theory or design, but from that of practical application and of how much a part produced will cost, you will be ready to agree with me on this point. I have in mind a particular example in our factory, a certain brake connection in which a slot has to be milled to remove a binding strip that holds the two halves of a piece together during the casting process. The removing of this binding strip calls for no particular accuracy, requires no power and would be regarded as a simple operation to be accomplished on a hand milling machine with a very simple fixture; the total cost of the complete equipment would not exceed $500. Such an equipment could produce approximately 700 pieces per day. With an unskilled operator, a cheap tool equipment, no floor space and practically no tool-designing or tool-maintenance charges, two of these equipments would have taken care of all the requirements of our plant for a long time. Nevertheless, the actual equipment installed consisted of a very large rotary milling machine, upon the table of which was mounted a fixture that accommodated approximately 40 pieces, the fixture and the machine in combination costing about $6,700. One machine would, of course, take care of the requirements of the plant but, as an assurance against break-down, a duplicate equipment was ordered, so that the investment was about $13,400, or $12,400 in excess of the first mentioned equipment. Had the second machine not been ordered and a second fixture been deemed sufficient, there would still have been an outlay of about $9,000.

It would be easy to show that the big machine with one operator, on a basis of 600 cars per day, would produce a piece more cheaply than the two machines with two operators; but this is a condition that we, and I think most of you, do not experience. We may have a production of 600 cars per day for 1, 2, 3 or 4 months but we do not have it for 12 months. We could afford

to run those two small machines with two operators during our peak period, since the cheapest kind of help could be used on them, better than we could afford to spend the money that was spent for the expensive equipment. If we do the obvious thing, discard this casting and use in its place the stamping, we shall have on our hands two fixtures, one of which costs more than the full machine and fixture equipment that was considered in the first case. These fixtures have no resale value and reduce our inventory or assets by the amount of their original or depreciated value. Furthermore, such a machine, with its multiplicity of holding devices, will produce work that varies more than that which comes from the simpler machine. The floor inspector, passing from time to time, can take one of the pieces from the small machine and be sure that those that preceded it will have like accuracy. If a machine has a multiplicity of holders he cannot be so sure and the inspection charge will be increased. The scrap will be increased for the same reasons that make for a higher inspection charge. As the machine is run under the conditions of stressed production, you will find that some of the compartments are out of order and cannot be used, so the vaunted high production may be reduced, depending upon the number of compartments that are discarded. You may say that all this is bad management, that the compartment should not be permitted to get out of order, but we are talking as practical production men and we know that if a fixture at our peak period will produce three-quarters or seven-eighths of its true output, we are likely to continue in that state until a letting-up of the demand permits us to repair it.

I shall touch also upon the importance of avoiding break-downs that call for the services of skilled tool makers when such men are at a premium. If our equipment is of such a type that the average machinist can effect a satisfactory repair, we shall be that much ahead when the tight point comes and calls for immediate repair.

Skilled Workmen for Standard Machines

Another point that we must consider in the use of standard machines is the supply of skilled help that is yearly being turned out of the plants in which these machines are produced. Some of the machine-tool builders make a special point of training men, either in their own plants, or those of their customers, and of instructing them in the better handling of the machines and in the adjusting and setting-up, even to the point of effecting repairs. Such men are the nucleus around which a classification of labor is built; they call for no breaking-in and for that reason simplify the labor problem. The main reason why companies take this step is that most of the machines they produce were considered in the past to be somewhat more complicated than ordinary machines of that period and, to offset a high tool-repair or maintenance charge being made against the machines, which would of course react against their sale, they have seen fit to train satisfactory operators; of such operators we should avail ourselves thoroughly. These men should, wherever possible, be incorporated into a machine-repair gang, because it goes without saying that the more complicated the machine, the more skill and special training required to dismantle and repair it. Much harm may be done by unskillful attempts to repair a machine.

Basis of Purchase

Now, having in mind these general considerations, we come to the reasons for purchasing new equipment or new machines. The one most frequently encountered is that the new machine will save money. It is not always expressed that way; it is sometimes put that the new machine will reduce the labor cost or will turn out a piece more quickly than under the old method; but these are not the real things to be considered. The only satisfactory reason is to reduce the cost and not to reduce the labor charge or increase the production per man; and in this cost reduction appears the consideration of the items that I have already touched on. The second reason is to increase production; in other words, to turn out more parts per year or per season. In this case the consideration will be whether to put in more machines of the type already in use or to purchase some machine that was an improvement but of the same general type, or to get an entirely new kind of machine.

There is much to be said for maintenance standards. If the records show that the tool you have been using is up to the average in productivity, it would be foolish to change to another make, even if a somewhat greater output could be shown. Unfortunately many machine-tools of the same general classification differ so much in detail that the equipment of one cannot be transferred to another; the T-slots in the tables, the taper hole in the spindles, the thread on the spindles, the form of the tool-holder, the method of clamping the tools, the arrangement of the control levers all these differ very widely. You are therefore forced to make up special fixtures differing in some details from those that you have been using on other machines. This means that if a breakdown occurs, and you have planned for it and have the extra tools available, you will have had to carry just twice as many fixtures in excess of actual requirements. If you have more than one make of machine you will not have the facility of immediate interchangeability; you will not be able to transfer the operators with any degree of certainty; the foremen will spend much more time in the instruction of the men; the time-study department will have to make changes in the times, because the speeds and feeds may differ somewhat; and it may mean even an adjusting of rates. Further, you will have to keep in stock certain replacement parts for these machines; that, of course, will be doubled in number if you have in use more than one type of machine.

Now, if we decide to put in an entirely new type of machine, we should give the matter a very careful analysis. The blanks that we are using at the Willys-Overland plant are shown. They can be used with additional machines, as the need of such machines appears in our schedule of production and also with new machines that are brought to our attention through the production of our competitors' shops, the trade journals or the visits of representatives of the builders or vendors of the machines. On the first blank reproduced you will notice the usual information as to the name of the part and the company by which the proposal is submitted and the basis on which the figuring is done so far as the number of cars per day is concerned. From there on a detailed comparison is made that gives the current operation as against the operation suggested, the name and number of machines required, the time study, the time per piece, the production per machine per 8 hr. and the cost per hundred.

I fear, generally speaking, that is all the information that is considered in the purchase of new equipment; we do not figure on the cost per piece or per hundred expressed in wages paid out on the job. You will note, however, that we go a little farther; we have specified the present resale value of the machinery now installed. This is to be used in replacement and

Oct. 24, 1922.

PROPOSED METHOD OF MACHINING PART No. 300,239—CRANKSHAFT
Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 500 cars per 8 hr.

PRESENT METHOD — Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Face Sides and Turn Flange	18-In. American Lathe	5	14.0	112	$4.00	$535.00
Face Inside of Flange	18-In. American Lathe	2	42.5	340	1.30	$535.00
Space and Rough Turn Rear Bearing	18-In. American Lathe	3	27.5	200	2.00	$535.00
Finish Turn Flange	18-In. American Lathe	2	39.0	312	1.40	$535.00
Under Cut Flange	18-In. American Lathe	2	55.0	440	1.00	$535.00
				TOTAL...	$9.70	$7,490.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Flange End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	6	10.8	92.8	$5.20	$3,143.50
				TOTAL...	$5.20	$18,861.00

REMARKS

Total resale value of machines for present method	$7,490.00	Present labor cost per hundred	$9.70
Total cost of machines for proposed method	$18,861.00	Labor cost by proposed method	$5.20
Total______on machines for______method	$11,371.00	Saving of labor per hundred	$4.50
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and floor space)		Machines figured for factor of safety	
		Cost of taking down and installing new equipment	
Tools retired are good for production of______cars per day		Cars required to pay for new equipment (including machinery, tools and installation)	252,700
Value of tools retired (original cost of tools less 50% per annum)			

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Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 500 cars per 8 hr.

PRESENT METHOD — Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Space Gear End Bearing	18-In. American Lathe	2	56.0	448	$0.99	$535.00
Rough Turn Gear End	18-In. American Lathe	2	42.5	340	1.30	$535.00
Finish Turn Gear End	18-In. American Lathe	2	55.0	280	1.58	$535.00
Neck and Chamfer Gear End	18-In. American Lathe	1	69.0	552	0.80	$535.00
Under Cut Gear End	18-In. American Lathe	1	162.0	1,296	0.39	$535.00
				TOTAL...	$5.06	$4,280.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Gear End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	3	23.6	202.9	$2.35	$2,883.50
				TOTAL...	$2.35	$8,650.50

REMARKS

Total resale value of machines for present method	$4,280.00	Present labor cost per hundred	$5.06
Total cost of machines for proposed method	$8,650.50	Labor cost by proposed method	$2.35
Total______on machines for______method	$4,370.00	Saving of labor per hundred	$2.71
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and floor space)		Machines figured for factor of safety	
		Cost of taking down and installing new equipment	
Tools retired are good for production of______cars per day		Cars required to pay for new equipment (including machinery, tools and installation)	164,304
Value of tools retired (original cost of tools less 50% per annum)			

Oct. 24 1922.

PROPOSED METHOD OF MACHINING PART No. 300,239—CRANKSHAFT
Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 300 cars per 8 hr.

PRESENT METHOD

Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Face Sides of Flange and Turn Face Inside of Flange	18-In. American Lathe 18-In. American Lathe	3 1	14.0 42.5	112 340	$4.00 1.30	$535.00 $535.00
Space and Rough Turn Rear Bearing	18-In. American Lathe	2	27.5	200	2.00	$535.00
Finish Turn Flange	18-In. American Lathe	1	39.0	312	1.40	$535.00
Finish Cut Flange	18-In. American Lathe	1	55.0	440	1.00	$535.00
				TOTAL...	$9.70	$4,280.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Flange End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	4	10.8	92.8	$5.20	$3,143.50
				TOTAL...	$5.20	$12,574.00

REMARKS

Total resale value of machines for present method	$4,280.00	Present labor cost per hundred	$9.70
Total cost of machines for proposed method	$12,574.00	Labor cost by proposed method	$5.20
Total________on machines for________method	$8,294.00	Saving of labor per hundred	$4.50
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and floor space)		Machines figured for factor of safety	
		Cost of taking down and installing new equipment	
Tools retired are good for production of_____cars per day		Cars required to pay for new equipment	160,530
Value of tools retired (original cost of tools less 50% per annum)		(including machinery, tools and installation)	

Oct. 24, 1922

PROPOSED METHOD OF MACHINING PART No. 300,239—CRANKSHAFT
Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 300 cars per 8 hr.

PRESENT METHOD

Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Space Gear End Bearing Rough Turn Gear End	18-In. American Lathe 18-In. American Lathe	1 1	56.0 42.5	448 340	$0.99 1.30	$535.00 $535.00
Finish Turn Gear End	18-In. American Lathe	1	35.0	280	1.50	$535.00
Neck and Chamfer Gear End	18-In. American Lathe	1	69.0	552	0.80	$535.00
Under Cut Gear End	18-In. American Lathe	1	162.0	1,292	0.34	$535.00
				TOTAL...	$4.93	$2,675.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Gear End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	2	23.6	202.9	$2.35	$2,883.50
				TOTAL...	$2.35	$5,767.50

REMARKS

Total resale value of machines for present method	$2,675.00	Present labor cost per hundred	$4.93
Total cost of machines for proposed method	$5,767.00	Labor cost by proposed method	$2.35
Total________on machines for________method	$3,092.00	Saving of labor per hundred	$2.58
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and floor space)		Machines figured for factor of safety	
		Cost of taking down and installing new equipment	
Tools retired are good for production of_____cars per day		Cars required to pay for new equipment	116,240
Value of tools retired (original cost of tools less 50% per annum)		(including machinery, tools and installation)	

wherever additional machines are installed; and against it we place the cost of the new machinery. In the tabulations at the bottom of the chart we show the total expenditure incurred, balancing the resale value of the machinery to be discarded against the expenditure required for the machinery to take its place. We add to this the cost of the new tools required for the proposed method and, if an increase in the floor space is required for the new tool, that also appears. Each foot of floor space carries a certain charge that varies with the building, and includes the items of power, light, heat, water, insurance and the like; in other words, a floor-space charge that is not an overhead charge. Below this is an item that may be considered only when we intend to change over to reduce costs, as it gives the production per day that the tools to be retired if the contemplated action is taken would be good for. This figure must be considered in connection with the actual labor cost of the parts, because an executive passing on this matter must know the contemplated production, as he would incline favorably toward the new equipment if he found that the old equipment were taxed nearly to its productive limit.

Below that appears the value of the tools retired, which is the original cost of the tools, less 50 per cent per year. In other words, if the special tools and fixtures had been used for a few months only, we should depreciate our inventory by the full value, or cost, of the equipment at a figure that would be carried on our books, and at the end of the year that equipment would be written off 50 per cent, the next year 50 per cent of the remainder, and so on.

A very material reduction thus takes place in the inventory value that avoids the piling-up of a so-called asset that really is no asset at all. Nevertheless, if no consideration is given to the inventory value of these tools, you might by carrying out the matter to a ridiculous extreme, wipe out in 1 day from the book value of the stock the whole item that is classified as small tools and fixtures and have nothing to show for it. Such action, of course, would involve you at once with the accounting department, and be very poor business. I do not wish to convey the impression that we actually write off our small tools, jigs and fixtures at the rate of 50 per cent per year, but for the purpose of figuring against contemplated installments, particularly if it is for the purpose of effecting reductions in cost and not of taking care of extensions in volume, this makes a rather satisfactory arrangement.

The next two items show the labor cost at present as against that of the proposed method, and the saving in money in labor charges. The next item, which is again an intangible one, shows the number of operators that would be eliminated by the proposed method. This is a matter in which the factory executive and the employment manager are vitally interested. Labor troubles and short labor markets will always be with us, and the larger and more unwieldy the business becomes in point of the number of employes, the more likely are we to have trouble in procuring and maintaining an adequate labor supply. The proportion of machines taken into account in determining a factor of safety and the cost of taking down and installing the new equipment are then listed, after which the gist of the whole matter is expressed in the final line "The number of cars required to pay for the new equipment." It is of no use to say that the equipment will pay for itself in 1 year or in 2 years, because time is an uncertain element. Few men are able to estimate exactly how much work will be produced by a factory in 1 or 2 years. They may give a general average, but a progressive company should climb steadily. It seems better, therefore, to say that to pay for this saving a certain number of cars will be required. This gives two avenues for criticism, (*a*) the approximate time the cars will take to absorb this expenditure can be determined at the date of consideration by our knowledge of the expected output; (*b*) the number of cars that we are likely to make before the part in question is changed and the equipment is thrown out of use. When we are considering the installation of equipment that is made necessary by an increased production, this last item is not very important, but when we are approached for the consideration of some new machinery to take the place of that for which we already have spent our money and which already has been installed, this item becomes of paramount importance.

It has been very interesting to make these comparisons between some of the oldest equipment now in use in the Willys-Overland plant and some of the latest and most uptodate equipment that is being offered. We find that, even when a great reduction in time per piece is guaranteed by the machine builders and a good resale price is allowed for the old equipment, the actual number of cars required to pay for the new equipment is such that a rather effective damper is put on many installations that otherwise look as if they should be approved and authorized at once. I think that is because we consider in the installation of machines not only the cost of labor but also the inventory value of the equipment already in use.

As a matter of fact, after a study of 15 pieces as manufactured on our small car, substituting for the present equipment the latest and best standard machine-tools as specially developed for the automobile industry, we find that an expenditure of $132,000 is required to effect a saving per car of approximately 42 cents. Against this we have an expected resale value of machinery now in use of not more than $25,000; and this I believe is taking an optimistic view as the book value of the machinery stands at a very much higher figure than that given. In addition, some $13,500 worth of special equipment, book value, would have to be discarded, so that we should have to produce about 179,000 cars of this model before we would be justified in throwing out what is universally regarded as old machinery to give place to what is regarded as the very latest product of the machine-tool builders' art. There are, of course, some items in which a saving can be made in from 40,000 to 50,000 cars; some of them, however, run up to nearly 500,000 cars, and I will say further that we have not included in our study some of the more elaborate equipments to amortize the expense of which would involve a production of nearly 1,000,000 cars. This is on a basis of labor-cost saving only and does not include burden saving which, though theoretically applicable, would nevertheless hardly be reflected in the cost for a very long time.

I am well aware that much exception can be taken to this line of reasoning. It does not make the easiest road for the machine-tool builder to follow, but I think the figures cannot be controverted.

SUMMARY

In summarizing, I want to make these points

(1) There is a surplus, both actual and potential, of machine-tool equipment of the standard types

(2) Machine-tool builders are devoting their thought to high-production single-purpose machines of standard types

(3) The craze for special machinery is passing

SAVINGS TO BE ACCOMPLISHED BY INSTALLATION OF NEW EQUIPMENT BASED ON THE PRODUCTION OF 500 MODEL 4'S PER DAY

Part No.	Name of Part	Name of Present Machinery	Name of New Machinery	Saving of Floor Space, Sq. Ft.	Increase of Floor Space, Sq. Ft.	Book Value Special Equipment To Be Discarded	Book Value of Present Machinery	Resale Value of Present Machinery	Cost of New Machinery and Tools	Required Expenditure	Saving per Car	Cars to Amortise at 500 per Day
300,562	Inner Brake-Band	2—Silver Drills	1—No. 2 Avey Drill	8	......	$564.00	$225.70	$159.00	$2,800.00	$2,641.00	$0.0060	221,700
303,584	Brake Locker Lever Bracket	4—No. 310 Baker Drills 4—25" Barnes Drills 1— 8" Henry & Wright Drill	1—No. 2 Avey Drill 1—No. 3 Avey Drill	97½	......	1,885.00	3,410.26	3,332.00	12,325.00	8,993.00	0.0470	191,240
300,387	Steering-Knuckle Arm	2—No. 310 Baker Drills 5—No. 4 Bardons & Oliver Lathes	2—Daniels Automatic Multiple-Spindle Chucker	113	......	2,475.00	7,808.35	5,981.00	18,868.00	12,887.00	0.0257	253,266
300,239	Crankshaft	6—16x32 Landis Grinders	6—Wickes Heavy-Duty Crankshaft Lathes	......	54	......	16,097.76	15,943.00	27,000.00	11,057.00	0.0773	146,357
300,180	Nut Welding ½x28 Thread	1—Garvin Vertical Tapper	1—D12 Fox Multiple Drill and Tapper	......	......	35.00	411.57	364.00	3,233.00	2,869.00	0.0060	444,390
300,382	Front Axle	3—Espen Lucas Millers 2—25" Barnes Drills 3—No. 7 Becker Millers	1—Ingersoll Drum Type Axle Miller	243	......	3,393.00	9,860.46	10,344.00	14,386.00	4,042.00	0.0610	71,125
300,313	Piston Pin	4—10x28 Norton Grinders	2—Model B Sanford Centerless Grinders	70	......	......	9,692.84	5,826.00	5,000.00	......	0.0070	......
300,620	Brake Support Bracket	3—No. 1 Kempsmith Millers 1—No. 3 Kempsmith Miller	1—40" Ohio Rotary Tilting Miller	89	......	1,425.00	5,023.65	2,880.00	4,520.00	164.00	0.0170	120,897
305,166	Brake-Spring Bracket	2—No. 3 Kempsmith Millers 1—18" Cincinnati Miller 1—No. 7 Becker Miller	1—40" Ohio Rotary Tilting Miller	85	......	1,650.00	4,589.40	3,323.00	4,735.00	1,412.00	0.0120	134,437
300,385–6	Steering-Knuckle	4—25" Barnes Drills 1—No. 7 Becker Miller 1—No. 1 Tol. Hd. Miller	1—40" Ohio Rotary Tilting Miller	183	......	1,282.00	1,483.96	1,254.00	8,990.00	7,736.00	0.0400	202,549
300,385–6	Steering-Knuckle	2—16" American Lathes 3—3½x30 LoSwing Lathes	2—No. 9 LeBlond Multi Cut Lathes	42	......	337.50	4,679.24	3,454.00	4,564.00	1,110.00	0.0120	53,889
300,209	Camshaft	8—10x28 Norton Grinders	4—Melling Cam Turning Lathes	167	......	......	19,385.68	11,872.00	15,805.00	3,933.00	0.1020	42,761
300,675	Steering Arm	3—No. 4 Bardons & Oliver Lathes	Daniels Automatic Multiple-Spindle Chucking Machine	47½	......	318.75	4,683.00	2,511.00	10,050.00	7,539.00	0.0190	399,866
	TOTAL			1,144½	54	$13,365.25	$86,351.57	$67,243.00	$132,276.00	$65,859.00	$0.4220	156,064

(4) Special machinery will not always stand a financial comparison with standard machinery

(5) We are not, as an industry, facing our responsibilities in the matter of training operative help for tool and die work

(6) When considering new equipment we cannot disregard the inventory value of existing equipment and the loss that would be shown on our balance sheet if the existing equipment were converted from productive machinery into excess machinery that will have to be offered for sale

(7) The only good reason for installing new machinery, old machinery or any machinery, apart from those causes where a better quality is demanded, is to reduce the total cost of production of the complete part

TRADE RELATIONS WITH SOUTH AMERICA

SINCE the war ended the European belligerents have sought to regain the markets in foreign countries that had been lost during the 4 years of conflict. In South America the United States had built up a vast foreign trade, supplying in large part the needs of the people there that had formerly been met by the United Kingdom, France, Germany, Italy and Belgium. The question was whether, on the renewal of competition from European countries, the United States would be able to hold the position it had won.

Up to the outbreak of the world-wide depression in 1920, the United States maintained its lead practically unimpaired, and in that year 42 per cent of South America's imports came from the United States, as against 15 per cent prior to the war and 46 per cent in 1917. The relative importance of the United States as a market for South American goods naturally declined in the face of Europe's extreme need of foodstuffs and raw materials. In 1920, 33 per cent of South American exports went to the United States, against 42 per cent in 1917 and 20 per cent, on the average, for the years before the war.

The United States is one of the greatest manufacturing countries in the world, but is unusual in that it produces most of its own foodsuffs and even a surplus for export. The only foodstuffs that this country needs to import in large quantities are the products of a tropical climate, such as coffee, cacao and sugar. A large proportion of the exports of Argentina and Uruguay is made up of wheat, flour and meat products, in which the United States is self-supporting. This situation tends to limit the possible volume of imports from Argentina and Uruguay, although there is an active demand in the United States for the industrial raw materials of these countries, such as flaxseed, wool and hides and skins.

In Brazil the important foodstuff exported is coffee, of which Brazil produces three-fourths of the world's supply. More than half the entire crop is sent to the United States. The other major food products, cacao and sugar, are imported into the United States in large quantities. Brazil's principal industrial raw material, rubber, goes almost entirely to the United States. The same is true of the nitrate and hides of Chile, the coffee and cacao of Ecuador, Colombia and Venezuela, and the copper, vanadium, rubber and cotton of Peru. The United States took 45 per cent of the total exports from these countries in 1919 and 47 per cent in 1920, as compared with 20 per cent from Argentina and Uruguay in 1919 and 19 per cent in 1920. In 1920 the United States took more than 85 per cent of the total exports from Colombia, most of which are products of a tropical climate.

These countries and the United States are best prepared to supply each other's requirements, and the commercial bonds between them should become more and more close. Normally, the United States imports more goods from the countries of South America than it sends to them. For the 5 years prior to the war the balance of trade in favor of South America averaged nearly $90,000,000 each year. The normal balance of trade in the future will probably be as large as or larger than that before the war, but the amount will be offset to a great extent by so-called "invisible" exports from the United States that have come into existence during the past few years.

Two items will serve as examples. In 1913, only 4 per cent of the shipping clearing from ports of the United States for South America with cargo was of American registry. In 1920, 52 per cent was of American registry. In the latter year the aggregate amount paid in freight rates to the shipowners of the United States by South American consumers of American goods was probably about $10,000,000. A second factor is the large increase in South American securities held in the United States. As is natural in a new territory in process of development, investments of foreign funds in South America are large. While the major part of this investment is still British and European, the volume of American investments has been substantially increased. On July 1, 1922, there were outstanding more than $400,000,000 of South American Government, State, municipal and corporate bonds that had been issued in this country, on which the annual interest charge is approximately $30,000,000. Of these securities the larger part has been negotiated during the last 3½ years, as is evidenced by the fact that on Jan. 1, 1919, there were only $60,000,000 outstanding. During the 18 months ended June 30, 1922, there were floated in the New York market some $334,000,000 of South American securities, as against $80,000,000 of such securities in London.

The United States is in a more favorable position to hold and build up a large trade with South America than before the war. Valuable relationships have been established, while the physical equipment for carrying on foreign trade, such as shipping facilities, has been increased and improved. It is not to be expected that the United States will regain the abnormal proportion of the South American trade that it held in 1917, but it is apparent that its share will be permanently larger than before the war.—*Commerce Monthly.*

Reports of Divisions to Standards Committee

THE following 16 Division reports of the automotive and parts and materials Divisions of the Standards Committee will be presented for approval at the Standards Committee Meeting on Tuesday, Jan. 9, at 10:30 a. m. in the south convention hall on the fifth floor of the Engineering Societies Building. The reports are printed in this issue of THE JOURNAL to allow sufficient time for the Society members and others to study the reports and prepare comments and suggestions for consideration at the Standards Committee Meeting.

Although only members of the Standards Committee may vote on the adoption of the reports at this meeting, discussion is invited from everyone, whether connected with the engineering, production or servicing branches of the industry, interested in the adoption in practice of the standards proposed.

It should be borne in mind that the recommendations of the various Divisions do not necessarily apply to present practice, although they are based to a large extent on it. They are intended to be followed by the various industries to which they are applicable when changes in design or production make it economically possible to do so. Therefore the general adoption in practice of a given recommendation may be a matter of months or years depending upon the conditions involved.

A general resume of the work of the Standards Committee during the current year is given on p. 555 of this issue. The detailed procedure followed by the Standards Committee and its Divisions is outlined on p. 556 of this issue.

AXLE AND WHEELS DIVISION REPORT

Division Personnel

G. W. Dunhan, *Chairman*	Savage Arms Corporation
C. C. Carlton, *Vice-Chairman*	Motor Wheel Corporation
R. S. Begg	Jordan Motor Car Co.
T. V. Buckwalter	Timken Roller Bearing Co.
A. C. Burch	Courier Motors Co.
R. J. Burrows	Clark Equipment Co.
L. W. Close	Bock Bearing Co.
J. Coapman	Russell Motor Axle Co.
C. S. Dahlquist	Eaton Axle Co.
F. S. Denneen	Grant Motor Car Co.
F. W. Gurney	Gurney Ball Bearing Co.
F. P. Hall, Jr.	Salisbury Axle Co.
G. W. Harper	Columbia Axle Co.
G. L. Lavery	West Steel Casting Co.
A. M. Laycock	Sheldon Axle & Spring Co.
H. V. Ludwick	Budd Wheel Corporation
C. T. Myers	Consulting Engineer
A. L. Putnam	Detroit Pressed Steel Co.
O. J. Rohde	Wire Wheel Corporation of America
H. Vanderbeek	Formerly with Timken-Detroit Axle Co.

PASSENGER-CAR FRONT-AXLE HUBS
(Proposed S.A.E. Recommended Practice)

The Axle and Wheels Division was appointed in 1921 as the successor to the Axle and Wheels Subdivision of the 1920 Truck Division, the Subdivision being given the status of a separate Division due to the importance of the subjects coming under this classification.

The principal work of the Subdivision since its organization has been the continuation of the front-axle hub standardization program instituted by the Subdivision in 1920. Following the adoption of the S.A.E. Recommended Practice on Motor-Truck Front-Axle Hubs, the Division continued its work looking toward formulating a standard for passenger-car front-axle hubs.

In order that the large amount of work involved in the formulation of such an important standard might not fall entirely on a few members of the Division, a Subdivision and, later, two Subcommittees were appointed to deal with different phases of the work. The personnel of the Subdivision and Subcommittees that formulated the present recommendation and are continuing the work as to other necessary features is as follows:

SUBDIVISION ON PASSENGER-CAR FRONT-AXLE HUBS

C. T. Myers, *Chairman*	Consulting engineer
T. V. Buckwalter	Timken Roller Bearing Co.
Claude Greenhoe	Hyatt Roller Bearing Co.
F. W. Gurney	Gurney Ball Bearing Co.
E. R. Jacobi	Hayes Wheel Co.
A. M. Laycock	Sheldon Axle & Spring Co.
G. L. Lavery	West Steel Castings Co.
A. L. Putnam	Detroit Pressed Steel Co.
O. J. Rohde	Wire Wheel Corporation of America
A. S. VanHalteren	Motor Wheel Corporation

ROLLER BEARING SUBCOMMITTEE

T. V. Buckwalter, *Chairman*	Timken Roller Bearing Co.
R. S. Begg	Jordan Motor Car Co.
L. W. Close	Bock Bearing Co.
C. S. Dahlquist	Eaton Axle Co.
A. M. Dean	Rubay Co.
G. W. Dunham	Savage Arms Corporation
Claude Greenhoe	Hyatt Roller Bearing Co.
C. T. Hagenlocher	Wright Roller Bearing Co.
A. M. Laycock	Sheldon Axle & Spring Co.
A. L. Putnam	Detroit Pressed Steel Co.
O. J. Rohde	Wire Wheel Corporation of America
R. G. Schaffner	Bower Roller Bearing Co.
L. M. Stellman	H. H. Franklin Mfg. Co.
H. Vanderbeek	Detroit, Mich.
A. S. VanHalteren	Motor Wheel Corporation

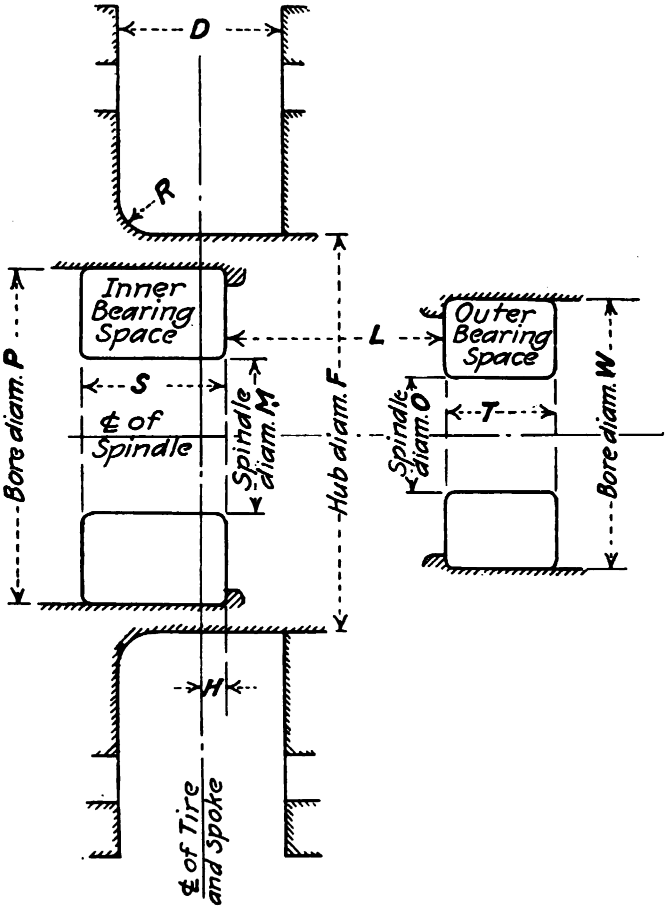
BALL BEARING SUBCOMMITTEE

F. W. Gurney, *Chairman*	Gurney Ball Bearing Co.
R. S. Begg	Jordan Motor Car Co.
H. E. Brunner	S. K. F. Industries, Inc.
E. R. Carter	Fafnir Bearing Co.
L. A. Cummings	Standard Steel & Bearings, Inc.
C. S. Dahlquist	Eaton Axle Co.
A. M. Dean	Rubay Co.
F. G. Hughes	S. K. F. Industries, Inc.
A. M. Laycock	Sheldon Axle & Spring Co.
A. L. Putnam	Detroit Pressed Steel Co.
O. J. Rohde	Wire Wheel Corporation of America
L. M. Stellman	H. H. Franklin Mfg. Co.
H. Vanderbeek	Detroit, Mich.
A. S. VanHalteren	Motor Wheel Corporation

The recommendation of the Subdivision was based largely on information obtained from passenger-car

manufacturers as to their current practice and from the files of the committee members. The Subdivision decided that five sizes of hub assembly, including the Ford type, would be sufficient to meet the requirements of the industry, the Ford type of spindle being included as it is suitable for light passenger-cars such as are being developed at the present time. The spindle diameters were selected from sizes most generally used and were classified according to tire sizes and approximate weights on the front axle as this method was considered the most logical.

As the actual practice followed for spacing the inner and outer bearings varied considerably, the method of spacing the bearings that was followed in formulating the present standard for Motor-Truck Front-Axle Hubs was used, this being to make the distance between the center-line of the bearings not less than 10 per cent of the tire outside diameter.



DIMENSIONS FOR INCH-SIZE TAPER-ROLLER BEARING HUBS

Hub and Spindle Number	Letter	RO	R1	R2	R3	R4
Inner edge of inner bearing to center-line of spoke	H[1]	−11/16	0	−3/32	−1/16	1/16
Inner bearing shoulder to outer bearing shoulder	L	3 11/32	2 1/32	2 11/32	2 11/32	1 11/16
Spindle diameter at inner bearing	M	1.1895 1.1890	1.3120 1.3115	1.3745 1.3740	1.4995 1.4990	1.6245 1.6240
Spindle diameter at outer bearing	O	¾"–16	0.9370 0.9365	0.9370 0.9365	0.9995 0.9990	1.1870 1.1865
Hub bore for inner bearing	P	2.7155 2.7140	2.6135 2.6120	2.9985 2.9970	3.1547 3.1532	3.2485 3.2470
Hub bore for outer bearing	W	1.9365 1.9350	2.1235 2.1220	2.1235 2.1220	2.4828 2.4813	2.8578 2.8563
Overall length of inner bearing	S	1.0425	11/16	11/16	1 1/32	1⅛
Overall length of outer bearing	T	11/16	11/16	11/16	11/16	1 1/16

[1] Minus dimensions indicate that the inner edge of the inner bearing is located to the right, or the side nearer the threaded spindle, of the center-line.

The inner edge of the bearing surface of the spindle shall be ¼ in. inside of the inner edge of the outer bearing.

Although it is the purpose of the Division to recommend hub dimensions for both taper-roller and ball bearings, it was decided at a joint meeting of the Axle and Wheels and the Ball and Roller Bearings Divisions on Oct. 4 that even though a considerable amount of work has been accomplished in establishing ball-bearing sizes for passenger-car front-axle hub applications, the work did not warrant submitting a recommendation at this time. As the work on roller-bearing applications had reached such a point that the boundary and spacing dimensions for the bearings and the corresponding spindle and hub dimensions could be recommended, such action was approved by the Divisions in joint session and subsequently by the Axle and Wheels Division. The recommendation completes a series of 10 applications of inch-size taper-roller bearings for front-axle spindles and hubs ranging from light passenger-cars to heavy motor-trucks, the sizes specified for the lighter type of motor trucks being the same for the heavier type of passenger cars. Therefore

The Axle and Wheels Division recommends for adoption as S.A.E. Recommended Practice the accompanying series of inch-type taper-roller bearings for passenger-car front-axle spindles and hubs.

BALL AND ROLLER BEARINGS DIVISION REPORT

Division Personnel

F. W. Gurney, *Chairman*	Manly & Veal
C. M. Manly, *Vice-Chairman*	Railway Roller Bearing Co.
J. T. R. Bell	Norma Co. of America
G. R. Bott	S K F Industries, Inc.
H. E. Brunner	Timken Roller Bearing Co.
T. V. Buckwalter	Fafnir Bearing Co.
E. R. Carter, Jr.	Bearings Co. of America
D. F. Chambers	Bock Bearing Co.
L. W. Close	Standard Steel & Bearings, Inc.
L. A. Cummings	Savage Arms Corporation
G. W. Dunham	Long Mfg. Co.
R. G. Hendricks	New Departure Mfg. Co.
F. G. Hughes	Gilliam Mfg. Co.
G. L. Miller	White Motor Co.
A. J. Scaife	Bower Roller Bearing Co.
R. G. Schaffner	Cadillac Motor Car Co.
W. R. Strickland	Hyatt Roller Bearing Co.
R. E. Wells	Gurney Ball Bearing Co.

BALL AND ROLLER BEARINGS DIVISION REPORT

METRIC-TYPE THRUST BALL-BEARINGS

(Proposed S.A.E. Standard)

In 1918 the Ball and Roller Bearings Division recommended for adoption certain inch tolerances for metric-type thrust ball-bearings, favorable action being taken by the Society in August of that year. The recommendation specified only the tolerances for various ranges of the boundary dimensions because full information as to foreign practice was not obtainable owing to the war. Early in 1921 information was obtained as to European practice

D. F. Chambers, *Chairman*	Bearings Co. of America
Frank Beemer	Nice Ball Bearing Co.
H. E. Brunner	S. K. F. Industries, Inc.
E. R. Carter	Fafnir Bearing Co.
F. Alton Collins	Auburn Ball Bearing Co.
H. H. Edwards	Bantam Ball Bearing Co.
S. A. Strickland	Imperial Bearing Co.
H. Wickland	U. S. Ball Bearing Mfg. Co.

and a Subdivision appointed to extend the present S.A.E. Standard shown on pp. C39 to C42 inclusive of the S.A.E. HANDBOOK, the personnel of which is given on p. 530. The work advanced until it was possible for a progress report to be submitted at the June meeting of the Standards Committee this year. Since the June meeting it has been possible for the Subdivision to obtain agreement as to the double-direction flat-face and self-aligning types.

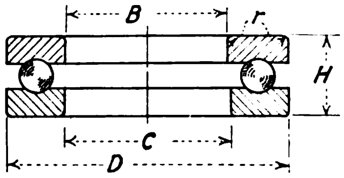
In the beginning, difficulty was encountered in that dimensions for the thickness and the outside diameters, as established by the several manufacturers, were not the same for corresponding bore diameters. These discrepancies were reconciled, dimensions having been generally increased rather than decreased to meet the existing individual standards, thus working toward the strengthening of thrust-washer sections and eliminating the weaker parts in the several series.

In the self-aligning series there was considerable variation in the thickness and the radii used in determining the curvative of the self-aligning seat. These differences have been eliminated and, in addition, dimensions giving the location of the radii have been specified. This is important, particularly to the users of self-aligning ball-bearings, as it is frequently desired to incorporate the self-aligning seat as an integral part of a mechanism.

In the proposed standards the present established dimensions of both foreign and American ball-bearing manufacturers were used as a basis for this work, but to obtain a consistent series of the double-direction type, it was found necessary to develop a new light series in both the plain and the self-aligning constructions. These correspond with the light series of the single-direction type. Therefore

> The Ball and Roller Bearings Division recommends that the accompanying series proposed by the Thrust Ball-Bearings Subdivision covering Light, Medium and Heavy Series for the Single-Direction Flat-Face Type, the Single-Direction Self-Aligning Type, the Double-Direction Flat-Face Type and the Double-Direction Self-Aligning Type of Thrust Ball-Bearings be adopted as an S.A.E. Standard.

The Subdivision report as received by the Division was accompanied by a minority report to the effect that the Subdivision report does not allow a sufficient amount of "land" in the retainer for good engineering practice. The Division recommended that the minority report be appended to the Division report in order that action would be taken at the Standards Committee meeting with a full knowledge of the questions involved. It was the consensus of opinion at the Division meeting that the dimensions finally chosen by the Subdivision allow for ample strength of separators or cages built in accordance with



SINGLE-DIRECTION, FLAT-FACE TYPE
LIGHT SERIES, METRIC SIZES

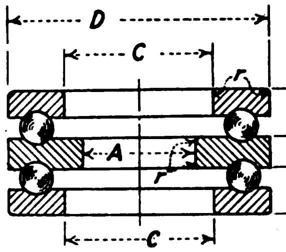
Bearing Number	Inside Diameter (B)		Diameter (C)		Outside Diameter (D)		Height (H)		Chamfer or Radius (r)	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
TP-L-10	10	0.3937	11	0.4331	26	1.0236	12	0.4724	1	0.040
TP-L-12	12	0.4724	13	0.5118	28	1.1024	12	0.4724	1	0.040
TP-L-15	15	0.5906	16	0.6299	31	1.2205	12	0.4724	1	0.040
TP-L-17	17	0.6693	18	0.7087	35	1.3780	12	0.4724	1	0.040
TP-L-20	20	0.7874	21	0.8268	37	1.4567	12	0.4724	1	0.040
TP-L-25	25	0.9843	26	1.0236	45	1.7717	14	0.5512	1	0.040
TP-L-30	30	1.1811	31	1.2205	50	1.9685	14	0.5512	1	0.040
TP-L-35	35	1.3780	36	1.4173	55	2.1654	16	0.6299	1	0.040
TP-L-40	40	1.5748	41	1.6142	60	2.3622	16	0.6299	2	0.080
TP-L-45	45	1.7717	46	1.8110	68	2.6672	16	0.6299	2	0.080
TP-L-50	50	1.9685	51	2.0079	74	2.9134	18	0.7087	2	0.080
TP-L-55	55	2.1654	56	2.2047	78	3.0709	18	0.7087	2	0.080
TP-L-60	60	2.3622	61	2.4016	82	3.2284	18	0.7087	2	0.080
TP-L-65	65	2.5591	66	2.5984	90	3.5433	20	0.7874	2	0.080
TP-L-70	70	2.7559	71	2.7953	95	3.7402	20	0.7874	2	0.080
TP-L-75	75	2.9528	76	2.9921	100	3.9370	20	0.7874	2	0.080
TP-L-80	80	3.1496	81	3.1890	110	4.3307	22	0.8661	3	0.120
TP-L-85	85	3.3465	86	3.3858	115	4.5276	22	0.8661	3	0.120
TP-L-90	90	3.5433	91	3.5827	120	4.7244	22	0.8661	3	0.120
TP-L-95	95	3.7402	96	3.7795	130	5.1181	25	0.9843	3	0.120
TP-L-100	100	3.9370	101	3.9764	135	5.3150	25	0.9843	3	0.120
TP-L-105	105	4.1339	106	4.1732	140	5.5118	25	0.9843	3	0.120
TP-L-110	110	4.3307	111	4.3701	145	5.7087	25	0.9843	3	0.120

SINGLE-DIRECTION, FLAT-FACE TYPE
MEDIUM SERIES, METRIC SIZES

Bearing Number	Inside Diameter (B)		Diameter (C)		Outside Diameter (D)		Height (H)		Chamfer or Radius (r)	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
TP-M-10	10	0.3937	11	0.4331	30	1.1811	12	0.4724	1	0.040
TP-M-12	12	0.4724	13	0.5118	32	1.2598	12	0.4724	1	0.040
TP-M-15	15	0.5906	16	0.6299	35	1.3780	14	0.5512	1	0.040
TP-M-17	17	0.6693	18	0.7087	38	1.4961	14	0.5512	1	0.040
TP-M-20	20	0.7874	21	0.8268	40	1.5748	14	0.5512	1	0.040
TP-M-25	25	0.9843	26	1.0236	48	1.8898	15	0.5906	1	0.040
TP-M-30	30	1.1811	31	1.2205	53	2.0866	15	0.5906	2	0.080
TP-M-35	35	1.3780	36	1.4173	62	2.4409	18	0.7087	2	0.080
TP-M-40	40	1.5748	41	1.6142	64	2.5197	18	0.7087	2	0.080
TP-M-45	45	1.7717	46	1.8110	73	2.8740	22	0.8661	2	0.080
TP-M-50	50	1.9685	51	2.0079	78	3.0709	22	0.8661	2	0.080
TP-M-55	55	2.1654	56	2.2047	88	3.4646	24	0.9449	2	0.080
TP-M-60	60	2.3622	61	2.4016	90	3.5433	24	0.9449	2	0.080
TP-M-65	65	2.5591	66	2.5984	100	3.9370	27	1.0630	3	0.120
TP-M-70	70	2.7559	71	2.7953	103	4.0551	27	1.0630	3	0.120
TP-M-75	75	2.9528	76	2.9921	110	4.3307	27	1.0630	3	0.120
TP-M-80	80	3.1496	81	3.1890	115	4.5276	31	1.2205	3	0.120
TP-M-85	85	3.3465	86	3.3858	125	4.9213	34	1.3386	3	0.120
TP-M-90	90	3.5433	91	3.5827	135	5.3150	36	1.4173	3	0.120
TP-M-95	95	3.7402	96	3.7795	140	5.5118	38	1.4961	3	0.120
TP-M-100	100	3.9370	101	3.9764	150	5.9055	38	1.4961	3	0.120
TP-M-105	105	4.1339	106	4.1732	155	6.1024	40	1.5748	3	0.120
TP-M-110	110	4.3307	111	4.3701	160	6.2992	40	1.5748	3	0.120

SINGLE-DIRECTION, FLAT-FACE TYPE
HEAVY SERIES, METRIC SIZES

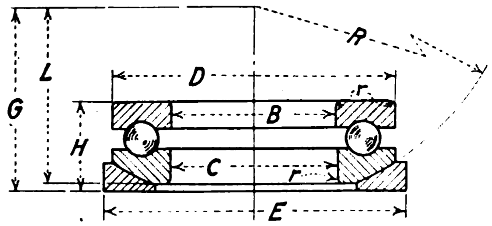
Bearing Number	Inside Diameter (B)		Diameter (C)		Outside Diameter (D)		Height (H)		Chamfer or Radius (r)	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
TP-H-25	25	0.9843	26	1.0236	52	2.0472	16	0.6299	2	0.080
TP-H-30	30	1.1811	31	1.2205	60	2.3622	19	0.7480	2	0.080
TP-H-35	35	1.3780	36	1.4173	68	2.6672	22	0.8661	2	0.080
TP-H-40	40	1.5748	41	1.6142	76	2.9921	25	0.9843	2	0.080
TP-H-45	45	1.7717	46	1.8110	85	3.3465	28	1.1024	2	0.080
TP-H-50	50	1.9685	51	2.0079	92	3.6221	31	1.2205	2	0.080
TP-H-55	55	2.1654	56	2.2047	100	3.9370	33	1.2992	3	0.120
TP-H-60	60	2.3622	61	2.4016	106	4.1732	35	1.3780	3	0.120
TP-H-65	65	2.5591	66	2.5984	112	4.4095	36	1.4173	3	0.120
TP-H-70	70	2.7559	71	2.7953	120	4.7244	38	1.4961	3	0.120
TP-H-75	75	2.9528	76	2.9921	128	5.0394	41	1.6142	3	0.120
TP-H-80	80	3.1496	81	3.1890	136	5.3543	44	1.7323	3	0.120
TP-H-85	85	3.3465	86	3.3858	145	5.7087	47	1.8504	3	0.120
TP-H-90	90	3.5433	91	3.5827	155	6.1024	50	1.9685	3	0.120
TP-H-95	95	3.7402	96	3.7795	165	6.4961	54	2.1260	3	0.120
TP-H-100	100	3.9370	101	3.9764	172	6.7717	57	2.2441	3	0.120
TP-H-105	105	4.1339	106	4.1732	180	7.0866	60	2.3622	3	0.120
TP-H-110	110	4.3307	111	4.3701	190	7.4803	63	2.4803	3	0.120



DOUBLE-DIRECTION, FLAT-FACE TYPE LIGHT SERIES, METRIC SIZES

Bearing Number	Inside Diameter (A)		Diameter (C)		Outside Diameter (D)		Height (H)		Thickness of Central Washer (S)	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
DTP-L-10	10	0.3937	16	0.6299	31	1.2205	22	0.8661	6.0	0.2362
DTP-L-12	12	0.4724	18	0.7087	35	1.3780	22	0.8661	6.0	0.2362
DTP-L-15	15	0.5906	21	0.8268	37	1.4567	22	0.8661	6.0	0.2362
DTP-L-20	20	0.7874	26	1.0236	45	1.7717	26	1.0236	7.0	0.2756
DTP-L-25	25	0.9843	31	1.2205	50	1.9685	26	1.0236	7.0	0.2756
DTP-L-30	30	1.1811	41	1.6142	60	2.3622	29	1.1417	7.5	0.2953
DTP-L-35	35	1.3780	46	1.8110	68	2.6672	29	1.1417	7.5	0.2953
DTP-L-40	40	1.5748	51	2.0079	74	2.9134	32	1.2598	8.0	0.3150
DTP-L-45	45	1.7717	56	2.2047	78	3.0709	32	1.2598	8.0	0.3150
DTP-L-50	50	1.9685	61	2.4016	82	3.2284	32	1.2598	8.0	0.3150
DTP-L-55	55	2.1654	66	2.5984	90	3.5433	36	1.4173	9.0	0.3543
DTP-L-60	60	2.3622	71	2.7953	95	3.7402	36	1.4173	9.0	0.3543
DTP-L-65	65	2.5591	76	2.9921	100	3.9370	36	1.4173	9.0	0.3543
DTP-L-70	70	2.7559	81	3.1890	110	4.3307	40	1.5748	10.5	0.4134
DTP-L-75	75	2.9528	86	3.3858	115	4.5276	40	1.5748	10.5	0.4134
DTP-L-80	80	3.1496	91	3.5827	120	4.7244	40	1.5748	10.5	0.4134
DTP-L-85	85	3.3465	96	3.7795	130	5.1181	45	1.7717	11.0	0.4331
DTP-L-90	90	3.5433	101	3.9764	135	5.3150	45	1.7717	11.0	0.4331
DTP-L-95	95	3.7402	106	4.1732	140	5.5118	45	1.7717	11.0	0.4331
DTP-L-100	100	3.9370	111	4.3701	145	5.7087	45	1.7717	11.0	0.4331

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.



DOUBLE-DIRECTION, FLAT-FACE TYPE MEDIUM SERIES, METRIC SIZES

Bearing Number	Inside Diameter (A)		Diameter (C)		Outside Diameter (D)		Height (H)		Thickness of Central Washer (S)	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
DTP-L-10	10	0.3937	16	0.6299	35	1.3780	26	1.0236	7.0	0.2756
DTP-L-15	15	0.5906	21	0.8268	40	1.5748	26	1.0236	7.0	0.2756
DTP-L-20	20	0.7874	26	1.0236	48	1.8898	28	1.1024	7.0	0.2756
DTP-L-25	25	0.9843	31	1.2205	53	2.0866	28	1.1024	7.0	0.2756
DTP-L-30	30	1.1811	41	1.6142	64	2.5197	32	1.2598	7.5	0.2953
DTP-L-35	35	1.3780	46	1.8110	73	2.8740	36	1.4173	8.0	0.3150
DTP-L-40	40	1.5748	51	2.0079	78	3.0709	36	1.4173	8.0	0.3150
DTP-L-45	45	1.7717	56	2.2047	88	3.4646	42	1.6535	9.0	0.3543
DTP-L-50	50	1.9685	61	2.4016	90	3.5433	42	1.6535	9.0	0.3543
DTP-L-55	55	2.1654	66	2.5984	100	3.9370	48	1.8898	11.0	0.4331
DTP-L-60	60	2.3622	71	2.7953	103	4.0551	48	1.8898	11.0	0.4331
DTP-L-65	65	2.5591	76	2.9921	110	4.3307	52	2.0472	15.0	0.5906
DTP-L-70	70	2.7559	81	3.1890	115	4.5276	56	2.2047	15.0	0.5906
DTP-L-75	75	2.9528	86	3.3858	125	4.9213	62	2.4409	15.0	0.5906
DTP-L-80	80	3.1496	91	3.5827	135	5.3150	64	2.5197	15.0	0.5906
DTP-L-85	85	3.3465	96	3.7795	140	5.5118	68	2.6672	16.0	0.6299
DTP-L-90	90	3.5433	101	3.9764	150	5.9055	68	2.6672	16.0	0.6299
DTP-L-95	95	3.7402	106	4.1732	155	6.1024	72	2.8346	17.0	0.6693
DTP-L-100	100	3.9370	111	4.3701	160	6.2992	72	2.8346	17.0	0.6693
DTP-L-105	105	4.1339	116	4.5669	165	6.4961	78	3.0709	18.0	0.7087
DTP-L-110	110	4.3307	126	4.9606	175	6.8898	82	3.2284	18.0	0.7087

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.

DOUBLE-DIRECTION, FLAT-FACE TYPE HEAVY SERIES, METRIC SIZES

Bearing Number	Inside Diameter (A)		Diameter (C)		Outside Diameter (D)		Height (H)		Thickness of Central Washer (S)	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
DTP-H-15	15	0.5906	26	1.0236	52	2.0472	29	1.1417	6	0.2362
DTP-H-20	20	0.7874	31	1.2205	60	2.3622	35	1.3780	8	0.3150
DTP-H-25	25	0.9843	36	1.4173	68	2.6672	40	1.5748	9	0.3543
DTP-H-30	30	1.1811	41	1.6142	76	2.9921	46	1.8110	10	0.3937
DTP-H-35	35	1.3780	46	1.8110	85	3.3465	52	2.0472	12	0.4724
DTP-H-40	40	1.5748	51	2.0079	92	3.6221	57	2.2441	13	0.5118
DTP-H-45	45	1.7717	61	2.4016	106	4.1732	65	2.5591	15	0.5906
DTP-H-50	50	1.9685	66	2.5984	112	4.4095	67	2.6378	16	0.6299
DTP-H-55	55	2.1654	71	2.7953	120	4.7244	71	2.7953	17	0.6693
DTP-H-60	60	2.3622	76	2.9921	128	5.0394	76	2.9921	18	0.7087
DTP-H-65	65	2.5591	81	3.1890	136	5.3543	81	3.1890	19	0.7480
DTP-H-70	70	2.7559	86	3.3858	145	5.7087	87	3.4252	20	0.7874
DTP-H-75	75	2.9528	91	3.5827	155	6.1024	92	3.6221	21	0.8268
DTP-H-80	80	3.1496	96	3.7795	165	6.4961	98	3.8583	22	0.8661
DTP-H-85	85	3.3465	106	4.1732	180	7.0866	109	4.2913	24	0.9449
DTP-H-90	90	3.5433	111	4.3701	190	7.4803	115	4.5276	25	0.9843
DTP-H-95	95	3.7402	116	4.5669	200	7.8740	120	4.7244	26	1.0236
DTP-H-100	100	3.9370	121	4.7638	210	8.2677	125	4.9213	27	1.0630
DTP-H-110	110	4.3307	131	5.1575	220	8.6614	131	5.1575	32	1.2598

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.

SINGLE-DIRECTION, SELF-ALIGNING TYPE LIGHT SERIES, METRIC SIZES

Bearing Number	Inside Diameter (B)		Diameter (C)		Outside Diameter (D)		Diameter (E)		Height (H)		Radius (R)		L		G	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
TS-L-10	10	0.3937	12	0.4724	26	1.0236	30	1.1811	13.5	0.5315	20	0.7874	17.86	0.7032	19.36	0.7622
TS-L-12	12	0.4724	14	0.5512	28	1.1024	32	1.2598	14.5	0.5709	20	0.7874	17.32	0.6819	19.82	0.7803
TS-L-15	15	0.5906	17	0.6693	31	1.2205	35	1.3780	14.5	0.5709	25	0.9843	22.20	0.8740	23.70	0.9331
TS-L-17	17	0.6693	19	0.7480	35	1.3780	37	1.4567	14.5	0.5709	25	0.9843	21.65	0.8524	23.15	0.9114
TS-L-20	20	0.7874	22	0.8661	37	1.4567	40	1.5748	15.0	0.5906	30	1.1811	26.53	1.0445	28.53	1.1232
TS-L-25	25	0.9843	27	1.0630	45	1.7717	48	1.8898	17.0	0.6693	35	1.3780	30.31	1.1933	32.31	1.2721
TS-L-30	30	1.1811	32	1.2598	50	1.9685	53	2.0866	17.0	0.6693	40	1.5748	34.64	1.3638	36.64	1.4425
TS-L-35	35	1.3780	37	1.4567	55	2.1654	60	2.3622	19.0	0.7480	45	1.7717	38.98	1.5346	40.98	1.6134
TS-L-40	40	1.5748	42	1.6535	60	2.3622	65	2.5591	19.0	0.7480	50	1.9685	43.30	1.7047	45.30	1.7835
TS-L-45	45	1.7717	47	1.8504	68	2.6672	72	2.8346	19.0	0.7480	55	2.1654	47.34	1.8638	49.34	1.9425
TS-L-50	50	1.9685	52	2.0472	74	2.9134	78	3.0709	21.0	0.8268	60	2.3622	51.37	2.0224	53.37	2.1012
TS-L-55	55	2.1654	57	2.2441	78	3.0709	82	3.2284	21.5	0.8465	65	2.5591	56.00	2.2047	58.50	2.3032
TS-L-60	60	2.3622	62	2.4409	82	3.2284	86	3.3858	21.5	0.8465	70	2.7559	60.33	2.3752	62.82	2.4732
TS-L-65	65	2.5591	67	2.6378	90	3.5433	95	3.7402	24.5	0.9646	75	2.9528	64.36	2.5339	66.86	2.6323
TS-L-70	70	2.7559	72	2.8346	95	3.7402	100	3.9370	24.5	0.9646	80	3.1496	68.69	2.7043	71.19	2.8028
TS-L-75	75	2.9528	77	3.0315	100	3.9370	105	4.1339	25.0	0.9843	85	3.3465	73.03	2.8752	76.03	2.9933
TS-L-80	80	3.1496	82	3.2281	110	4.3307	115	4.5276	26.0	1.0236	95	3.7402	82.27	3.2390	85.27	3.3571
TS-L-85	85	3.3465	87	3.4252	115	4.5276	120	4.7244	27.0	1.0630	100	3.9370	86.60	3.4095	89.60	3.5276
TS-L-90	90	3.5433	92	3.6221	120	4.7244	125	4.9213	27.0	1.0630	105	4.1339	90.93	3.5799	93.93	3.6980
TS-L-95	95	3.7402	97	3.8189	130	5.1181	135	5.3150	30.0	1.1811	110	4.3307	94.68	3.7276	97.68	3.8457
TS-L-100	100	3.9370	102	4.0157	135	5.3150	140	5.5118	30.0	1.1811	115	4.5276	99.01	3.8980	102.01	4.0161
TS-L-105	105	4.1339	107	4.2126	140	5.5118	145	5.7087	30.0	1.1811	120	4.7244	103.34	4.0685	106.34	4.1866
TS-L-110	110	4.3307	112	4.4095	145	5.7087	150	5.9055	30.0	1.1811	125	4.9213	107.67	4.2390	110.67	4.3571

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.

SINGLE-DIRECTION, SELF-ALIGNING TYPE MEDIUM SERIES, METRIC SIZES

Bearing Number	Inside Diameter (B)		Diameter (C)		Outside Diameter (D)		Diameter (E)		Height (H)		Radius (R)		L		G	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
TS-M-10	10	0.3937	12	0.4724	30	1.1811	35	1.3780	16	0.6299	20	0.7874	17.02	0.6701	19.02	0.7488
TS-M-12	12	0.4724	14	0.5512	32	1.2598	37	1.4567	16	0.6299	22	0.8661	19.05	0.7500	21.05	0.8287
TS-M-15	15	0.5906	17	0.6693	35	1.3780	40	1.5748	18	0.7087	25	0.9843	21.03	0.8280	23.93	0.9421
TS-M-17	17	0.6693	19	0.7480	38	1.4961	42	1.6535	18	0.7087	27	1.0630	23.93	0.9421	25.93	1.0209
TS-M-20	20	0.7874	22	0.8661	40	1.5748	45	1.7717	18	0.7087	30	1.1811	27.04	1.0646	29.04	1.1433
TS-M-25	25	0.9843	27	1.0630	48	1.8898	50	1.9685	19	0.7480	35	1.3780	30.87	1.2114	33.87	1.3335
TS-M-30	30	1.1811	32	1.2598	53	2.0866	60	2.3622	21	0.8268	40	1.5748	35.20	1.3858	38.20	1.5039
TS-M-35	35	1.3780	37	1.4567	62	2.4409	70	2.7559	24	0.9449	50	1.9685	44.90	1.7677	47.90	1.8858
TS-M-40	40	1.5748	42	1.6535	64	2.5197	72	2.8346	24	0.9449	50	1.9685	43.59	1.7161	46.59	1.8343
TS-M-45	45	1.7717	47	1.8504	73	2.8740	80	3.1496	28	1.1024	60	2.3622	53.32	2.0992	56.32	2.2173
TS-M-50	50	1.9685	52	2.0472	78	3.0709	85	3.3465	28	1.1024	65	2.5591	57.66	2.2701	60.66	2.3882
TS-M-55	55	2.1654	57	2.2441	88	3.4646	95	3.7402	32	1.2598	70	2.7559	62.00	2.4339	66.00	2.5984
TS-M-60	60	2.3622	62	2.4409	90	3.5433	100	3.9370	32	1.2598	75	2.9528	66.33	2.6114	70.33	2.7689
TS-M-65	65	2.5591	67	2.6378	100	3.9370	110	4.3307	36	1.4173	80	3.1496	70.67	2.7823	74.67	2.9398
TS-M-70	70	2.7559	72	2.8316	103	4.0551	115	4.5276	36	1.4173	85	3.3465	74.67	2.9398	78.67	3.0972
TS-M-75	75	2.9528	77	3.0315	110	4.3307	120	4.7244	36	1.4173	90	3.5433	79.33	3.1232	83.33	3.2807
TS-M-80	80	3.1496	82	3.2284	115	4.5276	125	4.9213	39	1.5354	95	3.7402	83.67	3.2941	87.67	3.4516
TS-M-85	85	3.3465	87	3.4252	125	4.9213	135	5.3150	42	1.6535	105	4.1339	93.64	3.6866	97.64	3.8441
TS-M-90	90	3.5433	92	3.6221	135	5.3150	140	5.5118	43	1.6929	110	4.3307	97.20	3.8268	101.20	3.9843
TS-M-95	95	3.7402	97	3.8189	140	5.5118	150	5.9055	45	1.7717	115	4.5276	101.27	3.9870	105.27	4.1445
TS-M-100	100	3.9370	102	4.0157	150	5.9055	160	6.2992	46	1.8110	120	4.7244	104.49	4.1138	108.49	4.2713
TS-M-105	105	4.1339	107	4.2126	155	6.1024	165	6.4961	48	1.8898	130	5.1181	112.58	4.4323	117.58	4.6291
TS-M-110	110	4.3307	112	4.4095	160	6.2992	170	6.6929	48	1.8898	135	5.3150	116.91	4.6028	121.91	4.7996

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.

SINGLE-DIRECTION, SELF-ALIGNING TYPE HEAVY SERIES, METRIC SIZES

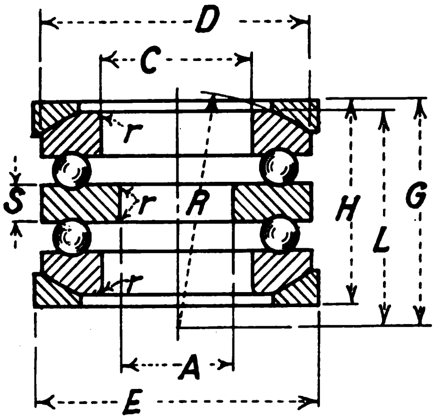
Bearing Number	Inside Diameter (B)		Diameter (C)		Outside Diameter (D)		Diameter (E)		Height (H)		Radius (R)		L		G	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
TS-H-25	25	0.9843	27	1.0630	52	2.0472	55	2.1654	22	0.8661	40	1.5748	34.64	1.3638	37.64	1.4819
TS-H-30	30	1.1811	32	1.2598	60	2.3622	65	2.5591	24	0.9449	45	1.7717	38.97	1.5343	41.97	1.6524
TS-H-35	35	1.3780	37	1.4567	68	2.6672	75	2.9528	27	1.0630	55	2.1654	47.63	1.8752	50.63	1.9933
TS-H-40	40	1.5748	42	1.6535	76	2.9921	85	3.3465	30	1.1811	60	2.3622	51.96	2.0457	54.96	2.1638
TS-H-45	45	1.7717	47	1.8504	85	3.3465	95	3.7402	33	1.2992	65	2.5591	56.29	2.2161	59.29	2.3343
TS-H-50	50	1.9685	52	2.0472	92	3.6221	100	3.9370	36	1.4173	75	2.9528	64.95	2.5571	67.95	2.6752
TS-H-55	55	2.1654	57	2.2441	100	3.9370	110	4.3307	39	1.5354	80	3.1496	69.28	2.7276	73.28	2.8850
TS-H-60	60	2.3622	62	2.4409	106	4.1732	115	4.5276	41	1.6142	85	3.3465	73.61	2.8980	77.61	3.0555
TS-H-65	65	2.5591	67	2.6378	112	4.4095	120	4.7244	42	1.6535	90	3.5433	77.94	3.0685	81.94	3.2260
TS-H-70	70	2.7559	72	2.8346	120	4.7244	130	5.1181	44	1.7323	95	3.7402	82.27	3.2390	86.27	3.3965
TS-H-75	75	2.9528	77	3.0315	128	5.0394	140	5.5118	47	1.8501	105	4.1339	90.93	3.5799	94.93	3.7374
TS-H-80	80	3.1496	82	3.2284	136	5.3543	145	5.7087	50	1.9685	110	4.3307	95.26	3.7504	99.26	3.9879
TS-H-85	85	3.3465	87	3.4252	145	5.7087	155	6.1024	54	2.1260	120	4.7244	103.92	4.0913	108.92	4.2882
TS-H-90	90	3.5433	92	3.6221	155	6.1024	165	6.4961	57	2.2441	125	4.9213	108.25	4.2618	113.25	4.4587
TS-H-95	95	3.7402	97	3.8189	165	6.4961	175	6.8898	61	2.4016	130	5.1181	112.68	4.4362	117.68	4.6331
TS-H-100	100	3.9370	102	4.0157	172	6.7717	185	7.2835	64	2.5197	140	5.5118	121.24	4.7732	126.24	4.9701
TS-H-105	105	4.1339	107	4.2126	180	7.0866	195	7.6772	67	2.6378	145	5.7087	125.57	4.9437	130.57	5.1406
TS-H-110	110	4.3307	112	4.4095	190	7.4803	205	8.0709	70	2.7559	155	6.1024	134.23	5.2847	139.23	5.4815

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.

DOUBLE-DIRECTION, SELF-ALIGNING TYPE LIGHT SERIES, METRIC SIZES

Bearing Number	Inside Diameter (A)		Diameter (C)		Outside Diameter (D)		Diameter (E)		Height (H)		Thickness of Central Washer (S)		Radius (R)		L		G	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
DTS-L-10	10	0.3937	17	0.6693	31	1.2205	35	1.3780	27	1.0630	6.0	0.2362	25	0.9843	22.20	0.8740	23.70	0.9331
DTS-L-12	12	0.4724	19	0.7480	35	1.3780	37	1.4567	27	1.0630	6.0	0.2362	25	0.9843	21.65	0.8524	23.15	0.9114
DTS-L-15	15	0.5906	22	0.8661	37	1.4567	40	1.5748	28	1.1024	6.0	0.2362	30	1.1811	26.53	1.0445	28.53	1.1232
DTS-L-20	20	0.7874	27	1.0630	45	1.7717	48	1.8898	32	1.2598	7.0	0.2756	35	1.3780	30.31	1.1933	32.31	1.2721
DTS-L-25	25	0.9843	32	1.2598	50	1.9685	53	2.0866	32	1.2598	7.0	0.2756	40	1.5748	34.64	1.3638	36.64	1.4425
DTS-L-30	30	1.1811	42	1.6535	60	2.3622	65	2.5591	35	1.3780	7.5	0.2953	50	1.9685	43.30	1.7047	45.30	1.7835
DTS-L-35	35	1.3780	47	1.8504	68	2.6672	72	2.8346	35	1.3780	7.5	0.2953	55	2.1654	47.34	1.8638	49.34	1.9425
DTS-L-40	40	1.5748	52	2.0472	74	2.9134	78	3.0709	38	1.4961	8.0	0.3150	60	2.3622	51.37	2.0224	53.37	2.1012
DTS-L-45	45	1.7717	57	2.2441	78	3.0709	82	3.2284	39	1.5354	8.0	0.3150	65	2.5591	56.00	2.2047	58.50	2.3032
DTS-L-50	50	1.9685	62	2.4409	82	3.2284	86	3.3858	39	1.5354	8.0	0.3150	70	2.7559	60.33	2.3752	62.82	2.4732
DTS-L-55	55	2.1654	67	2.6378	90	3.5433	95	3.7402	45	1.7717	9.0	0.3543	75	2.9528	64.36	2.5339	66.86	2.6323
DTS-L-60	60	2.3622	72	2.8346	95	3.7402	100	3.9370	45	1.7717	9.0	0.3543	80	3.1496	68.69	2.7043	71.19	2.8028
DTS-L-65	65	2.5591	77	3.0315	100	3.9370	105	4.1339	46	1.8110	9.0	0.3543	85	3.3465	73.03	2.8752	76.03	2.9933
DTS-L-70	70	2.7559	82	3.2284	110	4.3307	115	4.5276	50	1.9685	10.5	0.4134	95	3.7402	82.27	3.2390	85.27	3.3571
DTS-L-75	75	2.9528	87	3.4252	115	4.5276	120	4.7244	50	1.9685	10.5	0.4134	100	3.9370	86.60	3.4095	89.60	3.5276
DTS-L-80	80	3.1496	92	3.6221	120	4.7244	125	4.9213	50	1.9685	10.5	0.4134	105	4.1339	90.93	3.5799	93.93	3.6980
DTS-L-85	85	3.3465	97	3.8189	130	5.1181	135	5.3150	55	2.1654	11.0	0.4331	110	4.3307	94.68	3.7276	97.68	3.8457
DTS-L-90	90	3.5433	102	4.0157	135	5.3150	140	5.5118	55	2.1654	11.0	0.4331	115	4.5276	99.01	3.8980	102.01	4.0161
DTS-L-95	95	3.7402	107	4.2126	140	5.5118	145	5.7087	55	2.1654	11.0	0.4331	120	4.7244	103.34	4.0685	106.34	4.1866
DTS-L-100	100	3.9370	112	4.4095	145	5.7087	150	5.9055	55	2.1654	11.0	0.4331	125	4.9213	107.67	4.2390	110.67	4.3571

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.



DOUBLE-DIRECTION, SELF-ALIGNING TYPE MEDIUM SERIES, METRIC SIZES

Bearing Number	Inside Diameter (A)		Diameter (C)		Outside Diameter (D)		Diameter (E)		Height (H)		Thickness of Central Washer (S)		Radius (R)		L		G	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
DTS-M-10	10	0.3937	17	0.6693	35	1.3780	40	1.5748	34	1.3386	7.0	0.2756	25	0.9843	21.93	0.8634	23.93	0.9421
DTS-M-15	15	0.5906	22	0.8661	40	1.5748	45	1.7717	34	1.3386	7.0	0.2756	30	1.1811	27.04	1.0646	29.04	1.1433
DTS-M-20	20	0.7874	27	1.0630	48	1.8898	50	1.9685	36	1.4173	7.0	0.2756	35	1.3780	30.87	1.2154	33.87	1.3335
DTS-M-25	25	0.9843	32	1.2598	53	2.0866	60	2.3622	39	1.5354	7.0	0.2756	40	1.5748	35.20	1.3858	38.20	1.5039
DTS-M-30	30	1.1811	42	1.6535	64	2.5197	72	2.8346	44	1.7323	7.5	0.2953	50	1.9685	43.59	1.7161	46.59	1.8343
DTS-M-35	35	1.3780	47	1.8504	73	2.8740	80	3.1496	52	2.0472	8.0	0.3150	60	2.3622	53.32	2.0992	56.32	2.2173
DTS-M-40	40	1.5748	52	2.0472	78	3.0709	85	3.3465	52	2.0472	8.0	0.3150	65	2.5591	57.66	2.2701	60.66	2.3882
DTS-M-45	45	1.7717	57	2.2441	88	3.4646	95	3.7402	58	2.2835	9.0	0.3543	70	2.7559	62.00	2.4409	66.00	2.5984
DTS-M-50	50	1.9685	62	2.4409	90	3.5433	100	3.9370	58	2.2835	9.0	0.3543	75	2.9528	66.33	2.6114	70.33	2.7689
DTS-M-55	55	2.1654	67	2.6378	100	3.9370	110	4.3307	66	2.5984	11.0	0.4331	80	3.1496	70.67	2.7823	74.67	2.9398
DTS-M-60	60	2.3622	72	2.8346	103	4.0551	115	4.5276	66	2.5984	11.0	0.4331	85	3.3465	74.67	2.9398	78.67	3.0972
DTS-M-65	65	2.5591	77	3.0315	110	4.3307	120	4.7244	70	2.7559	15.0	0.5906	90	3.5433	79.33	3.1232	83.33	3.2807
DTS-M-70	70	2.7559	82	3.2284	115	4.5276	125	4.9213	72	2.8346	15.0	0.5906	95	3.7402	83.67	3.2941	87.67	3.4516
DTS-M-75	75	2.9528	87	3.4252	125	4.9213	135	5.3150	78	3.0709	15.0	0.5906	105	4.1339	93.64	3.6866	97.64	3.8441
DTS-M-80	80	3.1496	92	3.6221	135	5.3150	140	5.5118	78	3.0709	15.0	0.5906	110	4.3307	97.20	3.8268	101.20	3.9843
DTS-M-85	85	3.3465	97	3.8189	140	5.5118	150	5.9055	82	3.2284	16.0	0.6299	115	4.5276	101.27	3.9870	105.27	4.1445
DTS-M-90	90	3.5433	102	4.0157	150	5.9055	160	6.2992	84	3.3071	16.0	0.6299	120	4.7244	104.49	4.1138	108.49	4.2713
DTS-M-95	95	3.7402	107	4.2126	155	6.1024	165	6.4961	88	3.4646	17.0	0.6693	130	5.1181	112.58	4.4323	117.58	4.6291
DTS-M-100	100	3.9370	112	4.4095	160	6.2992	170	6.6929	88	3.4646	17.0	0.6693	135	5.3150	116.91	4.6028	121.91	4.7996
DTS-M-105	105	4.1339	117	4.6063	165	6.4961	175	6.8898	92	3.6221	18.0	0.7087	140	5.5118	121.65	4.7894	126.65	4.9862
DTS-M-110	110	4.3307	122	4.8032	175	6.8898	190	7.4803	100	3.9370	18.0	0.7087	150	5.9055	130.00	5.1181	136.00	5.3543

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.

DOUBLE-DIRECTION, SELF-ALIGNING TYPE HEAVY SERIES, METRIC SIZES

Bearing Number	Inside Diameter (A)		Diameter (C)		Outside Diameter (D)		Diameter (E)		Height (H)		Thickness of Central Washer (S)		Radius (R)		L		G	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
DTS-H-15	15	0.5906	27	1.0630	52	2.0472	55	2.1654	41	1.6142	6	0.2362	40	1.5748	34.64	1.3638	37.64	1.4819
DTS-H-20	20	0.7874	32	1.2598	60	2.3622	65	2.5591	45	1.7717	8	0.3150	45	1.7717	38.97	1.5343	41.97	1.6524
DTS-H-25	25	0.9843	37	1.4567	68	2.6672	75	2.9528	50	1.9685	9	0.3543	55	2.1654	47.63	1.8752	50.63	1.9933
DTS-H-30	30	1.1811	42	1.6535	76	2.9921	85	3.3465	56	2.2047	10	0.3937	60	2.3622	51.96	2.0457	54.96	2.1638
DTS-H-35	35	1.3780	47	1.8504	85	3.3465	95	3.7402	62	2.4409	12	0.4724	65	2.5591	56.29	2.2161	59.29	2.3343
DTS-H-40	40	1.5748	52	2.0472	92	3.6221	100	3.9370	67	2.6378	13	0.5118	75	2.9528	64.95	2.5567	67.95	2.6748
DTS-H-45	45	1.7717	62	2.4409	106	4.1732	115	4.5276	77	3.0315	15	0.5906	85	3.3465	73.61	2.8980	77.61	3.0551
DTS-H-50	50	1.9685	67	2.6378	112	4.4095	120	4.7244	79	3.1102	16	0.6299	90	3.5433	77.94	3.0685	81.94	3.2260
DTS-H-55	55	2.1654	72	2.8346	120	4.7244	130	5.1181	83	3.2677	17	0.6693	95	3.7402	82.27	3.2390	86.27	3.3965
DTS-H-60	60	2.3622	77	3.0315	128	5.0394	140	5.5118	88	3.4646	18	0.7087	105	4.1339	90.93	3.5799	94.93	3.7374
DTS-H-65	65	2.5591	82	3.2281	136	5.3543	145	5.7087	93	3.6614	19	0.7480	110	4.3307	95.26	3.7504	99.26	3.9079
DTS-H-70	70	2.7559	87	3.4252	145	5.7087	155	6.1024	101	3.9764	20	0.7874	120	4.7244	103.92	4.0913	108.92	4.2882
DTS-H-75	75	2.9528	92	3.6221	155	6.1024	165	6.4961	106	4.1732	21	0.8268	125	4.9213	108.25	4.2618	113.25	4.4587
DTS-H-80	80	3.1496	97	3.8189	165	6.4961	175	6.8898	112	4.4095	22	0.8661	130	5.1181	112.68	4.4362	117.68	4.6331
DTS-H-85	85	3.3465	107	4.2126	180	7.0866	195	7.6772	123	4.8425	24	0.9449	145	5.7087	125.57	4.9437	130.57	4.1406
DTS-H-90	90	3.5433	112	4.4095	190	7.4803	205	8.0709	129	5.0787	25	0.9843	155	6.1024	134.23	5.2847	139.23	4.4815
DTS-H-95	95	3.7402	117	4.6063	200	7.8740	215	8.4646	136	5.3543	26	1.0236	160	6.2992	138.56	5.4551	142.56	5.5732
DTS-H-100	100	3.9370	122	4.8032	210	8.2677	225	8.8583	143	5.6299	27	1.0630	170	6.6929	147.23	5.7965	153.23	6.0327
DTS-H-110	110	4.3307	132	5.1969	220	8.6614	235	9.2520	148	5.8268	32	1.2598	175	6.8898	151.55	5.9665	155.55	6.1240

The same chamfers or corner radii specified for the single-direction, flat-face type thrust ball bearings apply to this series.

the practice of at least six domestic manufacturers using this type of bearing and that it is necessary to consider European practice as the importance of international standardization cannot be overlooked.

MINORITY REPORT

The boundary dimensions, principally the inside and outside diameters, of some of the proposed metric-type thrust ball-bearings standards are such that if the proper ball-diameters are used, or those consistent with the size of the bearing, there is not a sufficient difference between the diameter of the ball and the width of the face of the bearing to permit the use of a ball retainer of sufficient strength to give satisfactory service.

It was suggested at the Subdivision meeting in New York City on Nov. 9 that smaller ball sizes might be used in certain bearing sizes to permit the use of a satisfactory retainer. While this is possible, the recommendations should be along the lines of good engineering and not recommend substitutes.

It was stated at the Subdivision meeting that the present demand for metric thrust ball-bearings represents only a very small percentage of the total demand for thrust bearings. Any action taken toward the standardization of an unknown quantity should certainly be considered in the light of past experience. It was also stated that certain of the proposed metric sizes have been made with retainers having either the inside or outside section, that part extending beyond the ball, cut away entirely so as to keep the bearing within the boundary dimensions referred to. It is not considered that this represents good ball-bearing practice.

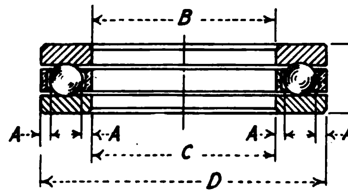
The demand for metric sizes apparently dates back to certain requirements during the war period that could not be taken care of by the importation of metric-type thrust ball-bearings. Keeping in mind that the Society of Automotive Engineers is representative of the American automotive industries whose output represents about 87 per cent of the world's production of automobiles, it is improbable that the demand arose in this Country. In view of these figures it is reasonable to assume that the European manufacturers would adopt our standards if they were properly presented.

The adoption of the report proposed at the Subdivision meeting held on Nov. 9 means that we are surrendering our American standard practice to that of European manufacturers.

We, the undersigned, therefore urge that this matter be given serious consideration and recommend that it be referred back to the Ball and Roller Bearings Division with instructions to submit a recommendation on the same general line of standardization of inch-type thrust-bearings, or redesign the present proposed metric sizes along the lines of the best engineering practice including the ball diameters.

The accompanying data are submitted in support of the above statements.

(Signed) Frank Beemer
S. A. Strickland
F. A. Collins, Jr.



CAGE DIMENSIONS ALLOWED BY PROPOSED S. A. E. STANDARD FOR SINGLE-DIRECTION, FLAT-FACE TYPE THRUST BALL-BEARINGS

Inside Diameter (*B*)		Diameter (*C*)		Outside Diameter (*D*)		Height		Ball Diameter	*A**
Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	In.	In.
25	0.9843	26	1.0236	45	1.7717	14	0.5512	9/32	0.0460
30	1.1811	31	1.2205	50	1.9686	14	0.5512	9/32	0.0460
35	1.3780	36	1.4173	55	2.1654	16	0.6299	5/16	0.0310
40	1.5748	41	1.6142	60	2.3622	16	0.6299	5/16	0.0310
60	2.3622	61	2.4016	82	3.2284	18	0.7087	11/32	0.0340
65	2.5591	66	2.5984	90	3.5433	20	0.7874	3/8	0.0480
70	2.7559	71	2.7953	95	3.7402	20	0.7874	3/8	0.0480

*Dimension *A* permits a clearance of 1 mm. on the inside diameter of the retainer when mounted on a shaft of diameter *B*. Dimension *A* does not make an allowance for the necessary clearance about the balls and a clearance between the outside diameters of retainer and bearing.

ELECTRIC VEHICLE DIVISION REPORT

Division Personnel

E. L. Clark, *Chairman*	Commercial Truck Co.
Karl Probst, *Vice-Chairman*	Milburn Wagon Co.
G. L. Bixby	Detroit Electric Car Co.
J. G. Carroll	Walker Vehicle Co.
H. M. Pierce	Ward Motor Vehicle Co.
F. E. Queeney	Lansden Co., Inc.
C. R. Skinner, Jr.	New York Edison Co.

ELECTRIC VEHICLE MOTORS

(*Proposed S.A.E. Recommended Practice*)

At the last meeting of the Electric Vehicle Division it was thought that a definite standard for the rating of vehicle motors would be of great advantage in comparing tests of motors of various makes and the adoption of the method of rating motors approved by the American Institute of Electrical Engineers was favored, provided a 4-hr. time limit and a normal load were specified when rating the motor. Therefore

The Electric Vehicle Division recommends the adoption as S.A.E. Recommended Practice for the following rating for electric vehicle motors.

The rating of electric automobile propulsion motors shall be based on a temperature rise not to exceed 65 deg. cent. (149 deg. fahr.) by thermometer, or 75 deg. cent. (167 deg. fahr.) by resistance after 4 hr. of continuous operation at normal rated load

The tests shall be made on a stand with a constant room-temperature of 20 deg. cent. (68 deg. fahr.) and with the motor covers arranged as in service.

With the exception of the time limit and rated load requirements, this rating conforms to Section 5205 of the 1921 A. I. E. E. Standardization Rules.

ELECTRICAL EQUIPMENT DIVISION REPORT

Division Personnel

F. W. Andrew, *Chairman*	Eisemann Magneto Corporation
T. L. Lee, *Vice-Chairman*	North East Electric Co.
Azel Ames	Kerite Insulated Wire & Cable Co.
G. S. Cawthorne	Master Trucks, Inc.
W. A. Chryst	Dayton Engineering Laboratories Co.
S. F. Evelyn	Continental Motors Corporation
C. F. Gilchrist	Electric Auto-Lite Corporation
W. S. Haggott	Packard Electric Co.
C. H. Kindl	Westinghouse Electric & Mfg. Co.
F. C. Kroeger	Remy Electric Co.
B. M. Leece	Leece-Neville Co.
A. D. T. Libby	Splitdorf Electrical Co.
Charles Marcus	Bijur Motor Appliance Co.
Ernest Wooler	Cleveland Automobile Co.

In 1920 the Electrical Equipment Division appointed a Subdivision to review the present S.A.E. Standards for generator and starting-motor mountings and to bring them into accord with the best engineering practice. A general letter was sent out to generator and starting-motor users requesting suggestions as to possible re-

visions of the present S.A.E. Standards, the suggestions received being referred to the Subdivision of which T. L. Lee, of the North East Electric Co., was chairman, the other members being C. H. Kindl, of the Westinghouse Electric & Mfg. Co., Ernest Wooler, of the Cleveland Automobile Co., B. M. Leece, of the Leece-Neville Co., and S. F. Evelyn, of the Continental Motors Corporation.

Several meetings of the Subdivision were held at which the present standards were considered and tentative revisions drafted. A joint meeting with the Automotive Electric Association was also held at White Sulphur Springs in June, largely for the purpose of securing more effective results through closer cooperation of the Society and that organization with the executives of the consuming interests.

The recommendations of the Subdivision were submitted to generator and starting-motor manufacturers and users and, as they met with general approval, they were approved by the Electrical Equipment Division.

It is therefore felt that the recommendations covering the present S.A.E. Standards will meet the objections which have been raised and have prevented the more extensive use of these important standards.

GENERATOR FLANGE MOUNTINGS

(Proposed Revision of S.A.E. Standard)

As considerable trouble has been experienced in using the present S.A.E. Standard Generator Flange Mountings, p. B17 of the S.A.E. HANDBOOK, due to the cotter-pin hole being located too close to the end of the shaft and as it is considered desirable to use S.A.E. Standard lock-washers and nuts for holding the gear on the shaft-end, revisions of dimensions *L* and *M* were proposed.

The Subdivision has also been advised that in some generator installations a gasket is frequently used under the flange, which is not completely dimensioned in the present standard. The Subdivision has therefore proposed additional contour dimensions and a maximum flange-thickness of 1/2 in. to complete the standard. It was also thought advisable by the Division that the detailed dimensions of the shaft-end be specified. Therefore

The Electrical Equipment Division recommends that the present S.A.E. Standard for Generator Flange Mountings, p. B17 of the S.A.E. HANDBOOK, be revised by

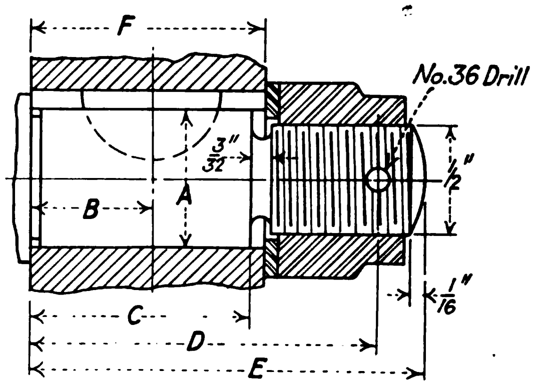
Increasing dimension *D* by 1/32 in. and dimension *E* by 3/32 in.

The addition of the 7/8-in. and 1 1/2-in. contour dimensions.

The addition of a flange thickness of 1/2 in.

The addition of the detailed shaft-end dimensions.

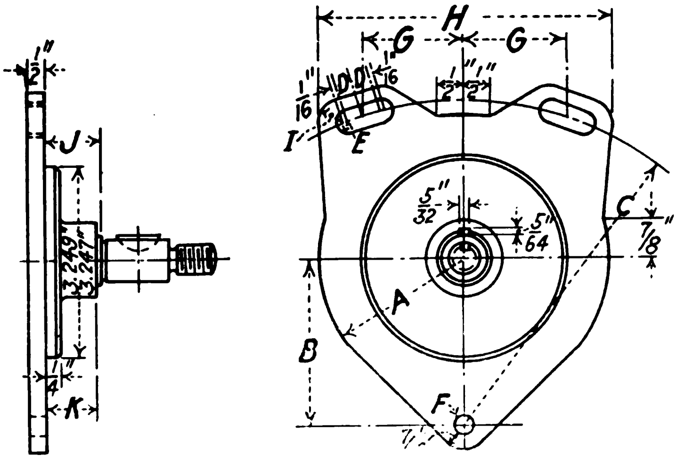
The present S.A.E. Standard revised as proposed by the Electrical Equipment Division is given in the accompanying tables.



FLANGE-TYPE GENERATOR SHAFT-END DIMENSIONS

Size No.	Diameter A		Woodruff Key		C	D	E	F
	Max.	Min.	Key No.	B				
1	0.6250	0.6245	6	5/16	1	1 5/16	1 13/16	1 5/16
2	0.7500	0.7495	8	13/16	1 1/4	1 13/16	2 5/16	1 5/16

The pinion shall be held on by a 11/16 x 1/16-in. S.A.E. Special Light lock-washer and a 1/2 in.-20 S.A.E. Standard castellated nut.



FLANGE-TYPE GENERATOR MOUNTING DIMENSIONS

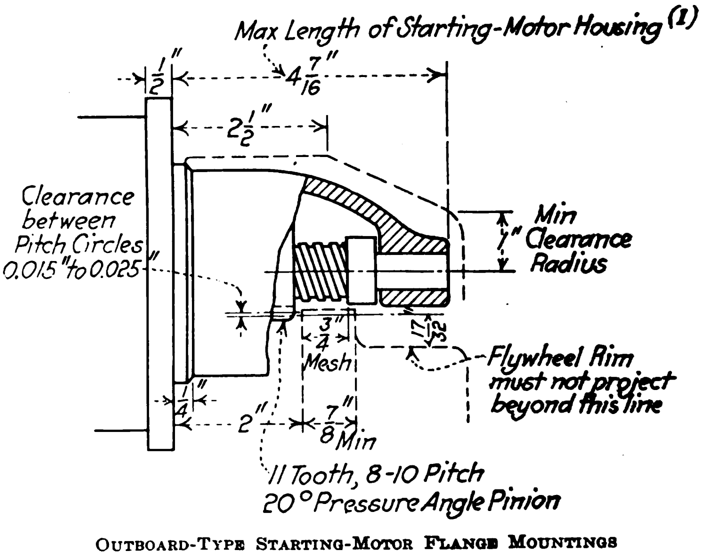
Size No.	A	B	C	D	E	F	G	H Max.	I	J ± 1/64	K Max.
1	2 3/8	2 13/16	5 3/8	1/4	7/32	13/32	1 5/8	4 3/4	7/16	13/16	3/4
2	2 5/8	3 1/16	5 7/8	5/16	1/4	13/32	1 13/16	5 3/8	1/2	1 1/16	1

STARTING-MOTOR FLANGE MOUNTINGS

OUTBOARD TYPE

(Proposed S.A.E. Recommended Practice)

As the Subdivision on Generator and Starting-Motor Mountings considered that the present S.A.E. Standard for Starting-Motor Flange Mountings, p. B19 of the S.A.E. HANDBOOK, should be amplified by a type of



OUTBOARD-TYPE STARTING-MOTOR FLANGE MOUNTINGS

In special cases where 13-tooth large-shift pinions are required, the maximum length of the starting-motor housing shall be 4 3/4 instead of 4 7/16 in. and the flywheel mesh 1 instead of 3/4 in.

The flange dimensions shall be the same as those specified for the present S.A.E. Standard for Inboard-Type Starting-Motor Flange Mountings, p. B19 of the S.A.E. HANDBOOK.

mounting for outboard installations only, a recommendation for this type of flange mounting similar to the present recommended practice for starting-motor barrel mountings of the outboard type was submitted to and favorably considered by the Division. Therefore

The Electrical Equipment Division recommends for adoption as S.A.E. Recommended Practice the outboard-type starting-motor flange dimensions given in the accompanying illustration [see p. 536].

INBOARD TYPE

(*Proposed Revision of S.A.E. Standard*)

As it is necessary in some installations to attach the starting motor to the flywheel housing by the use of studs instead of bolts because of insufficient clearance, it was suggested that the Electrical Equipment Division should specify the thickness of the starting-motor flange in order that the length of the studs might be readily determined. This matter was favorably considered by the Subdivision on Generator and Starting-Motor Mountings, and subsequently by the Division. Therefore

The Electrical Equipment Division recommends that the present S.A.E. Standard for Starting-Motor Flange Mountings, p. B19 of the S.A.E. HANDBOOK, be extended by specifying a ½-in. flange thickness.

STARTING-MOTOR PINIONS

(*Proposed Revision of S.A.E. Recommended Practice*)

As the present S.A.E. Recommended Practice for Starting-Motor Pinions, p. B18 of the S.A.E. HANDBOOK, has been misinterpreted to some extent since the last revision in wording was made in March, 1921, the Electrical Equipment Division has given careful consideration to rewording this recommendation. Although it is recognized that with the involute form of tooth the pitch-lines do not exist until the gears are mounted in their running positions, it is thought that the theoretical pitch-circle about which the teeth are generated may be considered as distinct from the running pitch-circles. Therefore

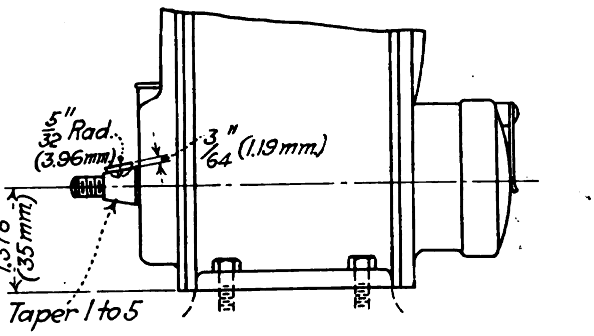
The Electrical Equipment Division recommends that the present S.A.E. Recommended Practice for Starting-Motor Pinions, p. B18 of the S.A.E. HANDBOOK be revised to read, "Flywheel starting-motors shall be equipped with an 8-10 pitch, 11-tooth, 20-deg. pressure-angle pinion and be installed so that the pitch-circle about which the teeth of the pinion are generated will be separated from 0.015 to 0.025 in. from the pitch-circle about which the teeth on the flywheel are generated."

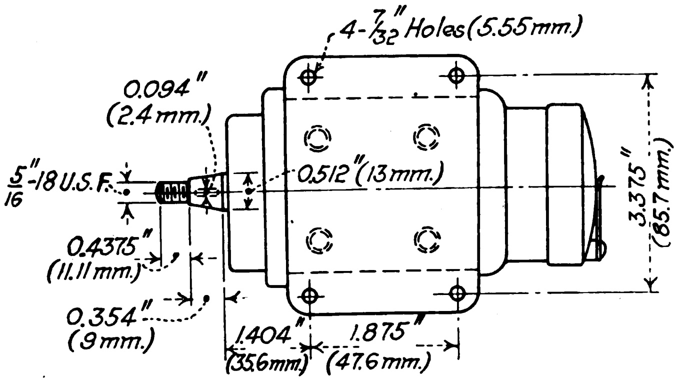
MAGNETO MOUNTINGS

(*Proposed S.A.E. Standard*)

Subsequent to the adoption in August of the S.A.E. Recommended Practice for the stationary-engine base-type magneto mounting, the Electrical Equipment Division was asked to consider the standardization of a flange type of mounting for installation where it is impossible to use the base type.

A review of the stationary-engine magneto situation indicated that such a standard mounting should be provided, and the Subdivision on magneto mountings was reappointed, with A. D. T. Libby, of the Splitdorf Electrical Co., chairman. At the October meeting of the Division Mr. Libby submitted a report covering a recommendation for this type of magneto, which is used primarily on one and two-cylinder stationary engines, the height from the base to the center-line of the shaft-end and the shaft-end dimensions being the same as the





dimensions specified in the present recommended practice, p. B16 of the S.A.E. HANDBOOK. The recommendation is in accord with magneto dimensions used extensively in this Country for the past 4 or 5 years. Therefore

The Electrical Equipment Division recommends for adoption as S.A.E. Recommended Practice the flange type of stationary-engine magneto mounting shown in the accompanying illustration.

SPARK-PLUGS

(*Proposed Revision of S.A.E. Standard*)

The present S.A.E. Standard for Spark-Plug Shells specifies ⅞ and 1⅛-in. widths across flats and does not include the dimensions for various types of terminal. As it was felt that it would be possible to standardize on certain types of terminal and to revise the standards so as to eliminate all but those dimensions necessary to permit interchangeability, a Subdivision was appointed to review the spark-plug standards and to formulate a report to the Electrical Equipment Division. The Subdivision appointed was as follows:

O. C. Rohde, *Chairman*	Champion Spark Plug Co.
D. L. Arnold	
B. M. deGuichard	A. C. Spark-Plug Co.
A. D. T. Libby	Splitdorf Electric Co.
C. S. Price	Bethlehem Spark-Plug Co.
M. J. Steele	Packard Motor Car Co.
L. M. Woolson	Packard Motor Car Co.

At a Subdivision meeting held in August it was stated that two-thirds of the spark-plugs manufactured at the present time have a ½-in. pipe thread, but that the general objection to pipe threads for spark-plugs is the difficulty of accurately maintaining the distance that the spark-plug screws into the cylinder; that metric spark-plugs are used only to a limited extent; that spark-plugs with ¾-in. pipe threads are being gradually discontinued, as are also the 5/16 and ⅜-in. spark-plugs with pipe threads; and that the majority of spark-plugs exclusive of the sizes mentioned are made with the S.A.E. Standard spark-plug thread of ⅞ in. -18. It was there-

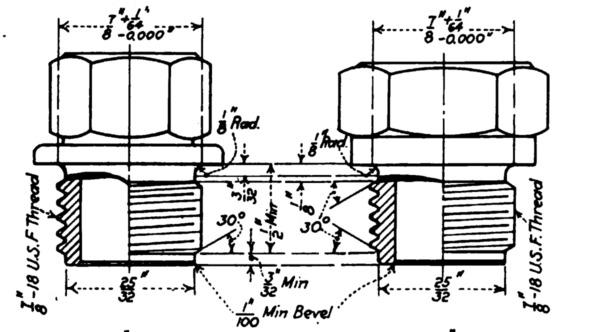
fore considered advisable by the Subdivision to retain the present standard thread as well as all dimensions affecting that portion of the spark-plug shell below the gasket.

In discussing the width across the flats of the hexagon or the "hex diameter" as it is generally called, it was brought out that 20 per cent of the spark-plugs with a ⅞-in. —18 thread have a ⅞-in. hex diameter, 20 per cent have a 1⅛-in. hex diameter and 60 per cent have a 15/16-in. hex diameter. Although there are two or three large users of the ⅞-in. hex spark-plugs, it was considered inadvisable to continue to recommend the use of this size because it does not leave sufficient wall to stand up under wrench strain, especially in two-piece spark-plug construction. It was recognized that it is possible to design two-piece spark-plugs for a ⅞-in. hex but as it would necessitate special inside parts that would not be used on the standard one-piece spark-plug, expensive tool-changes would be required. It would be, however, possible for machinery now used in making ⅞-in.-hex spark-plugs to be reset so as to make the 15/16-in.-hex without additional tooling expense.

At a subsequent meeting of the Electrical Equipment Division the report of the Subdivision was approved after careful consideration of the problems involved. It was recognized that the proposals of the Subdivision could not be adopted in present practice by certain automobile companies, but it was felt that the revised standards could be adopted at some future time by such companies simultaneously with other changes in engine design. Therefore

The Electrical Equipment Division recommends that the present S.A.E. Standard for Automobile Spark-Plug Shells, p. A10 of the S.A.E. HANDBOOK, be superseded by the following proposed S.A.E. Standard for Inch-Type Spark-Plugs.

INCH-TYPE SPARK-PLUGS



SMALL HEX. 15/16" ACROSS FLATS LARGE HEX. 1⅛" ACROSS FLATS

The spark-plug shell dimensions shall conform to those specified in the accompanying illustration for the 15/16 and 1⅛-in. hexagon spark-plugs respectively. All dimensions below the shoulder shall be identical for both sizes of spark-plug.

The outside diameter shall be 0.8750 in. and the number of threads per inch shall be 18. The limits for the pitch-diameter shall be 0.8389 in. maximum and 0.8348 in. minimum for the spark-plug threads and 0.8430 in. maximum and 0.8389 in. minimum for the tapped-hole threads.

SPARK-PLUG TERMINALS

Threaded Type.—The terminal thread shall be No. 8-32 (0.164 in. diameter) S.A.E. Coarse Thread

Ball Type.—For the ball type of spark-plug terminal the ball diameter shall be 13/32 in.

Slip-Type.—For the slip type of spark-plug terminal the ball diameter shall be 13/32 in. and shall have a groove width of 0.043 in. plus 0.006 minus 0.000 and a groove diameter of 0.203 in. plus 0.015 minus 0.000

Post Type.—For the post type of spark-plug terminal the large diameter of the taper post shall be 0.250 in. plus 0.000 and minus 0.003 and the neck diameter 0.220 in. plus 0.000 and minus 0.008

ENGINE DIVISION REPORT

Division Personnel

J. B. Fisher, *Chairman*	Waukesha Motor Co.
R. J. Broege, *Vice-Chairman*	Buda Co.
P. J. Dasey	Midwest Engine Corporation
S. F. Evelyn	Continental Motors Corporation
E. J. Hall	Hall-Scott Motor Co.
H. B. Massey	Holmes Automobile Co.
A. F. Milbrath	Wisconsin Motor Mfg. Co.
Louis Schwitzer	Automotive Parts Co.
M. J. Steele	Packard Motor Car Co.

FLYWHEEL HOUSINGS

(*Proposed Revision of S.A.E. Standard*)

Owing to the number of negative votes which were cast against the adoption of the Engine Division recommendation that the clearance space for crankshaft flywheel bolts should have a minimum diameter of 6½ in. and a minimum depth of ¾ in. in view of the general criticism that the recommendation was too limiting on clutch design, the Council referred the recommendation back to the Engine Division for joint consideration with the Transmission Division. The following conference Subdivision was therefore appointed: Messrs. Wemp and Copland representing the Transmission Division and Messrs. Evelyn and Steele the Engine Division:

E. E. Wemp, *Chairman*	Long Mfg. Co.
A. W. Copland	Detroit Gear Machine Co.
S. F. Evelyn	Continental Motors Corporation
F. J. Steele	Packard Motor Car Co.

A meeting of the Subdivision was held in Detroit on Oct. 24 and a report adopted which was submitted at subsequent meetings of the Engine and Transmission Divisions, the recommendation meeting with favorable consideration. The following is extracted from the Subdivision report.

An analysis of present practice indicates that the flywheel bolt clearance of small and medium-size engines comes well within the present standard of 6⅛ in., this dimension being equaled or exceeded only on engines of high torque capacity.

This condition will probably continue since it is to the engine builder's advantage from a cost standpoint to keep the flywheel bolt-circle diameter as small as possible, and as long as this is done the clutch manufacturer should have no difficulty in adapting his product to the engine. Insofar as the single-plate type of clutch is concerned, it is believed that the larger size of clutch required will provide for the necessary clearance space without difficulty.

With regard to the multiple-disc type, the Subdivision believes it is important to establish a maximum dimension from the flywheel face to the ends of the flywheel bolts so as to provide a clearance between the flywheel bolts and the driven member of the clutch.

Since ½-in. diameter crankshaft-bolts are used to such a large extent for this purpose, the Subdivision recommends a maximum dimension of 11/16 in. from the flywheel face to the ends of the flywheel bolts. This will allow the bolt to project ⅛ in. through a ½-in. castellated nut, or will easily take care of a ½-in. plain nut and a heavy lock-washer.

The Subdivision believes that until it is possible to standardize the actual crankshaft bolt-circle diameter the present clearance space of 6⅛ in. should be eliminated as being confusing and of no practical value. Therefore

The Engine Division recommends that the present S.A.E. Standard for Flywheel Housings, p. A1 of the S.A.E. HANDBOOK, reading, "The minimum diameter of the clearance space for crankshaft flywheel bolt-nuts shall be 6⅛ in. and the minimum depth ⅝ in.," should be omitted and the following substituted, "The maximum dimension from the flywheel face to the ends of the crankshaft flywheel bolts shall be 11/16 in."

FRAMES DIVISION REPORT

Division Personnel

E. V. Rippingille, *Chairman*	Watson Stabilator Co.
L. J. Fralick, *Vice-Chairman*	Hydraulic Steel Co.
C. C. Bowman	Standard Motor Truck Co.
E. A. DeWaters	Buick Motor Co.
O. B. Harmon	Parish & Bingham Co.
W. A. McKinley	Detroit Pressed Steel Co.
D. G. Roos	Locomobile Co.
C. W. Wright	A. O. Smith Corporation

RUNNING-BOARD BRACKETS

(Proposed Extension of S.A.E. Recommended Practice)

The present S.A.E. Recommended Practice for Running-Board Brackets, p. H23 of the S.A.E. HANDBOOK, was approved by the Society in March, 1922. Several negative votes were cast, however, because the gage of the stock was not specified, another variable thus being introduced in the recommended practice.

The Frames Division at a meeting in September reviewed this subject and felt that a definite gage-thickness would make the recommended practice more complete. Therefore

The Frames Division recommends that the present S.A.E. Recommended Practice for Running-Board Brackets, p. H23 of the S.A.E. HANDBOOK, be extended by the addition of a note reading, "The thickness of stock shall be 5/32 in. (0.156 in. or No. 9 U. S. gage).

IRON AND STEEL DIVISION REPORT

Division Personnel

F. P. Gilligan, *Chairman*	Henry Souther Engineering Corporation
W. C. Peterson, *Vice-Chairman*	Atlas Steel Corporation
R. M. Bird	Bethlehem Steel Co.
H. T. Chandler	C. H. Wills & Co.
A. L. Colby	Consulting Metallurgist
L. A. Danse	Cadillac Motor Car Co.
C. N. Dawe	Studebaker Corporation of America
B. H. DeLong	Carpenter Steel Co.
A. P. Eves	International Harvester Co.
H. L. Greene	Willys-Overland Co.
C. G. Heilman	General Motors Corporation
E. J. Janitzky	Illinois Steel Co.
J. B. Johnson	Air Service
F. C. Langenberg	Watertown Arsenal
A. H. Miller	Midvale Steel & Ordnance Co.
C. S. Moody	Minneapolis Steel & Machinery Co.
J. H. Nelson	Wyman-Gordon Co.
G. L. Norris	Vanadium Corporation of America
W. H. Phillips	R. D. Nuttall Co.
S. P. Rockwell	Consulting Metallurgist
M. P. Rumney	Detroit Steel Products Co.
C. F. W. Rys	Carnegie Steel Co.
R. B. Schenck	Buick Motor Co.
M. H. Schmid	United Alloy Steel Corporation
H. J. Stagg	Halcomb Steel Co.
J. M. Watson	Hupp Motor Car Corporation
J. H. G. Williams	Billings & Spencer

IRON AND STEEL SPECIFICATIONS

(Proposed Revision of S.A.E. Standard)

As a result of several proposals for the standardization of additional steel compositions, action was taken at the last meeting of the Iron and Steel Division adopting a policy of not including a new composition in the S.A.E. Standard for Iron and Steel Specifications unless it is shown that there is a sufficient tonnage of the steel used to warrant such action. Therefore, the Division felt that a statement should be included in the published report of the Iron and Steel Division to the effect that steels having different ranges of carbon than those specified in the present S.A.E. Standard are obtainable from the mills.

The Iron and Steel Division recommends that the present S.A.E. Standard for Iron and Steel Specifications be amplified by the inclusion of the following sentence in the first paragraph of Part I, p. D1 of the S.A.E. HANDBOOK, "The standard chemical compositions are for steels used by the automotive industry in large quantities, but other compositions having different ranges of carbon are obtainable from the mills."

In March 1922 the Society adopted the revised and extended report of the Iron and Steel Division covering the chemical compositions of the various iron and steel specifications. At that time the maximum sulphur-content recommended for S.A.E. Steel 3115 was 0.040 per cent, the maximum sulphur-content for the other steels of the 3100 series being 0.045 per cent. At the October meeting of the Division it was pointed out that there is no particular reason for not having the maximum sulphur-content the same for all the steels of the 3100 series. Therefore

The Iron and Steel Division recommends that the present S.A.E. Standard for Iron and Steel Specifications, Part III—Chemical Compositions, p. D4 of the S.A.E. HANDBOOK, be revised so as to specify a maximum sulphur-content of 0.045 instead of 0.040 per cent for S.A.E. Steel 3115.

LIGHTING DIVISION REPORT

Division Personnel

W. A. McKay, *Chairman*	Westinghouse Lamp Co.
C. A. Michel, *Vice-Chairman*	Guide Motor Lamp Mfg. Co.
J. T. Caldwell	National Lamp Works
C. E. Godley	Edmunds & Jones Corporation
C. A. B. Halvorson, Jr.	General Electric Co.
L. C. Porter	Edison Lamp Works
E. S. Preston	Chicago Electric Mfg. Co.
C. D. Ryder	Corcoran-Victor Co.
J. C. Stearns	Culver-Stearns Mfg. Co.
T. I. Walker	Providence Base Works
E. E. Wood	Miniature Incandescent Lamp Corporation
Ernest Wooler	Cleveland Automobile Co.

ELECTRIC INCANDESCENT LAMPS

(Proposed Revision of S.A.E. Standard)

The Society has adopted head-lamp illumination specifications requiring the use of 21-cp. electric incandescent lamps and, as this candlepower is required by law in many of the States having head-lamp regulations, it was felt at the last meeting of the Lighting Division that the present S.A.E. Standard for Electric Incandescent Lamps should be extended to specify this candlepower. Therefore

The Lighting Division recommends that the present S.A.E. Standard for Electric Incandescent Lamps, p. B3 of the S.A.E. HANDBOOK, should be amplified by the addition of a footnote reading, "Incandescent lamps for automobile head-lamps shall be of the gas-filled type and of 21 cp."

LUBRICANTS DIVISION

Division Personnel

H. C. Mougey, *Chairman*	General Motors Research Corporation
W. E. Jominy, *Vice-Chairman*	Studebaker Corporation of America

CRANKCASE LUBRICATING OILS

General.—These specifications cover grades of petroleum oil for the lubrication of internal-combustion engines, except aircraft, and are not recommended for the lubrication of turbines.

Compounded lubricating oils containing products other than those derived from petroleum are not dealt with in these specifications.

Specification No.[2]	Explanatory Grade Names	Flash-Point, Deg. Fahr., Min.	Fire-Point, Deg. Fahr., Min.	Viscosity, Saybolt Sec.				Dilution with Water-White Kerosene for No. 5 N.P.A. Color		Pour Test, Deg. Fahr., Max.	Conradson Carbon Residue, Per Cent, Max.	Corrosion Test
				100 Deg. Fahr.		210 Deg. Fahr.						
				Min.	Max.	Min.	Max.	Parts, Kerosene	Parts, Oil			
20	Light	325	365	180	220	42	...	50	50	35	0.20	Required for all grades
020	Light	325	365	180	220	42	...	50	50	10	0.20	
30	Medium	335	380	270	330	44	...	50	50	40	0.30	
030	Medium	335	380	270	330	44	...	50	50	10	0.30	
40	Medium	345	390	360	440	46	...	60	40	45	0.40	
50	Heavy	355	400	450	575	50	...	70	30	50	0.60	
60	Heavy	360	...	...	...	55	65	80	20	55	0.80	
80	Extra Heavy	380	...	...	...	75	85	85	15	55	1.50	
95	Extra Heavy	390	...	...	...	90	100	90	10	55	1.75	
115	Extra Heavy	400	...	...	...	110	120	95	5	60	2.00	

[2]For Specifications Nos. 20 to 50, inclusive, the numbers indicate the first two figures of the average Saybolt viscosity in seconds at 100 deg. fahr. for the grades indicated. The first cipher in Specifications Nos. 020 and 030 indicates that the pour-test value of these two grades is 10 deg. fahr. The numbers for Specifications Nos. 60 to 115, inclusive, indicate the average Saybolt viscosity in seconds at 210 deg. fahr.

Corrosion Test.—The following corrosion test shall not cause discoloration of copper strip. Place a clean piece of mechanically polished pure strip-copper about ½ in. wide and 3 in. long, and 10 cc. of the oil to be tested, in a clean test-tube. Close the tube with a vented stopper and hold for 3 hr. at 212 deg. fahr. Rinse the copper strip with sulphur-free acetone and compare it with a similar strip of freshly polished copper.

Sydney Bevin	Tide Water Oil Co.
P. J. Dasey	Midwest Engine Corporation
A. P. Eves	International Harvester Co.
W. H. Herschel	Bureau of Standards
K. G. Mackenzie	Texas Co.
W. E. Perdew	Union Petroleum Co.
W. D. Reese, Jr.	Fifth Ave. Coach Co.
H. G. Smith	Formerly with Atlantic Refining Co.
J. W. Stack	Standard Oil Co.

Crankcase Lubricating Oils
(Proposed S.A.E. Standard)

In 1912 the Society adopted a specification known as No. 26 for pure mineral automobile engine light lubricating oil. This specification proved satisfactory for a number of years as applying to paraffin-base oils. In April, 1920, at the request of the Tractor Division, the Lubricants Division prepared to make a careful study of the requirements for more extensive standard lubricating oil specifications. The basis for this work was Bulletin No. 4, dated April, 1920, issued by the Bureau of Mines. The Lubricants Division attended several meetings of the Government petroleum fuel committees and in November 1921 submitted a tentative specification based partly on the work of the Government and partly on information obtained through a questionnaire sent out by the Division. The comments on the tentative specification were considered and a revised proposal published in the June 1922 issue of The Journal for discussion at the Standards Committee meeting at White Sulphur Springs. Subsequent to this consideration of the report, a number of tests of five unknown kinds of oil were made by members of the Division, the samples being identified only by numbers. The results of these tests indicated the necessity for recognizing the difficulties in securing uniform results throughout the industry in checking oils to the proposed specifications. The limiting values in the specification were readjusted accordingly and the revised report published in the November 1922 issue of The Journal, one of the important points in the report being the classification of the grades of oils by numbers indicating the viscosity characteristics of the respective grades.

There were a number of differences between the proposed specification and those issued by the Interdepartmental Committee of the Government on Petroleum Specifications and arrangements were accordingly made for a joint conference of the Society's Lubricants Division, the Advisory Committee on Petroleum Specifications of the American Petroleum Institute and the Interdepartmental Committee in Washington on Nov. 13 to secure uniformity in the specifications. The various points were thoroughly discussed and informally acted upon, following which the members of the Division voted to submit the following specifications for adoption as S.A.E. Standard for internal-combustion engine lubricating oils as it was felt that the Society should adopt such specifications as soon as possible. The chairman of the Interdepartmental Committee stated at the conference that his committee could not take formal action at that time, but would probably meet early in February to consider the points under discussion in connection with the revised specification of the Society and the Governmental specifications. Therefore

The Lubricants Division recommends the adoption of the accompanying specification as S.A.E. Standard.

NOMENCLATURE DIVISION REPORT

Division Personnel

H. L. Pope, *Chairman*	Wright Aeronautical Corporation
W. P. Kennedy, *Vice-Chairman*	Kennedy Engineering Corporation

J. B. Bartholomew	Avery Co.
W. F. Borgerd	International Harvester Co.
W. S. Bouton	Hendee Mfg. Co.
W. J. Brandon	Avery Co.
H. R. Cobleigh	National Automobile Chamber of Commerce
W. P. Culver	American Auto Parts Co.
A. B. Cumner	Autocar Co.
L. S. Keilholtz	Delco-Light Co.
V. E. McMullen	Hercules Gas Engine Co.
Leonard Ochtman, Jr.	Joseph Van Blerck, Inc.
W. T. Thomas	Thomas-Morse Aircraft Corporation
J. G. Vincent	Packard Motor Car Co.
L. C. Voyles	Nordyke & Marmon Co.

Automobile Nomenclature

(Proposed Revision of S.A.E. Standard)

The function of the Nomenclature Division in the standardization of automotive nomenclature is primarily to unify and correlate nomenclatures worked out by other Divisions of the Standards Committee, and to arrange them properly in the present S.A.E. Standard for Automobile Nomenclature, p. K1 of the S.A.E. HANDBOOK.

The present nomenclature of differential gears in Division XII, Group 4, on p. K15 of the S.A.E. HANDBOOK, is considered obsolete because of the developments in differential design. The Axle and Wheels Division therefore appointed the following Subdivision on Differentials to revise this part of the present standard.

S. O. White, *Chairman*	Warner Gear Co.
F. E. McMullen	Gleason Works
D. D. Ormsby	Brown-Lipe-Chapin Co.

Mr. White was also a member of a similar committee appointed by the American Gear Manufacturers Association, that considered and approved the report of the Subdivision prior to its final consideration by the Axle and Wheels Division. Therefore

> The Nomenclature Division recommends that the following nomenclature be adopted as a revision of the present differential nomenclature, Division XII, Group 4 of the present S.A.E. Standard for automobile nomenclature.

Differential Nomenclature

FOUR-PINION TWO-PIECE CASE BEVEL DRIVE

Differential[3]
Bevel-Drive Pinion[4]
Bevel-Drive Gear
Differential Case Flange Half
Differential Case Plain Half
Differential Bearing Sleeve
Differential Case Bolt
Bevel-Drive Gear Rivet or Screw
Differential Side Gear
Differential Spider Pinion
Differential Spider

TWO-PINION ONE-PIECE CASE BEVEL DRIVE

Differential[3]
Bevel-Drive Pinion[4]
Bevel-Drive Gear
Differential Case
Differential Bearing Sleeve
Bevel-Drive Gear Rivet or Screw
Differential Side Gear
Differential Cross-Pin Pinion
Differential Cross Pin
Differential Cross-Pin Lock
Differential Side Gear Spacer

[3] *Differential.*—A differential comprises a case and internal parts only.
[4] *Bevel-Drive Pinion or Worm.*—A bevel-drive pinion or worm may be of either the "bored" or the "shaft" type.

WORM-GEAR DRIVE

Differential[3]
Worm[4]
Worm Gear
Differential Case, right hand
Differential Case, left hand
Differential Bearing Sleeve
Differential Case Bolt
Worm-Gear Rivet or Screw
Differential Side Gear
Differential Pinion
Differential Spider or Cross Pin
Differential Cross-Pin Lock
Differential Side Gear Spacer

PARTS AND FITTINGS DIVISION REPORT

Division Personnel

F. G. Whittington, *Chairman*	Stewart-Warner Speedometer Corporation
W. C. Keys, *Vice-Chairman*	Gabriel Mfg. Co.
Clarence Carson	Dodge Bros.
E. R. Douglas	Formerly with Cincinnati Ball Crank Co.
H. B. Garman	Garman Mfg. Co.
H. S. Jandus	C. G. Spring Co.
F. W. Slack	Peerless Motor Car Co.
C. W. Spicer	Spicer Mfg. Corporation
Alex Taub	General Motors Corporation
E. W. Weaver	Weaver & Kemble Co.

Plain Washers

(Proposed S. A. E. Standard)

In August, 1922, the Society adopted the report of the Parts and Fittings Division covering Plain Washers for S.A.E. Standard bolts. The Division has continued this standardization to include plain washers for machine screws in order that the standards for screws, bolts and nuts, lock-washers and plain washers may be complete. Therefore

> The Parts and Fittings Division recommends that the present S.A.E. Standard for Plain Washers, p. C5c of the S.A.E. HANDBOOK, be extended to include the dimensions for washers used with screws below 1/4 in. diameter as given in the accompanying table. As the material used for the proposed washers may be other than steel, it is also recommended that the title of the present standard be changed to "Plain Washers."

The present S.A.E. Standard for Plain Washers is given also in the accompanying table.

PROPOSED S. A. E. STANDARD DIMENSIONS FOR PLAIN WASHERS

Screw and Bolt Sizes	Inside Diameter	Outside Diameter	Thickness ±0.010
2	3/32	1/4	1/32
4	1/8	5/16	1/32
6	5/32	3/8	3/64
8	3/16	7/16	3/64
10	7/32	1/2	1/16
12	1/4	9/16	1/16
1/4	9/32	5/8	1/16
5/16	11/32	11/16	1/16
3/8	13/32	13/16	1/16
7/16	15/32	15/16	1/16
1/2	17/32	1 1/16	3/32
9/16	19/32	1 3/16	3/32
5/8	21/32	1 5/16	3/32
11/16	23/32	1 3/8	3/32
3/4	13/16	1 1/2	1/8
7/8	15/16	1 3/4	1/8
1	1 1/16	2	1/8
1 1/8	1 3/16	2 1/4	1/8
1 1/4	1 5/16	2 1/2	5/32
1 3/8	1 7/16	2 3/4	5/32
1 1/2	1 9/16	3	5/32

All dimensions in inches. Washers shall be flat and free from burrs.

Rod-Ends

(Proposed Revision of S.A.E. Standard)

The present S.A.E. Standard for Rod-Ends, p. C8 of the S.A.E. Handbook, specified a 3/16-in.-32 thread for the smallest size of the light series rod-end, whereas the S.A.E. Standard for Ball-and-Socket Joints, p. C52, specified a No. 10-32 thread for the corresponding size. As generally accepted standards for screws, bolts, nuts and similar threaded parts are made in numbered or decimal sizes below ¼ in., instead of in fractional sizes, and as it should be possible to use the same thread at both ends of a rod when used with a ball-and-socket joint at one end and an adjustable rod-end at the other,

The Parts and Fittings Division recommends that the present S.A.E. Standard for Rod-ends, p. C8 of the S.A.E. Handbook, be revised to specify a No. 10-32 in place of the 3/16-in.-32 thread for the smallest size of the light-rod-end series.

PASSENGER-CAR BODY DIVISION REPORT

Division Personnel

G. E. Goddard, *Chairman*	Dodge Bros.
A. J. Neerken, *Vice-Chairman*	Hupp Motor Car Corporation
S. J. Baum	Brewster & Co.
E. G. Budd	Edward G. Budd Mfg. Co.
J. S. Burdick	Buffalo Body Corporation
O. H. Clark	Zeder-Skelton-Breer Engineering Co.
A. E. Garrels	Studebaker Corporation of America
E. W. Goodwin	Maxwell Motor Corporation
G. W. Kerr	Rolls-Royce of America, Inc.
G. J. Mercer	Consulting Engineer
H. C. Nelson	Mullins Body Corporation

Door Locks and Handles

(Proposed S.A.E. Recommended Practice)

In March, 1922 the Passenger-Car Body Division report on door-handle squares was approved by the Society. Further discussion of this recommendation at recent meetings of the Division indicated the desirability of specifying the size of the broached hole in the door-handle cam. Data obtained from manufacturers formed the basis for the following recommendation.

The Division has also considered the standardization of door-handle escutcheon plates. The Subdivision that investigated this subject recommended that the center-to-center distance of the two holes and the size of screws should be standardized in accordance with the prevailing practice as given below.

The Division has felt that in many instances door-handle bars are too short and that a sufficient clearance between the handle and the face of the door panel is not allowed. Although this is not considered very important as a standard, it is thought that a recommendation indicating desirable practice would lead to improvements in many designs.

The Division has also considered the various bevels used for door-lock face-plates and believes that a standard will be of value, especially to door-lock manufacturers. An analysis of current production has shown that a large number of angles are used. This subject has been considered also by the Automobile Body Builders' Association, which suggested adopting angles of 6 and 9 deg., that is, included angles of 96 and 99 deg. It was felt, however, that one angle is sufficient for doors for all types of body. Therefore

The Passenger-Car Body Division recommends that the present S.A.E. Recommended Practice for Door-Handles, p. K45 of the S.A.E. Handbook, be extended by the addition of the following recommendations:

(1) No. 10 wood screws 1 in. long shall be used for door-handle escutcheon plates which shall have countersunk holes spaced 1⅛ in. from center-to-center

(2) The broached hole in door-handle cams on the inside handle shall be 5/16 in. with tolerances of plus or minus 0.001 in.

(3) The included angle of the two sides of door-lock face-plates shall be 96 deg. for doors for all types of body and shall be inspected to gage

(4) Bar-type door-handles shall have a bar at least 3½ in. long and the clearance from the inside of the bar to the face of the door panel shall be not less than 1½ in.

Wood Screws

(Proposed S.A.E. Recommended Practice)

One of the important problems in passenger-car body building is the drilling of holes for wood screws so that the maximum holding power of the screw may be obtained. The specifications formulated by the Federal Specifications Board for wood screws were reviewed by the Passenger-Car Body Division at the October meeting, but it was felt that further simplification than that proposed could be accomplished by limiting the recommendation to the even numbered sizes beginning with the No. 4 size, except the No. 5 size which is used to a large extent. It was considered that the use of wood screws having cut threads should be recommended, as their holding power is greater than that of screws made with rolled threads. Therefore

The Passenger-Car Body Division proposes the following for adoption as S.A.E. Recommended Practice:

(1) Only even number sizes of wood screws commencing with the No. 4 size, with the exception of the No. 5 size, shall be used in automobile body construction. The heads may be either upset or turned, but the slots in the heads shall be milled and the threads cut

(2) The diameter of the drill used shall be the same as the maximum wood-screw diameter at the root of the full thread. All countersunk holes for wood screws shall have an included angle of 82 deg.

Plate Glass

(Proposed S.A.E. Recommended Practice)

One of the most important materials entering into body construction is the glass used in windshields and windows, and the need for dimensional specifications thereof was recognized by the Division. A. J. Neerken, who was appointed a Subdivision of one to prepare a tentative recommendation, obtained information from body builders as to the dimensions and grades of glass used and a recommendation was based on this information and specifications were drawn up by the glass manufacturers and users in cooperation with the Bureau of Standards. It is felt that one of the most valuable results of this standardization will be to make possible a clear and undistorted view. The recommendations proposed will be of value to glass users also in drawing up purchasing specifications and lead to the elimination of many odd-size sheets. Therefore the Passenger-Car Body Division recommends that the following proposal be adopted as S.A.E. Recommended Practice.

Glass sizes for windshields and doors shall be selected in increments of even 2 in. The glass producer should be furnished with a templet of the finished glass.

Only polished plate glass shall be used in windshields and front-quarter door windows.

Polished plate glass for windshields and passenger-car windows shall be of two grades: "Selected Glazing" and "Glazing."

"Selected glazing" plate glass shall contain practically no visible imperfections under specified conditions of inspection. Very fine scattered seeds are permissible.

"Glazing" plate glass shall contain no other visible imperfections than a few scattered seeds and occasional faint strings or faint short finish-marks.

The thickness of plate glass shall be 3/16 in. with a tolerance of plus or minus 1/32 in. with variations in individual plates of not more than 1/64 in.

The following definitions of the terms used in the above specifications are taken from the report of the committee on standards of plate glass organized by the Bureau of Standards:

Seeds.—Minute bubbles smaller than 0.015 in. diameter. These are visible only on close inspection, usually appearing as small specks, and are an inherent defect in the best quality of plate glass

Strings.—Light, wavy, transparent lines on the surface, appearing as though a thread of glass had been partially incorporated into the sheet

Short Finish.—Poor polish is lack of smoothness, an improperly finished surface which has the appearance of being slightly pitted and wavy when the surface is viewed by reflected light. These indentations which are slight have a polished surface rather than a ground surface, but the general effect is a slight dulling of the surface. Poor polish is usually caused by improper grinding. Spots on the surface where the fine grinding has not proceeded far enough to produce a smooth surface before polishing will not polish smooth

SCREW-THREADS DIVISION REPORT

Division Personnel

E. H. Ehrman, *Chairman*	Chicago Screw Co.
O. B. Zimmerman, *Vice-Chairman*	International Harvester Co.
Earle Buckingham	Pratt & Whitney Co.
E. Burdsall	Russell, Burdsall & Ward Bolt and Nut Co.
Luther Burlingame	Brown & Sharpe Co.
G. S. Case	Lamson & Sessions Co.
W. R. Mitchell	National Acme Mfg. Co.
Alex Taub	General Motors Corporation

Gages and Gaging

(*Proposed General Information*)

The Screw-Threads Division has undertaken to prepare a series of articles covering various phases of screw-thread practice and matters germane thereto for publication as general information in THE JOURNAL and subsequently in the S.A.E. HANDBOOK. The Division submitted an article to the Standards Committee in June on the fundamentals of gages and gaging for screw-thread products, but this report was referred back for amplification with regard to the importance of gaging for lead error. The second and third paragraphs of Section 4 dealing with this phase of the subject have been added. Therefore

The Screw-Threads Division recommends that the following article be approved by the Standards Committee for publication as general information only.

GAGES AND GAGING FOR SCREW-THREADS

I. Introductory

The art of measuring screw-threads has developed very rapidly during the past few years. This development still continues, so that it would be extremely inadvisable to attempt to specify any one definite method as standard for this purpose. The object of this report is to establish so far as possible the fundamentals of this subject, and to point out various practices now successfully used.

II. Fundamentals

Object of Gaging.—The final result sought by gaging is interchangeable manufacture in some degree. This means that the mating parts can be assembled without *fitting* one part to another and, when assembled, the mechanism will function properly. Gaging should be employed more to prevent unsatisfactory parts from being produced than to sort out the correct parts from the incorrect ones.

Direction of Tolerances on Gages.—The extreme sizes for all limit-gages shall never exceed the extreme limits of the part being produced. All variations in the gages, whatever their cause or purpose, shall bring these gages within these extreme limits. Thus a gage that represents a minimum limit may be larger, but never smaller, than the minimum size specified for the part being produced, while the gage that represents a maximum limit may be smaller, but never larger, than the maximum size specified for the part being produced.

Temperature at Which Gages Shall Be Standard.—Gages shall be standard at a temperature of 68 deg. fahr.

Standard or Basic Size.—The standard or basic size, as physically represented by a correct standard master-gage, is the line at which interference begins between mating parts.

Purpose of "Go" and "Not Go" Gages.—The "Go" gages, which are the gages that represent the maximum limit of the internal member and the minimum limit of the external member, control the allowances between mating surfaces and also control interchangeability. "Go" gages control the maximum tightness in the fit of mating parts. Parts that are acceptable to proper "Go" gages will always interchange. Successful interchangeable manufacturing has been carried on for many years with the use of "Go" gages only.

The "Not Go" gages limit the extent of the permissible variations, thus limiting the amount of looseness between mating parts. "Not Go" gages control the maximum looseness in the fit of mating parts and thus control, in large measure, the proper functioning of the assembled mechanisms.

III. Gage Classification

Master Gage.—The master gage is a plug thread-gage that represents as exactly as possible the physical dimensions of the nominal or basic size of the component. A *standard master-gage* shall be accompanied by a record of its measurement and the gage should be used with knowledge of any deviations or corrections. In case of question, the deviations of this gage from the exact standard shall be ascertained by the Bureau of Standards.

Reference Gage.—A commonly used name for a master gage. Sometimes such gages include those that represent the extreme limits of the product and are used to check the inspection and working gages.

Gages Used to Measure the Product.—The gages used to check the product may be divided into two general types: mechanical and optical. Both types, however, are controlled by the master gages. In general, most of the parts accepted by one method of gaging will be accepted by the other. It should be pointed out, however, that those parts which are close in size to either rejection-point, may be accepted by one system and rejected by the other.

Mechanical gages are often divided into two classes: inspection gages and working gages. Inspection gages are for the use of the inspector in accepting the product. They are generally of the same design as the working gages and the dimensions are such that they represent very nearly the extreme limits of the part being produced. Working gages are those used by the workman to check the parts as they are machined. It is recommended that, when successive inspections are required, the working gages, by either design or selection, be of such dimensions that they are inside the limits of the gages used in succeeding inspections.

When gages of the optical type are employed, the same or duplicate instruments are used for both classes of inspection. No distinction in size is necessary, as the elements of wear and "feel" are not involved in this method of measuring.

IV. Gaging Practices and Gages

The production of accurate parts is primarily a matter of eternal vigilance. The smaller the limits that are to be maintained, the more complete the inspection or gaging system must be. To secure satisfactory results, the manufacturing tools provided must be sufficently accurate and the manufacturing methods sufficiently reliable to produce the required results. After tools and methods of proved reliability are provided, the next point is to watch the wear on the tools or their set-up to assure the maintenance of the required conditions. This is accomplished sometimes by a periodical test of the tools, sometimes by periodical gaging of the product, and sometimes by both.

A screw-thread is comprised of several elements: first, the outside or major diameter; second, the pitch-diameter; third, the core or minor diameter; fourth, the angle of the thread form; and fifth, the lead. There is a broad general principle in regard to limit gages that should always be kept in mind. Where compound tolerances are not involved, a "Go" gage with fixed measuring surfaces may check as many dimensions at one time as desired, and effective inspection will be secured. On the other hand, an effective "Not Go" gage can check only one dimension. By effective inspection is meant assurance that specified requirements in regard to size are not exceeded.

The most difficult element of a screw-thread to gage is the lead. Lead-testing devices for checking tools and gages are available, but in general their operation is too slow for use as production inspection equipment. In addition, the lead is the most important element of a screw-thread as regards the nature of the contact between mating parts. Furthermore, an error in lead has almost double the effect of an equal error in diameter as regards interchangeability. Thus, for exacting threaded work, if the method of inspection of the parts produced does not effectively inspect for lead errors, the tools used to produce these parts must be carefully inspected for lead.

Thread Micrometers.—Thread micrometers are used extensively to measure the pitch-diameter of taps and threaded internal parts. Thread micrometers should be calibrated periodically against a master gage, to avoid errors due to wear on the anvils of the instrument. Thread micrometers give no indication of lead and angle errors; therefore, the results of tests with thread micrometers alone cannot be taken as conclusive.

Thread Snap-Gages.—Thread snap-gages, generally consisting of conical points, are commonly used to measure the pitch-diameter of screws and other threaded internal parts. As in the case of thread micrometers, these gages give no indication of lead and angle errors. Therefore, the results of tests with them alone cannot be taken as conclusive.

Ring Thread-Gages.—Ring thread-gages are used extensively to measure the thread on internal parts. These are usually adjustable and are adjusted to suitable master or reference gages. Where parts are to be produced within specified limits, "Go" and "Not Go" gages are required. The thread on the "Go" gage is made of full form with its major diameter cleared or undercut to give a suitable clearance for grinding or lapping. The "Not Go" gage should be made primarily to check the minimum pitch-diameter. The minor diameter of such a gage should therefore never be smaller than the minor diameter of its corresponding "Go" gage, and its major diameter should be cleared as in the case of the "Go" gage. The use of such gages gives a certain measure of lead and angle errors, as well as of pitch-diameter errors. A proper "Go" gage will reject any parts that exceed the maximum dimensions specified. The "Not Go" gage, however, does not necessarily reject all parts that exceed the specified cumulative tolerance. It is possible, with the use of such gages, to accept parts that exceed this cumulative tolerance because of lead or angle errors, or both. With the proper check on tools and manufacturing methods, however, such possibilities are the exception. Such gages have been used successfully for many years.

Thread Comparators.—A recent development in the art of measuring threaded parts is the thread comparator, usually an optical instrument. These optical instruments throw an enlarged image of the thread upon a screen where it is compared with the enlarged outline of the required form. The location of the form used for comparison is made to agree with the image of the master gage. With such instruments all errors, both individual and cumulative, of diameter, lead and angle can be determined readily. These instruments can be adapted to measure taps and threaded internal parts.

Plug Thread-Gages.—Plug thread-gages are used exclusively at the present time to measure threaded holes or threaded external parts. Where parts are to be produced within specified limits, "Go" and "Not Go" gages are required. The thread on the "Go" gage is made of full form with its minor diameter cleared or undercut to give a suitable clearance for grinding or lapping. The "Not Go" gage should be made primarily to check the maximum pitch-diameter. The major diameter of such a gage should therefore never be larger than the major diameter of its corresponding "Go" gage, and its minor diameter should be cleared as in the case of the "Go" gage. The use of such gages gives a certain measure of lead and angle errors, as well as of pitch-diameter errors. A proper "Go" gage will reject any parts that exceed the minimum dimensions specified. The "Not Go" gage, however, as in the case of the ring thread-gage, does not necessarily reject all parts that exceed the specified cumulative tolerance.

Methods of Inspecting Screws.—One practice of inspecting screws produced on automatic machines is to provide a ring thread-gage set to approximately the mean size between the maximum and minimum limits. The threading tools are then set so that the product enters this intermediate gage, but will not enter the minimum or "Not Go" gage. The machine is then started up and parts are tested periodically with the regular "Go" gage and the intermediate gage. When the parts have increased in size so that they will not enter the intermediate gage more than three or four turns, the set-up is changed, even though the parts are still acceptable to the "Go" gage.

A very similar plan can be followed when a screw-thread comparator is employed. The original set-up should be toward the minimum limit and the set-up should be changed as the maximum limit is approached.

Reference has been made to successive inspections. Although the manufacturer may give but one inspection, it should be realized that the purchaser often inspects the product to assure that the prescribed specifications have been fulfilled. Therefore, to reduce the possibilities of disagreement to a minimum, the manufacturer should strive to produce parts well within the specified limits rather than close to the limiting sizes.

Thread micrometers and thread snap-gages are used extensively for testing the product as it is produced. As these instruments do not test all elements of the screw-thread, a "Go" gage should always be used as a supplementary test. Thread micrometers are a very effective means of watching the change in set-up due to wear on tools, etc.

Methods of Inspecting Tapped Holes.—One practice of inspecting tapped holes is first to inspect the tap, and then test the tapped holes periodically with suitable gages. The tap can be watched for wear by testing the tapped holes with a "Go" thread-gage. One widely used practice consists of using a "Go" plug thread-gage and a "Not Go" plain plug-gage for the minor diameter.

Another practice of inspecting taps is to measure the several elements, such as pitch-diameter, angle and lead; and still another consists of tapping a hole with each tap before it is issued from the tool-crib and testing these tapped holes with "Go" and "Not Go" plug thread-gages.

V. Inspection of Gages

When successive inspections in the same plant are involved, it is good practice to inspect all gages of the same nominal size against each other periodically, and to distribute these gages so that the earlier inspections will be made with those that are the greatest amount inside of the component tolerance, and the later inspections with those gages closest in size to the component tolerance.

STORAGE-BATTERY DIVISION REPORT

Division Personnel

W. E. Holland, *Chairman*	Philadelphia Storage Battery Co.
I. M. Noble, *Vice-Chairman*	Consulting Engineer
G. L. Bixby	Detroit Electric Car Co.
R. N. Chamberlain	Gould Storage Battery Co.
E. L. Clark	Commercial Truck Co.
Bruce Ford	Electric Storage Battery Co.
W. E. Gossling	Prest-O-Lite Co., Inc.
C. T. Klug	Willard Storage Battery Co.
R. C. Mitchell	Edison Storage Battery Co.
J. L. Rupp	Westinghouse Union Battery Co.
G. W. Vinal	Bureau of Standards
W. G. Wall	National Motor Car & Vehicle Co.

Ratings of Storage Batteries for Isolated Electric-Lighting Plants

(Proposed S.A.E. Recommended Practice)

At a meeting of the Storage Battery Division in September the work in establishing a storage-battery rating for isolated electric-lighting plants was reviewed with particular reference to the action of the Isolated Electric-Lighting Plant Division on July 28, 1921, in recommending a standard rating of capacity expressed in terms of watt-hours based on a continuous 8-hr. discharge test. This recommendation did not receive general approval when submitted to letter ballot of storage-battery and lighting-plant manufacturers.

In view of the fact that at the present time several methods of rating storage-batteries are used, it is considered advisable by the Storage-Battery Division to recommend that either the intermittent or the continuous ratings should be used in accordance with the recommendation of the Storage-Battery Subdivision of the Electrical Equipment Division made on May 28, 1919. The present Storage-Battery Division is the successor of the Storage Battery Subdivision in existence at that time.

As the subject of storage-battery ratings was reassigned to the Storage Battery Division by the Council, the Storage Battery Division recommends for adoption as S.A.E. Recommended Practice the following methods of rating storage batteries for isolated electric-lighting plants.

RATINGS OF STORAGE BATTERIES FOR ISOLATED ELECTRIC-LIGHTING PLANTS

(1) Lead-acid storage batteries for isolated electric-lighting-plant service shall have two ratings, a continuous rating and an intermittent rating. The ratings shall be determined at an initial temperature of 80 deg. fahr. and be based on a final voltage of not less than 1.75 volts per cell.

(2) *Continuous Rating*—The continuous rating shall be the capacity in ampere-hours of the battery when it is discharged continuously at the 8-hr. rate.

(3) *Intermittent Rating*—The intermittent rating shall be the capacity in ampere-hours of the battery when it is discharged intermittently over a period of 72 hr.

In order to avoid night work, the following test is suggested:

> Discharge at a rate of current equal to 1/24 of the rated ampere-hour capacity of the battery for an initial discharge period of 4-hr., followed by a 16-hr. rest; then two 8-hr. discharge periods, each followed by a 16-hr. rest; the final discharge period being 4-hr.

The short periods at the beginning and at the end permit the test to begin at noon of the first day and end at noon of the fourth day.

(4) Except for the final voltage per cell the ratings apply to nickel-iron batteries as well as to lead-acid batteries.

(5) Battery manufacturers should specify both ratings in their catalogs using the following form:

S.A.E. Ampere-Hour Capacity Ratings: 100-140

The first number represents the capacity based on the continuous rating and the second number the capacity based on the intermittent rating.

As it is thought that the situation can be covered adequately at this time only by specifying both the continuous and the intermittent ratings, the Storage Battery Division makes this recommendation with the express understanding that in case either one of the two ratings proposed is eliminated by subsequent Standards Committee or Society action, the recommendation will be referred back to the Division.

Storage-Battery Monoblock Containers

(Proposed S.A.E. Recommended Practice)

In November, 1921 the standardization of storage-battery containers was suggested by a member of the Storage-Battery Division. A tentative proposal was submitted and considered at several meetings of the Storage-Battery Division during 1922, the proposal having met with the favorable consideration of storage-battery and hard-rubber manufacturers before final action was taken. Therefore

> The Storage Battery Division recommends for adoption as S.A.E. Recommended Practice the accompanying dimensions for storage-battery monoblock containers.

MONOBLOC CONTAINERS FOR STARTING AND LIGHTING BATTERIES

Height.—Containers shall be made in two heights only; namely, the B height for plates approximately 4¾ in. high, and the C height for plates approximately 5¼ in. high.

Overall Height.—The height from the outside of the bottom of the case to the top of the handle shall not exceed 9¼ in. for B containers, or 9⅞ in. for C containers.

Height from Top of Ribs to Top of Container.—These heights shall be:

B containers 6½ in. } Plus 0
C containers 7 in. } Minus 1/16 in.

Maximum variation between different compartments, 3/32 in.

Inside Width of Compartments.—5 31/32 in., plus or minus 1/32 in.

Inside Lengths of Compartments.—(a) 6-Compartment Containers

B Containers		C Containers	
S-3-B	1 5/16 in.	S-4-C	1 1/2 in.
S-4-B	1 1/2 in.	S-5-C	1 11/16 in.
S-5-B	1 11/16 in.		

(b) 3-Compartment Containers

B Containers		C Containers	
S- 7-B	2 1/16 in.	S- 8-C	2 3/8 in.
S- 8-B	2 3/8 in.	S-10-C	2 13/16 in.
S- 9-B	2 7/16 in.	S-13-C	3 1/4 in.
S-10-B	2 13/16 in.	S-14-C	3 5/16 in.
S-16-B	3 11/16 in.	S-16-C	3 11/16 in.
S-18-B	3 15/16 in.		
S-19-B	4 1/8 in.		

(c) Tolerances of plus 0 and minus 1/32 in. shall apply to container lengths of 3 5/16 in. or less and tolerances of plus 1/64 in. and minus 1/32 in. to lengths of over 3 5/16 in.

Partitions Between Compartments.—The thickness

of the partitions between compartments shall be 3/16 in. minimum and ¼ in. maximum.

Overall Width.—The overall width shall not exceed 7½ in.

It is the opinion of the Division that approval of the proposed recommended practice will result in its gradual adoption throughout the industry and effect a simplification of mold equipment. Although the monoblock container is relatively a new development, standardization will do much to prevent needless expense to the hard-rubber manufacturers in meeting unnecessary special demands of the automobile and the battery manufacturers.

It is not considered advisable by the Division to specify the outside dimensions for the containers as this would tend to limit development. It is thought, however, that the dimensions agreed upon will be a good basis for determining the ultimate outside dimensions.

TRANSMISSION DIVISION REPORT

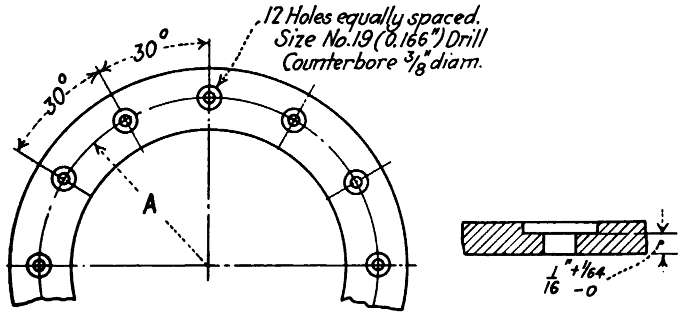
Division Personnel

A. C. Bryan, *Chairman*	Durston Gear Corporation
L. C. Fuller, *Vice-Chairman*	Fuller & Sons Mfg. Co.
A. W. Copland	Detroit Gear & Machine Co.
A. M. Dean	Rubay Co.
D. E. Gamble	Borg & Beck Co.
A. A. Gloetzner	Covert Gear Co., Inc.
C. H. Grill	Foote Bros. Gear & Machine Co.
Joseph Jandesek	Formerly with Olds Motor Works
W. C. Lipe	Brown-Lipe Gear Co.
W. M. Petty	Service Motor Truck Co.
C. E. Swenson	Mechanics Màchine Co.
E. E. Wemp	Long Mfg. Co.
S. O. White	Warner Gear Co.

CLUTCH FACINGS

(*Proposed S.A.E. Recommended Practice*)

In March, 1922 the Society adopted the recommendation of the Transmission Division for clutch facings, the recommended practice now being printed on p. E19 of the S.A.E. HANDBOOK. At a meeting of the Division held in October a report was submitted by A. C. Bryan covering the location of the rivet holes and the size of rivets, this having been submitted to manufacturers and users for comment. Although the Subdivision report at first recommended two concentric rivet-hole circles, it was revised to specify only one rivet-hole circle for each size of facing used for multiple discs, and the total number of circles for all facing sizes was reduced to three, thus necessitating but three sizes of drilling jig.



PROPOSED DIMENSIONS FOR CLUTCH-FACING RIVETS AND RIVET HOLES

Facing Size		A ±1/32
Outside Diameter	Inside Diameter	
8½	6	7 3/16
8½	6¼	7 7/16
8½	6½	7 7/16
8¼	6	7 3/16
8¼	6¼	7 3/16
8¼	6½	7 7/16
8	5¾	6 15/16
8	6	6 15/16
8	6¼	7 3/16

This action was considered to be in accordance with good clutch design as the members of the Division have experienced no trouble with fabric or molded types of clutch facing curling even with wider facings, when the rivets were placed in one rivet-hole circle. The maximum amount of wear was provided for by specifying that the thickness of facing from the bottom of the counterbore to the under side of the facing should be 1/16 in. with tolerances of plus 1/64 in. and minus zero, this being the minimum thickness to give the rivets sufficient hold. Therefore

> The Transmission Division recommends that the present S.A.E. Recommended Practice for clutch facings, p. E19 of the S.A.E. HANDBOOK, be extended to specify the dimensions for locating the rivet holes and the rivet sizes for multiple-disc facings shown in the accompanying table.

The Division plans to give further consideration to the standardization of the rivet locations for single-plate clutch facings.

TRUCK DIVISION REPORT

Division Personnel

F. A. Whitten, *Chairman*	General Motors Truck Co.
A. J. Scaife, *Vice-Chairman*	White Motor Co.
A. K. Brumbaugh	Autocar Co.
G. S. Cawthorne	Master Trucks, Inc.
J. R. Coleman	Selden Truck Corporation
F. W. Davis	Consulting Engineer
H. E. Derr	International Harvester Co.
A. W. Herrington	War Department
M. C. Horine	International Motor Co.
H. B. Knap	Packard Motor Car Co.
W. M. Petty	Service Motor Truck Co.
C. B. Veal	Manly & Veal

VERTICAL DUMPING-HOIST PLATFORMS

(*Proposed S.A.E. Recommended Practice*)

The standardization of vertical dumping-hoist platform mountings was discussed at a recent meeting of the Truck Division, but it was felt that not much can be done in this connection except to indicate in a general way what is considered good practice as to the method of fastening the hoist platform to the truck frame. Therefore

> The Truck Division recommends for adoption as S.A.E. Recommended Practice that in mounting vertical dump-hoist platforms no holes shall be drilled through the frame flanges or in the web near the flanges, but that where the frames are drilled the holes shall be on or near the neutral axis of the frame section.

THREE-JOINT PROPELLER-SHAFTS

(*Proposed Revision of S.A.E. Recommended Practice*)

The present S.A.E. Recommended Practice for Three-Joint Propeller-Shafts, p. E6a of the S.A.E. HANDBOOK, was criticized at the time of its adoption in March, 1922 by several of the Society members to the effect that the square type of shaft-end should not be included in the standard as it is not widely used. This matter was considered at a recent meeting of the Truck Division and, although it was felt that the square shaft-end is not the

best construction for three-joint propeller-shafts, it was decided not to revise the recommended practice unless a general inquiry should substantiate the criticism. A letter was therefore sent out and, as the replies indicated that this type of shaft-end is not used extensively,

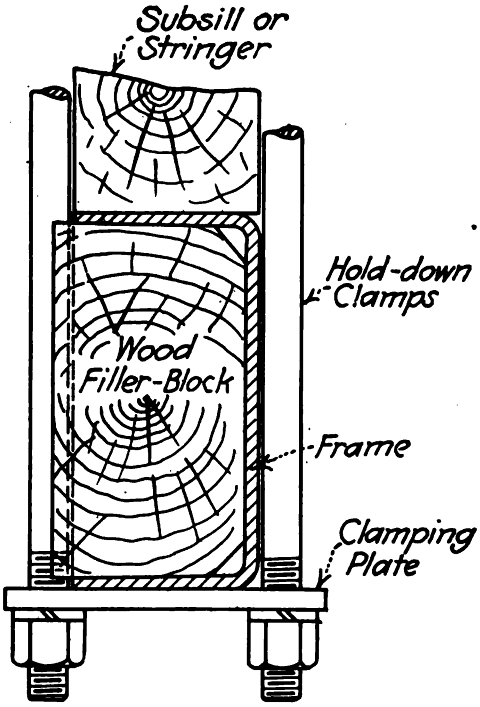
The Truck Division recommends that the present S.A.E. Recommended Practice for Three-Joint Propeller Shafts, p. E6a of the S.A.E. HANDBOOK, be revised by eliminating the square type of shaft-end.

BODY HOLD-DOWN CLAMPS

(Proposed S.A.E. Recommended Practice)

Discussion of body hold-down clamps at the Truck Division meeting in July indicated that standardization of a general type of clamp would lead to improved practice in many instances. Although this subject will probably be of more interest to body builders than to truck builders inasmuch as the former usually make their own clamps, it is felt that a recommendation should be adopted as a guide to the more general use of an effective, inexpensive method of fastening truck bodies to frames. Therefore

The Truck Division recommends for adoption as S.A.E. Recommended Practice that neither the top nor the bottom flange of motor-truck frames shall be drilled for body or hoist platform hold-down clamps, but that U-clamps shall be used with a wood block filler between the frame flanges to prevent bending, in a manner similar to the construction shown in the accompanying illustration; and that the use of too many hold-down clamps for securing the body to the frame be guarded against, particularly in mounting a very stiff body such as those used for oil-tanks.



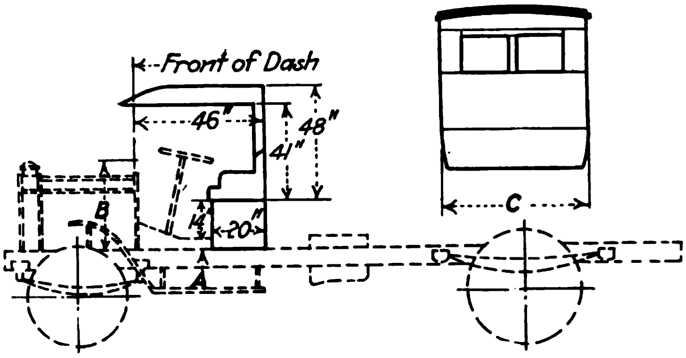
MOTOR-TRUCK CABS

(Proposed S.A.E. Recommended Practice)

Early last spring it was decided to undertake the standardization of motor-truck cab mounting-dimensions to save the loss of time and expense in fitting cabs to truck chassis and to make it possible for truck operators to change a cab from one chassis to another when desired. Information was obtained from truck builders as to the mounting dimensions required for cabs used on their trucks; this was tabulated and analyzed, and a tentative recommendation was based on it. The recommendation was reviewed, together with comments received from motor-truck cab builders, and was modified.

The revised recommendation was referred to motor-truck and cab manufacturers for final comment, the comments received being referred to the members of the Truck Division and final Division action taken by letter ballot. Therefore

The Truck Division recommends for adoption as S.A.E. Recommended Practice the motor-truck cab dimensions given in the accompanying table.



PROPOSED MOTOR-TRUCK CAB DIMENSIONS

Truck Capacity, Tons	A	B[a]	C
1, 1½, 2, 2½	3½	35	48
3, 3½, 4, 5	4	39	56

[a] Minimum for windshield lower edge. Where cab sides, doors or side curtains are used, care should be taken that the driver's vision shall be interfered with as little as possible.

It is not expected that the cab dimensions proposed will permit the installation of stock cabs on all chassis now built. The modus operandi has been to strike an average of the prevailing practice with due regard to the possibility of tempering such practice to meet that of the largest number of users. The recommendation will offer the opportunity.

Detroit Production Dinner

AT the Production Dinner in Detroit in October, which was not only an unusually intimate occasion but also a liberal education on the necessities and possibilities of cooperative methods in production, H. H. Emmons, who was in charge of Liberty-engine production during the war, Past-President Kettering and P. S. du Pont, president of the General Motors Corporation, gave inspiring talks. These are reproduced in large part herewith. A. B. C. Hardy, president and general manager of the Olds Motor Works, was the principal speaker of the evening. His address is printed elsewhere in this issue of THE JOURNAL.

P. S. DU PONT

I AM struck frequently, as we go on with our daily problems in the automobile industry, how discouraged we are over many things that in the future will seem very light. It is because we do not understand our problems; we are fearful, because we do not understand, that somebody else may and get ahead of us. We in the General Motors are fearful of somebody else and somebody else is fearful of us. We are struggling against each other in imagination, but really not in practice. We are all working on a common problem. What one man finds out quickly becomes the property of everyone to expand still further. There is nothing fearful in this industry, any more than in the simple problem of smokeless powder before it was turned out finally as a finished product.

If we work together in the industry the problems will be easily solved. If we are not fearful of the problems, they will be still more easily solved.

We all know that our present cars, our present engines, everything, are very imperfect. I believe all of you could tell me, as I cannot tell you, just how imperfect our operations are. That makes the problem all the more interesting. We are fortunate to be in this industry which is an infant. There is before us now, not only the world of our own industry, with its many ramifications and possibilities of development, but there are thousands of uses of the automobile that we do not picture to-day. We have only scratched the field. This is the only country in the world that uses the automobile, as you know, and yet the bulk of the population is elsewhere with its own problems, peculiar problems, if you will, but all before you to be solved. All of us have a wonderful opportunity because we will all have some years in this industry and the next few years are going to be full of developments of interest for us.

I congratulate the Society on mustering so many men together, and so many brains, on these problems. I wish you all happiness in the development of them and I only hope I may share to a certain extent in your troubles and in your successes.

H. H. EMMONS

I AM very glad to have the privilege of being here tonight and seeing so many of the men with whom I was associated during the 2 years that I look back on with the most pleasure of any in my existence, because the production of aviation engines during the war, under the conditions in which it was started and in the way in which it had to be handled and the tremendous results that were achieved, constitutes, I think, the most remarkable production program that was ever put through in the history of industry. That was due to the men in the factories, the engineers and the production men.

It is a real pleasure to think of the time when that wonderful job was accomplished, when the Liberty engine was struck from the minds of two or three men in the course of 2 or 3 days and was put into production by one of the factories here in Detroit whose production man sits at this table, within 4 months after it had passed its tests, the production reaching within 13 months 15,570 engines complete with spare parts equal to 55 per cent in addition. There were working on the program for the making of 100,993 aviation engines, over 200,000 men and women in the factories and parts plants, an enthusiastic, whole-hearted crew of people, who had but one thought, to get out the most engines, of the best quality, in the shortest possible time. The result was that before the armistice was declared, of all seven types of aviation engine made in this Country, over 33,000 were delivered into service complete with their spare parts, between May, 1917, and November, 1918.

The Liberty aviation engine has never been surpassed for all-round efficiency by any other engine of similar type. It is used to-day by all the services of the Government engaged in aviation activities. The Post Office Department uses it almost exclusively. Some of the races that were held in Detroit recently were won by airplanes that were propelled by the Liberty engine. Other engines of excellent design and workmanship were made here during the war, such as the Hispano-Suiza. The work that was done at that time by the engineers and the production men stands to-day for the benefit and service of the Nation; it is one of the greatest assets the Air Service has.

The main function of every one connected with industry is to help the production man, assist in the production of material. A lawyer may draw a wonderful contract or an engineer make a wonderful design, but neither of them is worth the paper it is written on if it does not help some fellow to produce something that somebody wants and can use. Now that the automotive industry has passed from infancy to manhood, the proposition of engineering and production comes again into the public eye, in the birth of a new baby, namely, commercial aviation material. That is the next thing before us to handle. That material will be needed as badly and as quickly as the other automotive material has been heretofore. Those who have studied the question of aerial navigation with relation to national defense know that it is most important that we get aviation on a basis where civilian airships and airplanes will be constructed so that they will furnish the absolutely essential and fundamental element of national defense, as to both material and personnel. There is no place where engineering and production should be combined so promptly and with such effect as in the development and production of this material.

The time of fighting wars in the old style has gone. A war between two adjoining nations will be decided in a few hours. There will be no time to mobilize a body of troops or handle a fleet of battleships. The nation that will defend itself must have control of its own air. We have an enormous boundary-line, both land and water. To provide in the naval or military service a defense in

the air would cost us an enormous amount of money. If we develop civil aerial navigation with equipment that can be converted for use in war, and have the aviators trained and regulated so that they will be able to handle the dropping of bombs as well as the transportation of passengers and express, we shall have very cheaply and very efficiently the national defense we need. To that end, I hope that the engineer and the production man will take to heart all of the lessons that have been learned in automotive industry.

C. F. KETTERING

WHEN we engineers design a thing, and the production man starts to make it and puts it together, it is never what we designed. Then we have to go back and do the job all over again. There is a certain amount of relief in that, because nobody is responsible for the production then. There has been a good bit of reason for that. Things have had to be produced in a certain length of time, and engineers have proposed certain types of material that could not be obtained. Since we have had modern metallurgy, when a part is not strong enough, all we have to do is get another type of alloy-steel. That serves until the customer gets it. I was in a shop not long ago where they had 57 different kinds of alloy-steel to cure the same trouble. By the time it was cured the article was out of production.

The time for the production engineer to sit in is before the thing is designed. I believe that this industry has spent hundreds of millions of dollars, foolishly, by not calling the production man in when the design was in process. We do many things in engineering work that inflict on the factory problems that have had no thought at all.

Your designing engineer will work up the machine. You may build a model. A thing may function properly. The drawings will be prepared for production. A number of the little details in those production drawings are put in without any brains at all. A draftsman you hired day before yesterday will put in fillets and impossible sets of conditions that the production man cannot carry out. Those drawings will go to your tool-design department and your tool-makers will try to make the tools from the design. You will attempt to manufacture the parts, with the net result that the whole thing is made exceedingly difficult to do, many times, when neither the designing engineer nor the production manager has had a chance to touch the real cause of the trouble.

I believe that we do not need to be overly secret after we get a design worked out so far as the production engineering is concerned. I think that the production men should have a chance to look at the design before it is finally crystallized. No designing engineer should be fastidious about changing a drawing to facilitate production.

We have found that turning wooden models of the pieces under question over to the factory men to look at has paid for the cost of the models more than a hundredfold. Making up a casting in such a way has resulted in suggestions from the pattern-shop and the foundry that have simplified the design, the methods of location and what not, saving thousands of dollars in tooling.

There should be no dissension among production and engineering men. The two closest fellows in any organization should be the production man and the engineer. A third man who has never been there should come into the picture, the accounting man. Cost accounting as we know it to-day is historical; sometimes very historical. Our designing engineers and our production engineers will not get to the real finality of this thing until the question of costs is considered in connection with every line of the design. I do not mean that a design should necessarily be cheap, but that it is the worst type of economy to spend a dollar in the production of any piece that does not render a dollar's service to the customer. If the cost accountant injects fundamental economies into the job when it is being designed, the designer, the production man and he can work out a product that will be better, serve the customer better and make the final amount in the bank greater.

It seems to me that this Production Meeting marks an epoch in the history of the automobile engineer. We should consider more and more the question of the productability of a piece and its cost, before completing designs. Once we do that, we shall proceed into the finality of the job, just as the Society has tried to carry out the standardization of materials.

In the case of any design we should not think of it first in terms of a special machine. A special machine should be considered only after a standard machine has been thoroughly considered. The question of design should carry with it a picture of the floor space, the capital investment and all the other things that enter into the ultimate cost of the article. We have been interested too much in labor and material; and overhead has been a plaster that we put on regardless, without bearing in mind that it is just as great a factor as, and sometimes many times greater than, the labor, and occasionally equal to the labor and the material. We do not always recognize that when we design a piece of apparatus we are automatically laying-out shops, including all the things that are involved in the fabrication.

So, in this twenty-first year of our industry, it seems to me that we should appreciate that the ultimate cost of an article is its cost when it is on the shipping platform and that this cost can never be reduced to its lowest possible minimum until all of the factors that enter into the cost sheet shall have been taken into consideration.

I am certainly glad to have had the opportunity to be at this meeting and see the production men and the engineers together. We have been a loyal bunch of scrappers. Each side has thought the other has been somewhat off color and perhaps both were right. If we get together in all of our work we shall produce a better article for our customers.

We ought to make automobiles that can be repaired when and as they break. We should be able to repair them with some of the tools that are in the tool-kit. The thickness of the instruction-book should be greatly reduced. To-day, the kit of tools and the instruction book are the great engineering alibis of the automotive industry.

It does not cost much to mark on the tool the use to which it is to be put. Many of you have never bought a new automobile. Maybe you have driven some you have designed. If you should buy a car, have something happen to it, and take out and look at the tool-kit, what you would finally do is to take the pliers and fix the trouble.

For a small price you can indicate with a stamp what those tools are to be used for. All such things are a part of engineering. All of them will help to make our boss, the customer, happy and willingly subscribe a certain amount to the payroll. If we will recognize that the future of our industry depends upon how well and

how easily we can put into the hands of the customer the automatic transportation he never had before, we will have solved the great problem of the automotive engineer.

It is not making so many machines. It is not melting up so many tons of cast iron. It is not making so many wonderful designs, putting on various types of things. That is not the question. We must do something else. To-day, we have bumpers on the front and the rear of the car. We have cozy wings, cigar lighters. We have every conceivable thing that you can think of on cars. We cannot get anything more on unless we increase the wheelbase.

The big question is whether that provides economical and desirable transportation. In the degree that you render that desirable transportation by furnishing a minimum amount of material, and a minimum number of places for the car to get out of order, just so far will you have gone in fulfilling the mission for which the Lord has apparently created you.

S.A.E. STANDARD BABBITT SPECIFICATIONS

A SERIOUS error in the publication of the S.A.E. Standard specifications for babbitt, or white bearing metals, in the August 1922 issue of the data sheets has been brought to the attention of the Society. Specifications Nos. 11, 12, 13 and 14 do not cover the maximum amounts of the various impurities that were given in the original babbitt specifications of which they were revisions. It was not the intent of the Non-Ferrous Metals Division to omit specifying the impurities but, as the report of the Division as acted upon at the Standards Committee Meeting in June did not refer to allowable impurities in the specifications, the report was so printed in the S.A.E. HANDBOOK, pp. D103 and D104.

In order that the members of the Society may note in their Handbooks the additional elements covered by these specifications, the complete babbitt specifications are reprinted below, the matter omitted in the August 1922 issue of data sheets being printed in italics.

WHITE BEARING METALS

The limits for the chemical compositions specified for metal in ingot form are closer than the limits specified for cast products, as allowances have been made for variations in the chemical content due to casting.

SPECIFICATION NO. 13, BABBITT

Composition in percentage.

	Cast Products	Ingots
Tin	4.50 to 5.50	4.75 to 5.25
Antimony	9.25 to 10.75	9.75 to 10.25
Lead, max.	86.00	85.50
Copper, max.	0.50	0.50
Arsenic, max.	*0.20*	*0.20*
Zinc and Aluminum	*None*	*None*

General Information.—This is a cheap babbitt and serves successfully where the bearings are large and the service light. It should not be used as a substitute for a babbitt with a high tin-content. It is also suitable for die-castings.

SPECIFICATION NO. 14, BABBITT

Composition in percentage.

	Cast Products	Ingots
Tin	9.25 to 10.75	9.75 to 10.25
Antimony	14.00 to 16.00	14.75 to 15.25
Lead, max.	76.00	75.25
Copper	*0.50*	*0.50*
Arsenic, max.	*0.20*	*0.20*
Zinc and Aluminum	*None*	*None*

General Information.—This is a cheap babbitt and serves successfully where the bearings are large and the service light. It should not be used as a substitute for a babbitt with a high tin-content. It is also suitable for die-castings.

SPECIFICATION NO. 10, BABBITT

Composition in percentage:

	Cast Products	Ingots
Tin, min.	90	90.75
Copper	4 to 5	4.25 to 4.75
Antimony	4 to 5	4.25 to 4.75
Lead, max.	0.35	0.35
Iron, max.	0.08	0.08
Arsenic, max.	0.10	0.10
Bismuth, max.	0.08	0.08
Zinc and Aluminum	None	None

When finished bronze-backed bearings are purchased a maximum of 0.6 per cent lead is permissible in scraped samples provided a lead-tin solder has been used in bonding the bronze and the babbitt.

General Information.—This babbitt is very fluid and may be used for bronze-backed bearings, particularly for thin linings such as are used in aircraft engines. It is also suitable for die-castings.

SPECIFICATION NO. 11, BABBITT

Composition in percentage:

	Cast Products	Ingots
Tin, min.	86.00	87.25
Copper	5.00 to 6.50	5.50 to 6.00
Antimony	6.00 to 7.50	6.50 to 7.00
Lead, max.	*0.35*	*0.35*
Iron, max.	*0.08*	*0.08*
Arsenic, max.	*0.10*	*0.10*
Bismuth, max.	*0.08*	*0.08*
Zinc and Aluminum	*None*	*None*

General Information.—This is a rather hard babbitt which may be used for lining connecting-rod and shaft bearings which are subjected to heavy pressures; its "wiping" tendency is very slight. It is also suitable for die-castings.

SPECIFICATION NO. 12, BABBITT

Composition in percentage:

	Cast Products	Ingots
Antimony	9.50 to 11.50	10.25 to 10.75
Copper	2.25 to 3.75	2.75 to 3.25
Lead, max.	26.00	25.25
Tin, min.	59.50	60.00
Iron, max.	*0.08*	*0.08*
Bismuth, max.	*0.08*	*0.08*
Zinc and Aluminum	*None*	*None*

General Information.—This is a relatively cheap babbitt and is intended for bearings subjected to moderate pressures. It is also suitable for die-castings.

Coming Meetings of the Society

THE ANNUAL MEETING

THE Annual Meeting of the Society will be held in New York City, Jan. 9 to 12. These dates occur during the week of the National Automobile Show and the Automobile Body Show, thus enabling the members to visit these important exhibitions while in the city for the Society meeting. The valuable knowledge that can be gained from the two shows and the representative group of papers at the Society meetings will amply repay each member for the time and expense of attendance. The following paragraphs announce such details of the program and arrangements as are definite at this time. Lay your plans to be in New York City from Jan. 9 to 12.

Reduced Railroad Fares

Negotiations are under way with the Trunk Line Association to secure reduced railroad fare concessions for the benefit of members who attend the Annual Meeting. Members will be required to secure special certificates from the local railroad agent when purchasing tickets to New York City. These certificates must be presented for validation at the Transportation Desk at the Annual Meeting. *Absolutely no certificates will be validated for any person not a member of the Society.* The Society is held accountable under the Interstate Commerce laws for any violation of the agreement limiting the reduced-fare privilege to members and dependent members of their families. Regardless of any assurance given by local railroad officials, there will be no certificates validated at New York City unless they are presented by persons whose membership is evidenced by our records. Complete details of the reduced-fare plan will reach the members in an early issue of the *Meetings Bulletin.*

The Annual Dinner

The Annual Dinner will be held at the Hotel Pennsylvania Thursday evening, Jan. 11. The Dinner is one of the established features of Show Week and is nationally regarded as the outstanding gathering of representative automotive men held during the year. Our genial friend, C. F. Kettering, will serve us again as toastmaster. There will be one formal after-dinner address and past experience has demonstrated that the speaker will be selected because his message is of direct interest to the men in the industry. Tickets for the Dinner should be ordered at once using the form printed at the bottom of this page. Preference as to location of tables will be accorded applications in the order of their receipt.

It has been decided by the Council and Meetings Committee to eliminate the Carnival of past years from the 1923 program. This is largely due to the fact that, when staged with the proper degree of decoration, novelty and splendor this event can not be made self-supporting financially unless prohibitive rates are charged for admission.

The Technical Sessions

All technical meetings will be held in the Engineering Societies Building at 29 West 39th Street. The morning meetings will start promptly at 9:30, one-half hour earlier than in previous years. The afternoon meetings will start at 2 o'clock and the evening meetings at 8 o'clock.

The Annual Standards Committee Meeting, Tuesday morning and afternoon, Jan. 9, will include many important reports upon which action as to adoption, revision or rejection will be taken. The reports to be considered will be found in this issue of The Journal and should be studied by the members so that pertinent suggestions or criticisms may be presented at the meeting.

Body Engineering

It is probable that two Body Engineering Sessions will be held at the Annual Meeting, on Tuesday and Wednesday afternoons, respectively. L. V. Pulsifer will present a very comprehensive paper on the testing of automobile paints and varnishes including actual demonstrations with laboratory apparatus. George J. Mercer will read a paper on the design and construction of less expensive closed bodies. The standardization of lumber sizes and specifications will be the

S. A. E. ANNUAL DINNER

APPLICATION FOR TICKETS

Society of Automotive Engineers, Inc.
29 West 39th St., New York City.

Order of Receipt ——————

Gentlemen:—

Please mail me tickets for the Annual Dinner at the Hotel Pennsylvania, New York City, Jan. 11, as follows:

.... tickets for members at $6.00.
.... tickets for non-members at $7.00.
I enclose check for $........ and list of guests.

I understand that all dinner seats are reserved; that preference as to location will be accorded applications in the order of their receipt; and that each table seats ten.

I also understand that dinner tickets are not subject to cancellation or refund after Tuesday, Jan. 9.

Signed ....

Please enclose list of those for whom these tickets are intended.

Address ....

....

topic of a paper by F. F. Murray. Other papers are under consideration and will be announced later.

Aeronautic Session

The plan for the Aeronautic Session, Tuesday evening, Jan. 9, is rather unique. Five or six outstanding authorities on commercial aircraft are being invited to present brief opinions on certain fundamental design problems whose solution is considered necessary before commercial aviation can progress. These opinions will be thoroughly discussed and debated with the hope that the resultant interchange of thought may clarify the present dilemma as to what constitutes a truly practicable and profitable commercial airplane. Names of the contributors to this symposium will be announced later.

The meeting on Wednesday morning, Jan. 10, will include the annual reports of the Treasurer and of the Sections, Meetings and Membership committees. Ballots for the election of officers for 1923 will be counted and the result announced. President Bachman will present his annual presidential address at this session.

Fuel and Engine Papers

Prof. G. A. Young will present a paper on Practical Methods of Securing High Compression Without Detonation at a technical session to be held Wednesday afternoon, Jan. 10. This paper brings out some interesting results of research work conducted in the laboratories of Purdue University. This will be followed by a paper by Thomas Midgley, Jr., outlining the Fundamental Laws Governing Detonation. A third paper on a Means of Measuring Detonation and Comparing Fuels for Use in High-Compression Engines will be presented by Stanwood W. Sparrow and S. M. Lee, of the Bureau of Standards research staff. It will be noted that this group of three papers is concentrated on the detonation phase of the automotive fuel problem.

Robert E. Wilson will present the first paper at another meeting devoted to fuels on Thursday afternoon, Jan. 11. His paper will discuss the Function of Oil and Fuel in Crankcase Dilution. C. S. Kegerreis will read a paper on the Carburetion of Gasoline and Kerosene. It will outline metering characteristics and the temperatures necessary for good performance; data on air and fuel flow will be supplied and specifications for an ideal carbureter presented.

Cooperative Fuel Research Session

One entire session on Thursday morning, Jan. 11, will be devoted to reports and discussion of the progress made in the Cooperative Fuel Research project that was formulated by the Research Department of the Society. The expense of this fuel-volatility research has been borne jointly by the National Automobile Chamber of Commerce and the American Petroleum Institute representing the automobile and petroleum industries, respectively. The research work has been done largely at the Bureau of Standards in the city of Washington although many other agencies have been cooperating in the tests. Dr. H. C. Dickinson, manager of the Society's Research Department, W. S. James and R. E. Carlson, of the Bureau of Standards, and H. M. Crane, chairman of the Research Committee, will address this meeting. The discussion of the conclusions reached as a result of the tests is expected to be most valuable and complete.

Air-Cooled Engines

Two simultaneous sessions are scheduled for Friday morning, Jan. 12, with two papers in each.. A very complete paper on Air-Cooling of Passenger Car Engines will be presented by S. D. Heron of the McCook Field engineering staff. Mr. Heron has had very broad experience in the development of satisfactory air-cooled airplane engines both in England and this Country. He will set forth certain design recommendations from his experience that are applicable to passenger-car engine design. The paper will carry appendices by C. F. Taylor on the testing of air-cooled engines and by E. H. Dix, Jr., on the foundry and metallurgical problems of the air-cooled cylinder. Prof. E. H. Lockwood will read the other paper in this session and will present data and empirical formulas on the Cooling Capacity of Automobile Radiators. Thus the two opposing methods of engine cooling will be touched upon in this session.

Herbert Chase has written a comprehensive paper on Steering and Steering Gears that follows in plan his commendable paper on Clutches. He will include a very complete analysis of the conditions responsible for front wheel wobble.

A paper on the lubrication of pistons and cylinder walls, by A. Ludlow Clayden, will also be presented at this session.

It should be apparent to any one studying this program of valuable papers that attendance at the forthcoming Annual Meeting will result in material benefit to him. The technical sessions may be likened to post-graduate lectures on automotive engineering designed to keep the engineers in the industry informed of the very latest scientific advances in the art. The standard of papers maintained at the Society's national meetings has resulted in the raising of its programs to the highest plane. Engineering authorities, research laboratories, educators and scientific men look upon these meetings as logical forums for the presentation of their research data. The supply of papers was so great this year that it became necessary to close the Annual Meeting program in September. This reflects a very healthy condition so far as the continuing success of the technical activities of the Society is concerned.

Arrange NOW to be in New York City, Jan. 9 to 12!

THE CHICAGO SERVICE MEETING

THE Chicago Meeting of the Society will be held during Chicago Show Week on Wednesday, Jan. 31. Morning and afternoon technical sessions are being arranged by a local committee headed by B. S. Pfeiffer. The papers and discussion will be confined to vehicle operation and maintenance, particular attention being given to the work of the service-station. The speakers selected will be men either operating large service-stations or responsible for the maintenance of large fleets of vehicles. The program of papers will be announced in the January issue of The Journal.

The Chicago Dinner will be held on the same evening. Here again, automotive service will be the major topic. The arrangement of the Dinner program is vested in a Chicago Committee of which Taliaferro Milton is chairman. Definite plans will be announced later.

It is anticipated that reduced railroad-fare concessions can be secured from points in the automotive territory for the benefit of members attending the Chicago Meeting. Watch for definite announcement regarding this in an early issue of the *Meetings Bulletin.*

COMING SECTIONS MEETINGS

Will Be Found Described in This Issue of THE JOURNAL on p. 564

Research Topics and Suggestions

THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.

SPRINGING AND COMFORT

MUCH has been written of late, both here and abroad, on the subject of springing and riding qualities, but most of the literature passes over one phase of the subject that is certainly of major importance, namely, the physiological effects produced by the action of the vehicle upon the average person. The subject has received some general consideration, to be sure. As early as 1907, F. W. Lanchester made a statement, which may be found in the *Proceedings of the Institution of Automobile Engineers*,[1] to the effect that a period of vibration longer than 90 to 100 oscillations per min. is comfortable, while a shorter period is uncomfortable; he does not state, however, what is the basis for this conclusion. Comfort in railroad travel is discussed in a paper by Georges Marie in the *Revue Generale des Chemins de Fer*[2]; the author of this article, however, deals with the comfort of invalids rather than that of normal passengers. In the *Proceedings of the Institution of Automobile Engineers*,[3] there is a discussion by a number of engineers as to whether it is the acceleration or the rate of change of the acceleration that is felt by the passenger. Nowhere in the literature do we find an attempt to make an analysis of the relation between vehicle performance and riding comfort. There are certainly various features of performance that affect the average person favorably or unfavorably. Probably not all people are sensitive to the same characteristics, or in the same degree. The fact remains, however, that the so-called riding qualities of different cars are remarkably different, and that the degree of fatigue experienced on long trips is very different for different vehicles.

If spring-suspensions and vehicle design in general are to continue to improve as regards comfort, it is important to know definitely what constitutes comfort, and this should be known as regards the average person. We shall confine this discussion entirely to the mechanical treatment that the passenger receives from the vehicle, omitting such questions as quietness, visibility and temperature, as well as driving qualities as distinct from riding qualities.

Most engineering problems are extremely complex, presenting many variable factors. Experimental attack on any problem, however, requires that it be expressed in reasonably simple terms and that the variables be reduced to a very few. The problem in hand is no exception. The possible motions of a vehicle are infinite in amplitude, period and direction. Yet, to be susceptible of experimental study, they must be classified and expressed in reasonably simple terms. We believe this can be accomplished to some extent and some suggestions along this line are given in the following paragraphs. Moreover, the experimenter almost always works on some hypothesis, which, however, he must always remember is only a hypothesis until proved a fact. We shall venture to suggest a few hypotheses.

If a vehicle were traveling at a uniform speed in a straight line, the occupants would be entirely unaffected by the motion; only movements of the vehicle that are departures from uniform straight-line motion produce reactions that can affect him. We may therefore neglect the uniform forward velocity and consider only such motions as are superposed on this. Considered in this way, the chassis and body have six degrees of freedom; they can have translations in each of three directions and can rotate about three axes, within the limits of motion imposed by the springs, the road, and other limiting factors. The only three axes of translation and rotation that can be chosen logically in this instance are the longitudinal axis of the vehicle; a transverse axis at right angles to it, and a vertical axis.

Bearing in mind that uniform motion in a straight line does not have any reaction on the passengers, and therefore considering only those motions of the vehicle that are a departure from this condition, the possible motions are somewhat as follows:

(1) Motions of translation in the direction of the axis, the result of acceleration or deceleration from engine power, brakes or road grades or shocks
(2) Vertical motions due to road-surface irregularities and grades, to which attention is usually confined in considering spring action
(3) Transverse motions mainly due to curves in the line of travel
(4) Rotary motions about the longitudinal axis of the vehicle, due to inequalities in the road from side to side, or to irregular engine-torque
(5) Rotary motions about a transverse axis, or "pitching" due to road-surface irregularities
(6) Rotation about a vertical axis that occurs in driving around curves

So much for the *direction* of transverse and rotary motions. As for the character of these motions, they may be of almost infinite variety, from the gradual acceleration on speeding-up smoothly from a low to a high speed, to the intense shock produced by striking an obstruction in the road, but they have one feature in common; they are all felt by the passenger as *pressures* applied at some part of his body. These pressures are proportional to the accelerations that are imparted to the passenger, but it is probable that the duration of these pressures or the rate at which they change has as much to do with the sensations produced as does the magnitude of the pressures. The duration is certainly of importance because, if the pressure be applied for a very short time as, for example, when the displacement is very small, as when striking a small road irregularity, the passenger is subjected to only a sort of surface effect, while, if the displacement be greater or the pressure applied for a longer time, as when passing over a considerable bump in the road, the motion is felt internally as well as externally.

In the normal operation of vehicles, the kinds of motion that occur in the different degrees of freedom are decidedly different.

(1) Horizontal motions in the direction of driving are of two distinct kinds
  (*a*) Accelerations or decelerations due to engine power or brakes. These motions are of relatively long period and of moderate intensity. The average car has on direct drive an acceleration under one-tenth that of gravity; hence the occupant is seldom subjected to a horizontal force much exceeding one-tenth of his weight in acceleration. In braking, the force in the other direction may be considerably greater, but such violent use

[1] See *Proceedings of the Institution of Automobile Engineers* (1907-8), p. 132.

[2] See *Revue Generale des Chemins de Fer*, May, 1907, p. 249, and June, 1907, p. 367.

[3] See *Proceedings of the Institution of Automobile Engineers*, vol. 7, p. 75.

of the brakes is hardly a common occurrence, or should not be

(b) The other class of motions are those due to inequalities of road surface and resulting component of impact in a horizontal direction, since the effect of striking a small obstruction is to produce only a small change in a relatively high velocity and, since the effect is also combined with a marked vertical motion, the effect of the horizontal component of these forces often has been overlooked. A paper by H. M. Crane, entitled A New System of Spring-Suspension for Automotive Vehicles,[4] discusses this point. If tests were made with a vehicle at rest on a moving bumpy road, doubtless this effect would be more generally recognized. Most spring-suspensions have no cushioning effect whatever in the horizontal direction; it is left to the tires to absorb such shocks. The forces transmitted to the occupants of the car in a horizontal direction would probably be found to be unexpectedly large if adequate measurement of them were made. A few measurements have been made

(2) Vertical displacements are, of course, generally considered the most important. It is these that are considered almost exclusively in spring design. So much has been said and written on this subject that it is necessary here only to call attention to the fact that one class of motions that is usually not included in the discussion of spring action may have an unexpected importance, namely, the very-short-period vibrations of parts of the chassis due to shocks of various kinds. It is said that acceleration as high as four times that of gravity has been measured under conditions where the spring deflection could not account for half this acceleration. Such motions must be of exceedingly small amplitude, but they may have great physiological importance

(3) In a properly designed vehicle transverse forces probably are confined almost entirely to centrifugal action due to curvature of the line of motion, since there are no important transverse forces other than those due to centrifugal force. These forces are relatively small and infrequent in ordinary driving, and we doubt whether their effect on the occupants of the average vehicle is of much importance

(4) The effects of rotary motion about the longitudinal axis of the vehicle are recognized as unpleasant, whether they be due to "rolling" of the chassis or to irregular engine-torque. The former are more or less periodic, with a relatively slow period, and the latter are of such short period as to have, probably, about the same effect as vibrations of very short period and small amplitude in any other direction

(5) The pitching motions of a vehicle, or rotations about a transverse axis, seem to be in a class by themselves. Their period is much more nearly uniform than that of any other type of motion discussed here, and is of the same order of magnitude for all cars. These motions tend to be periodic; in fact, on most roads they seem to be of small importance except in vehicles that tend to have a distinct resonance period in pitching. The physiological effects of this sort of motion may be of considerable importance

(6) Disregarding the centrifugal effects discussed above, the rotations that accompany changing direction impart very small reactions on the occupants of a vehicle and, since these are almost independent of the design of the vehicle, they hardly need further consideration here

Having pointed out some of the types of vehicle reaction that may irritate or tire the passengers in greater or less degree, it is pertinent to ask what can be done about it. In the first place, it seems that we should have a satisfactory means of observing and recording the actual characteristic motions of different vehicles on the usual types of road. This would mean a record of the motions or the accelerations or, better still, of both of them, with reference to all of the six degrees of freedom mentioned above, or at least the first five of them. A number of reasonably successful instruments that have been produced give some of this information, but, so far as we know, none of them gives it all at one time. An ideal instrument for this purpose should be capable of occupying the place of a passenger on the upholstery of a seat, as well as of being rigidly attached to the body of the vehicle. It should be easily transferable from car to car, and should require a minimum of attention.

Having secured a record of the sorts of motion to which a passenger is subjected, the next step would be to find out, if possible, which motions are most irritating and which least irritating or fatiguing to the average person. It is likely that the problem could be attacked after the manner of the experimental psychologist by laboratory methods. A "torture chair" could be designed to reproduce and repeat any type of motion, once we had some suggestion as to what sort of motion to try.

One suggestion along this line comes from the general experience of the biologist and the physiologist. This is that human beings, in common with other animals, usually, if not always, tolerate those conditions to which their race has been subjected during the process of evolution and are likely to be intolerant or very adversely affected by conditions even a little outside the range of racial experience. A suggested application to the question in hand, which may not have the slightest significance, is as follows: vertical accelerations and shocks are natural every-day occurrences in the life of the individual. Every step constitutes a vertical shock along the vertical axis of the body; hence we might expect that vertical motions would not be particularly fatiguing. On the other hand, sudden horizontal accelerations are not by any means common. It is possible, therefore, that these components in the forces to which the occupant of a vehicle is subjected might be proved to be of considerable importance in the fatigue they produce.

So far as the mechanical nature of the forces discussed above is concerned, there appear to be two classes into which they may be divided, although the division is rather arbitrary. They are

(1) Those forces of such short duration or corresponding to motions of such small amplitude that they affect only the surface of the body and do not react on its system as a whole. The effect of these should be reasonably independent of the direction in which they are applied, whether horizontally or vertically, although their effects on different parts of the human body may be very different

(2) Forces of longer duration which have an effect on the entire system, probably depending very much on the direction in which they are applied

Looking at the whole problem in a general way, it appears that the advance which has been made in riding qualities has been very largely a matter of natural selection. There are not even yet any entirely satisfactory principles of design that apply specifically to comfort of riding. It seems that such general principles might be developed through systematic research. The problem is one, the solution of which should appeal to some of our universities and technical-school laboratories in view of the several distinct phases that could be taken up more or less independently and of the opportunities it offers for training in applied mathematics and analytical mechanics.

[4] See THE JOURNAL, June, 1922, p. 463.

WORK OF THE 1922 STANDARDS COMMITTEE

THE acceptance of the reports of the various Divisions of the Standards Committee at the meeting on Jan. 9, 1923 will complete the work of the 1922 Standards Committee of the Society. The personnel of the 1923 Standards Committee will be appointed at a meeting of the Council to be held during the Annual Meeting. As, from year to year, a large majority of the Standards Committee members are reappointed, the 1923 Divisions will be able to carry on the current work of the Divisions without any loss of time or interest.

As at present organized, the Standards Committee consists of 9 automotive and 18 parts and materials Divisions as listed below.

AUTOMOTIVE DIVISIONS

- Aeronautic Division
- Agricultural Power-Equipment Division
- Electric Vehicle Division
- Isolated Electric-Lighting Plant Division
- Motorboat Division
- Motorcycle Division
- Passenger-Car Division
- Stationary-Engine Division
- Truck Division

PARTS AND MATERIALS DIVISIONS

- Axle and Wheels Division
- Ball and Roller Bearings Division
- Chain Division
- Electrical Equipment Division
- Engine Division
- Frames Division
- Iron and Steel Division
- Lighting Division
- Lubricants Division
- Nomenclature Division
- Non-Ferrous Metals Division
- Parts and Fittings Division
- Passenger-Car Body Division
- Radiator Division
- Screw-Threads Division
- Springs Division
- Storage-Battery Division
- Transmission Division

As indicated by their names, the Automotive Divisions represent the various automotive industries and the Parts and Materials Divisions the various parts and materials industries supplying one or more of the automotive industries. The total membership of the 1922 Divisions and Subdivisions is 419, the Division members numbering 293; and the Subdivision members 274, of whom, however, 148 are also members of one or more of the Divisions. Over 90 Subdivisions were appointed during the year to handle the more important subjects before the Divisions.

The work involved in the formulation of the Division reports may well be appreciated from the fact that 46 Division meetings and over 15 Subdivision meetings were held during the year. The number of Subdivision meetings is not definitely known as many are called by the Subdivision chairman instead of by the Society office as in the case of Division meetings.

Although the Division reports submitted at the June meeting of the Standards Committee and the reports appearing on p. 529 of this issue, which will be acted upon at the January meeting of the Standards Committee, represent the completed work of the 1922 Standards Committee, a large number of other subjects have been under consideration during the year. Many of these subjects were being studied prior to the life of the present Standards Committee, their nature being such that it has been found impossible to approve definite recommendations at this time. Other subjects have only been assigned to the different Divisions recently.

The accompanying list of the subjects that have been assigned to the various Divisions of the Standards Committee is therefore given in order that the members may appreciate the large amount of work in progress and ascertain if any of the subjects are of special interest to them. Reports as to the progress on these subjects are published in the monthly articles on current standardization, but the Standards Department stands ready to supply anyone with more detailed information if desired. The majority of subjects listed have been referred to Subdivisions, the chairmen of which will be glad to receive any suggestions or assistance in connection with their work.

Subsequent to the consideration at the Standards Committee meeting of the definite Division reports, printed on p. 529 of this issue, progress reports may be submitted by Division chairmen on some of the more important subjects among those listed hereinafter.

Although 51 subjects have been assigned to the Aeronautic Division, they are not listed herein as conditions in the aeronautic industry have not warranted recent standardization activity.

AGRICULTURAL POWER-EQUIPMENT DIVISION

- Tractor Belts and Pulleys
- Tractor Plowing Speeds
- Tractor Rating
- Tractor Testing Forms

AXLE AND WHEELS DIVISION

- Brake-Drums
- Differential Gears
- Wire-Wheel Spokes

BALL AND ROLLER BEARINGS DIVISION

- Annular Ball Bearings, Wide Type
- Roller Bearings, Inch Type
- Roller Bearings, Metric Type
- Shaft and Housing Fits and Tolerances
- Thrust Ball Bearings, Inch Type

CHAIN DIVISION

- Roller Chains
- Roller-Chain Sprockets
- Roller-Chain Sprocket-Cutters

ELECTRIC VEHICLE DIVISION

- Battery Trays for Electric Trucks
- Charging Plugs and Receptacles
- Lamp Bulbs for Electric Vehicles
- Storage-Battery Tray Terminals

ELECTRICAL EQUIPMENT DIVISION

- Automobile Wiring
- Cable Clips
- Cable Terminals
- Flexible Conduit
- Generator Through-Drive Shafts
- Insulated Cable
- Magnet Wire
- Rubber Bushings

ENGINE DIVISION

- Crankcase Drain-Plugs
- Engine Numbers
- Engine Testing Forms
- Fan-Belts and Pulleys
- Motorcycle Carbureter Flanges
- Poppet Valves
- Starting-Cranks

FRAMES DIVISION

- Motor-Truck Frames
- Passenger-Car Frames

IRON AND STEEL DIVISION

- Cast Iron
- Chemical Compositions
- General Heat-Treatments
- Sheet Steel
- Specification Numbers
- Structural Steel Tubing

LIGHTING DIVISION

- Bases, Sockets and Connectors
- Focusing Mechanisms
- Lamp Glasses
- Motorboat Lighting Equipment
- Tail-Lamp Illumination

LUBRICANTS DIVISION

- Cup Grease
- Transmission Lubricants

MOTORBOAT DIVISION

- Engine Testing Forms
- Exhaust-Manifold Connections
- Control Levers
- Engine Couplings
- Nomenclature
- Tachometer Drives
- Trial Performance Forms
- Stuffing-Boxes

NOMENCLATURE DIVISION

Aeronautics
Highway Transport
Industrial Service
Motorcycling
Power Farming
Stationary-Engine Operation
Water Transport

NON-FERROUS METALS DIVISION

Brass Alloys
Bronze Alloys
Wrought Non-Ferrous Alloys
Specification Numbers

PARTS AND FITTINGS DIVISION

Brake-Lining
Cotter-Pins
Fuel and Lubrication Pipe-Fittings
Felt Specifications
License-Plates and Brackets
Radiator Hose
Wire Mesh

PASSENGER-CAR DIVISION

Brake Tests
Car-Performance Tests
Frame Numbers

PASSENGER-CAR BODY DIVISION

Body Nomenclature
Door Hinges
Nickel-Plating
Upholstery Leather Rolled Sections

SCREW-THREADS DIVISION

Bolts
Gages and Gaging
Rivets
Screws, Bolts and Nuts
Screw-Thread Practice
Tire Valves

SPRINGS DIVISION

Definitions
Leaf-Spring Nomenclature
Motor-Truck Springs
Passenger-Car Springs
Center-Bolts
Spring Specifications
Tests for Parallelism of Spring-Eyes

STATIONARY-ENGINE DIVISION

Connecting-Rods
Crankshafts
Piston-Rings and Grooves
Poppet Valves
Stationary-Engine Belt Speeds

TRANSMISSION DIVISION

Clutch Standardization
Clutch Facings
Transmission Tire-Pump Mounting

TRUCK DIVISION

Motor-Truck Rating
Power Take-Off Shaft-Ends

THE SOCIETY'S STANDARDIZATION PROCEDURE

IT is the function of the Society's Standards Committee to formulate, if feasible, standards and recommended practices that will simplify and coordinate routine and engineering practice on all subjects assigned to it by the Council. To facilitate the work, the Standards Committee is resolved into 27 Divisions, each being representative of a particular branch of the industry.

PERSONNEL

The Standards Committee is presided over by a chairman and two vice-chairmen. Each Division has a chairman and one vice-chairman. The Standards Committee and Division chairmen are designated by the President of the Society, the other personnel of the various Divisions being appointed by the Council. In addition to the Division personnel, Subdivisions are appointed to formulate tentative reports covering important subjects that are under consideration, the Subdivision members being appointed by the Division chairmen. The Subdivision chairmen are usually members of the Divisions to which the Subdivisions report, but other Subdivision members are selected from the industries at large to assure the assistance of the best-qualified men in each particular field.

Members of the Standards Committee need not be necessarily members of the Society, such committee members being known as conferees and having all the privileges of regular committee members except that of voting. In selecting members of the Standards Committee importance is placed on obtaining men of broad experience and so far as possible familiarity with standards work. Many of them naturally are associated with the older and better-known companies, but they are selected for their personal qualifications.

INITIATION AND PROCEDURE

The assignment of subjects to the various Divisions is generally based on requests that have been received from members of the Society or the industry directly affected. Subjects are not assigned by the Council unless their standardization is considered feasible as well as desirable. Upon assignment to the proper Divisions, the various matters are studied with relation to the limiting phases within which standardization can be accomplished and the probable requirements of the industry affected. If a subject is involved, the features that the standard should embody are outlined and an approval of the outline obtained from the companies interested. The industries are then circularized for data representing current practice and suggestions for consideration by the Division members. The information is then, as a rule, referred to a Subdivision for the formulation of a tentative proposal which, when submitted, is referred to the industries for comment. The tentative proposals, together with the comments received, are then referred to the Divisions, and revised to meet all valid objections which may have been made. The Division recommendations are then printed in THE JOURNAL prior to consideration of them at the next meeting of the Standards Committee.

Division reports may be approved only by the Standards Committee at regular meetings held semi-annually or at special meetings called by direction of the Council. These meetings are open to Society members and non-members alike. The reports are discussed and may be approved as submitted or in revised form or referred back to the respective Divisions for further consideration. After Standards Committee approval the reports go to the Council and if approved are acted upon at a regular business meeting of the Society. The reports may be amended at these meetings but are usually approved as submitted. The reports are then submitted to the voting members of the Society for adoption by letter ballot, a majority of the votes cast being necessary to make the recommendations official S.A.E. Standard or Recommended Practices. The results of the letter ballot are referred to the Council, and in case a recommendation encounters several negative votes supported by sound engineering reasons, the Council may withhold its publication in the S.A.E. HANDBOOK until the reasons submitted shall have been reviewed by the Division making the recommendation.

Although the regulations provide that a majority of votes

(*Concluded on p.* 560)

Motor-Vehicle Head-Lamps and Road-Lighting

By R. C. Gowdy[1] and J. G. Balsillie[2]

Illustrated with Photograph and Charts

SINCE headlight lenses and other dispersing devices mainly perform a secondary function in taking the light from a parabolic reflector and redistributing it and because more effective road-lighting is necessary to compensate for the increase in the average speed of driving and in the congestion of traffic, the authors state that a direct solution of the problem lies in modifying the reflector so that the light is distributed properly at its first reflection and lenses or secondary devices are rendered unnecessary.

The consensus of opinion of many motorists regarding the proper degree of illumination is stated and light-distribution experiments are described. Types of reflector and the amount of light-distribution desirable are discussed, an exposition being given also of the corrugated reflector and the merits and demerits of hyperbolic and parabolic basic curvatures. The subject of light concentration is given brief consideration.

WITH the rapid increase in the use of motor vehicles, the providing of adequate and safe road-lighting for night driving has become an acute problem. The acetylene lamp was displaced by electrical equipment because of the inconvenience of using acetylene, rather than on account of any lack in its efficiency, and the later developments have been the result of efforts to improve the utility of the electric devices.

So long as motor vehicles were few in number, the driver was the one principally to be considered, and the parabolic reflector with a bulb of high candlepower met his not too exacting requirements. But when an approaching vehicle ceased to be a rarity, protection from glare became necessary. Various types of shield and visor came into use, but these removed the difficulty only by removing half the amount of light and were not effective unless properly located for the particular focal adjustment.

The increase in the average speed of driving and the congestion of traffic demand some more effective lighting. To do this many forms of lens and other dispersing devices have been developed. Nearly all of these, however, perform a secondary function in taking the light from a parabolic reflector and redistributing it. It is clear that a direct solution of the problem lies in modifying the reflector so that the light is distributed properly at its first reflection and lenses or secondary devices are rendered unnecessary.

The development of the corrugated reflector, which accomplishes the purpose, dates from experiments that were begun in Australia some 5 years ago. The first problem was to find out what was required. The consensus of opinion of many motorists who were consulted showed that, as a general rule, they desired an illumination of the total breadth of the road at a distance of 50 ft. ahead of the vehicle, and that the light be distributed along the road from about 20 to about 150 ft. ahead. A simple sketch of such an illuminated road area shows that the cross-section of the beam of light must be decidedly elliptical, with a much greater horizontal breadth than vertical depth. This shape cannot be obtained from a reflector that causes the light to diverge as from a point, but some surface must be used that causes the light to diverge as if it were coming from a source elongated horizontally; that is, the reflecting surface must be astigmatic.

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Light-Distribution Experiments

The effectiveness of this type of distribution was tried with some unexpected results. The projector used for the experiments consisted of a powerful light-source and two lenses, one spherical and the other cylindrical. With these two lenses a fan-shaped beam of light could be produced and the depth and spread could be varied by adjustment of the lenses relative to the source. The projector was mounted at a height of 4 ft. above a level road, and the apparatus tilted so that no rays were sent above lamp level. The apparatus was adjusted so that the roadway was illuminated for a distance of 150 yd. and, at this distance, for a breadth of 58 yd.

It was thought that, with this arrangement, road-illumination from a motorist's viewpoint would be perfect; but such was not the case. The sharp upper boundary of the beam cut-off the upper portions of pedestrians and vehicles, and the observer's eyes, being unaccustomed to such fragmentary appearances, were unable to recognize the obstructions properly. Another interesting effect was noticed, in that, when pedestrians walked straight away from the projector, distinct visibility for those behind the projector ceased at about 65 yd., although the pedestrian could still see clearly at that distance for an additional distance of 150 yd. When the road-light up to 65 yd. was dimmed by the use of screens, the range of visibility for observers behind the projector was increased to 170 yd. From this it was concluded that the light reflected from the brilliantly illuminated foreground produced a glare which interfered with the vision of an observer stationed behind the projector.

These experiments indicated clearly the general type of distribution that would prove successful, but they also brought out the danger attending the sharp upper cut-off of the beam. Clear resolution for the driver requires that the upper parts of vehicles and pedestrians be illuminated, but such illumination usually means that those approaching the light will be dazzled. A practical solution of this difficulty does not seem possible without elaborate optical devices. The best that can be done is to allow as much light above the lamp level as can be had safely without blinding those who approach.

Many experiments have been carried on, both in this Country and abroad, to determine the intensity of light that produces glare. Estimates by various observers differ widely, but it is significant that observers who permit an abnormally high intensity without experiencing glare are usually found to have astigmatism. When this

error is present in the eye, a point source of light is not focused sharply but is spread into an elongated spot in the direction in which the defect occurs. Conversely, for a normal eye, an astigmatic source or one from which the rays appear to come from an extended region, may not produce glare at a given intensity when a point source of the same apparent intensity does produce this effect.

Types of Reflector

With these experimental facts as a basis, the original corrugated reflectors were made by forming a parabolic reflector with fine vertical stripes, using 12 or more per in. Each of these small stripes was considerably curved in section to produce the necessary spreading of the light. The very large number of beams projected by such a reflector causes a soft diffusion of the light and practically eliminates glare. Reflectors of this type are much in use abroad but, until the present, they have not been available in this Country because of our more strict legal specifications and certain difficulties attending quantity production by American methods. After more than a year of intensive research, development and experimentation, these difficulties have been overcome and the American model of the corrugated reflector, illustrated in Fig. 1, has gone into production.

In adapting the principle of corrugation, two points must receive consideration; first, that the distribution of the projected light comply with the strictest legal requirements in force in any State and, second, that a satisfactory driving light be produced. At present, the specifications adopted by the Illuminating Engineering Society and the Society of Automotive Engineers seem to be at once the most exacting and the most generally adopted for the control of roadway illumination. These specifications stipulate the apparent candlepower that must be provided in certain angular directions in front of the vehicle. Their purpose is to insure adequate light and, at the same time, to prevent excessive glare to approaching machines or pedestrians. It does not follow, however, that a headlight that complies strictly with these regulations will necessarily provide satisfactory road-light or be free from glare, since the desirability of a light depends upon many other considerations than simply its candlepower at a few specified points, and the sensation of glare is not due simply to high apparent candlepower, but also depends upon the illuminated area that is responsible for this apparent intensity. The specifications mentioned are perhaps the best that have yet been produced and, although they probably will be subject to some slight modifications in the future, they provide a fairly satisfactory code for testing lighting devices of widely differing types. It has been our experience, in testing many different designs of reflector and lens, that those which provide a satisfactory road-light do as a rule comply with these specifications, although many devices that do comply strictly with these requirements are not altogether satisfactory.

Desirable Light-Distribution

It has been found that there is a wide difference of opinion as to what distribution of light is most desirable for road-lighting. This difference of opinion arises from three sources, the idiosyncrasies of the driver, his habitual speed and the character of road upon which he generally drives. Some drivers prefer the light near the vehicle and some prefer it at a greater distance, some prefer a very wide spread of illumination, and others are content with a narrower pathway. A light that is satisfactory on a white road will be found entirely inadequate on a dark surface, and any road presents an extremely difficult problem of illumination when it is wet.

A light that is satisfactory for low speeds will be found unsatisfactory at high speeds, and it seems improbable that sufficient light can be provided for safe driving at speeds greater than 30 m.p.h. without the use of such powerful beams that glare is almost sure to result. After many experiments we have found that about 25 deg. of horizontal spread meets with general approval, that more than 20,000 cp. is necessary in the center of the beam and that smoothness of distribution and absence of any sharply defined bright areas are of greater importance. A sufficient length of road for average conditions will be illuminated by a beam that has a vertical depth of 6 to 7 deg. in the center and 3 to 4 deg. toward its sides. For the protection of the on-coming driver and pedestrian, it is essential that the cut-off across the top of the beam be curved downward at the side and, since the driver generally desires the light to come nearer the vehicle in the middle of the road than at the sides, this same contour is most satisfactory for the bottom of the beam also.

As a result, it appears then that the ideal beam should be somewhat elliptical in shape with a gradual increase in its candlepower toward the center. So far as the driver alone is concerned, it would perhaps be more desirable to throw the maximum candlepower near the top of the beam where it would be effective at a greater distance from the vehicle; but such a distribution is dangerous for the on-coming driver, since the pitching of the vehicle will then throw this high intensity above the lamp level and cause blinding flashes in the eyes of those approaching. It seems advisable, therefore, to protect the on-coming driver at some slight expense in distant road-lighting.

Corrugated Reflectors

The development of corrugated reflectors began with the configuration of parabolic surfaces. If a point source be placed at the focus of a simple paraboloid such as is commonly used in headlights, a beam of parallel rays of the diameter of the reflector would be projected. With a source of finite size, such as the filament of a lamp bulb, it is no longer possible to obtain a parallel beam, but the reflector will project a divergent beam whose spread depends on the size of the filament. The filaments in use at present produce a cone of rays of about 4-deg. divergence.

If the paraboloid be corrugated, the light from each corrugation will be spread into a broadened cone, but the spreading will occur only in the plane at right angles to the corrugation; that is, if the corrugation is vertical, the light reflected from it still will have the same vertical depth of beam that would be projected from the corresponding portion of the simple paraboloid, but horizontally the beam might be distributed over a wider angle. Such a beam is astigmatic; the rays that compose it diverge, not from a point, but from an elongated image of the source. All the corrugations on a parabola throw overlapping fan-shaped beams, the centers of which register one over the other.

The amount of horizontal spread that will be produced by a given corrugation depends upon its width, curvature and distance from the source and, if the width be chosen arbitrarily, the distribution of the light projected by the reflector still can be controlled by the curvature of the stripes. The width of the stripe will be chosen for appearance, ease of production and control of diffused light since, in general, the larger the number of stripes is, the greater the amount of light diffused

from them will be. The stripes can be made either concave or convex, as convenience of design or manufacture may demand. If the corrugations are convex, the light reflected from them will be diverging; if concave, the curvature may be chosen to give either continuously diverging beams or beams that first converge and then diverge.

The distribution of light obtained with the concave stripes that give convergent, diverging beams is shown in the photometric chart, reproduced in Fig. 2. This chart shows the apparent candlepower at the various points in the field. The point *A* is at lamp level and directly in front of the vehicle, and the other points are located by their *angular* positions as they appear when looking forward from the vehicle. The contour lines in the left half of the field are lines of equal candlepower of the values 1000, 2000, 5000 and 15,000 cp. respectively. This reflector gives a very intense beam with 15 deg. of horizontal spread. Such a distribution is favored highly by those who are accustomed to drive at high speeds on dark-surfaced roads, but is not particularly desirable for city driving, nor on rough roads where the speed must be

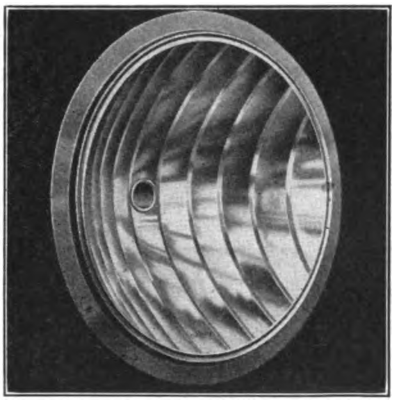

Fig. 1—Type of Corrugated Reflector Produced in America

reduced and the pathway continually shifted. Many other designs embodying stripes of various widths and curvatures were prepared and satisfactory road-lighting was obtained, but it was found that all of these reflectors had very critical focal adjustment and generally insufficient vertical depth of beams. The critical focal adjustment and the lack of vertical depth are inherent defects of the parabolic form as a basic curvature for the reflecting surface. The vertical spread from the parabola is due entirely to the fact that the filament of the lamp has finite size, and the top and bottom of the beam are projected by the central portion of the reflector, while the center of the beam comes from the outer portion of the reflector. Any displacement of the filament affects the outer portions of the beam very much more than it does the center of the beam, since the change in angular position of the filament with respect to the center of the reflector is greater than the corresponding change for the edge of the reflector. Consequently, any displacement of the lamp will greatly affect the cut-off of the beam, and any irregularities near the apex of the reflector will tend to produce undesirable stray rays. To obviate these inherent difficulties of the parabola, the basic curvature of the reflector was changed to the hyperbolic form. The hyperbola permits much greater flexibility of design and affords a greater control of projected light.

Hyperbolic versus Parabolic Reflectors

The hyperbolic surface projects a divergent beam. Its divergence is determined by the major axis, the focal

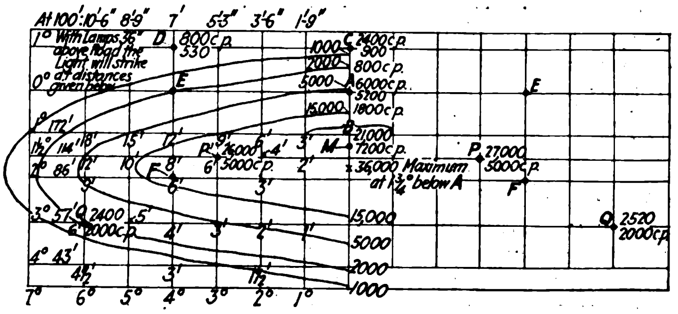

Fig. 2—Photometric Chart Showing the Distribution of Light Obtained with the Concave Stripes

Where Three Sets of Figures Are Given for a Point the Middle One Is That Obtained in the Test While the Upper and Lower Figures Represent Respectively the Maximum and Minimum Called for by the Lighting Specifications. Where Two Figures Are Given the Upper Figure for Points Above the 0-Deg. Line and the Lower Figure for Points Below This Line Indicate the Requirements of the Specifications. The Other Figure in These Cases Represents the Value Obtained in the Test

length chosen and the diameter of the reflector. In the hyperbolic reflector the axial rays are projected by the central portion of the reflector and the more divergent rays by the outer portions in succession to the edge of the reflector. In the parabolic reflector all of the stripes throw the beams that register almost exactly, one over the other, while in the hyperbolic reflector there is a displacement of the beam from each successive stripe to the right and left. This failure of the beams to register causes a patching-up of any irregularities in distribution rather than their accentuation, as in the case of the parabola. The shape of the cut-off from the hyperbolic reflecter is essentially elliptical, but minor alterations in form and in the distribution of the light laterally can be obtained through proper curvature of the stripes. As has been mentioned, the control of the distribution at the top and bottom of the pattern is more readily obtained with the hyperbola than with the parabola, since these regions are affected by the end portions of the stripes rather than the center portions, and these end portions are more readily controlled in manufacture than the center of the reflector.

The hyperbolic forms chosen for the standard sizes of reflector have the same diameters as the parabolas, and are slightly shallower and of somewhat shorter focal length. The focal length is shortened from 0.03 to 0.04 in. to make the hyperbola conform to the standard dimensions of lamp bodies. Curiously enough, these hyperbolas are somewhat similar in shape to parabolas of slightly greater focal-length than the standard parabola. Hence, a solid beam of light will be projected from a source placed anywhere between the actual hyperbolic focus and the focus of a parabola of similar dimensions. This means that the device possesses considerable lati-

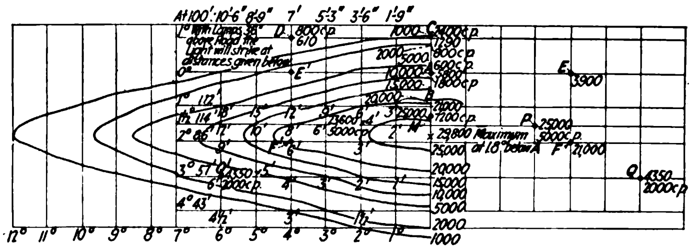

Fig. 3—Chart Giving the Result of Test Made with a Hyperbolic Reflector

Where Three Sets of Figures Are Given for a Point the Middle One Is That Obtained in the Test While the Upper and Lower Figures Represent Respectively the Maximum and Minimum Called for by the Lighting Specifications. Where Two Figures Are Given the Upper Figure for Points above the 0-Deg. Line and the Lower Figure for Points below This Line Indicate the Requirements of the Specifications. The Other Figure in These Cases Represents the Value Obtained in the Test

tude of focal adjustment within which a passabiy satisfactory beam will be projected. There is, of course, a most desirable adjustment with the hyperbola, but failure to produce such an adjustment is not attended by such evidently undesirable results as will be obtained with a parabola out of focus by a similar displacement of the filament. By slightly shifting the source of light forward from the hyperbolic focus, an increased concentration toward the center of the beam is obtained; and, by withdrawing the filament toward the apex of the reflector, the projected beam is increased in depth and larger portions of the light are thrown toward the top and bottom, producing a more uniformly illuminated pattern. Throughout this range of adjustment, the general shape of the cut-off is not altered nor are any undesirable stray rays produced. Thus, within certain limits, the user can produce the type of distribution that best suits him by a slight alteration of the focal adjustment of the lamps, while the essential requirements of legal specifications are fulfilled within these limits of adjustment. A characteristic chart for the hyperbolic reflector is shown in Fig. 3.

Light Concentration

It has been found by experiment that the concentration necessary for the illumination of dark-surfaced highways cannot be obtained readily with a series of stripes of uniform and circular curvature. The production difficulties attending the use of special contours in the formation of the stripes make it inadvisable to depart from the circular sections where curvature is required. The problem of obtaining a sufficient central light has been solved by the introduction of another series of narrower stripes that retain the basic hyperbolic curvature. By breaking up this hyperboloid surface into a number of narrow stripes, any contour surrounding the bright portion in the center of the light pattern is eliminated and a gradual blending from the high intensity in the center to the dimmer edges is thus obtained. Our experiments on the road have shown that an extreme smoothness of distribution and a gradual change in intensity are of even more importance than actual candlepower, and a properly distributed beam will give the impression of much more light on the road than an improperly distributed pattern, even though its maximum candlepower may be very much higher, and any marked contours or patterns thrown on the road always prove distracting when the light is shifted by the jostling of the vehicle.

It is pointed out that the reflector has a decided and inherent advantage over any lens in that the light coming directly from the lamp without reflection is free from the streaked appearance that all lenses produce near the vehicle. This is, of course, of little importance in driving, but is of marked advantage when parking the machine at the curb, because the side light near the vehicle shows uniformly and produces no illusion of irregularities such as may often be occasioned by the directly refracted rays through lenses.

THE SOCIETY'S STANDARDIZATION PROCEDURE

(*Concluded from p.* 556)

shall be necessary to approve a recommendation, a recommendation is seldom approved unless the action is practically unanimous. The reasons for this are patent, as well-founded objections to any adopted recommendation would militate against its general reduction to practice, resulting in a "paper standard" only.

The time required for the stated procedure varies from 3 months to in some cases a number of years, depending upon the conditions involved. A large amount of office work is required in corresponding with the industries and members of the Divisions, arranging meetings and keeping progress records.

S.A.E. Handbook

Subsequent to the Standards Committee meeting, the revisions of the reports together with the discussions thereon are printed in the following issue of The Journal, the revised reports as approved being printed in the next issue of S.A.E. Handbook of data sheets. It is essential that all standards be published in a clear, concise and uniform manner. This was recognized in the beginning by the pioneer members of the Society and its Standards Committee, and the present well-known loose-leaf S.A.E. Handbook has proved the wisdom of selecting this form of publication. Clear detailed drawings with tables and notes are used to set forth the standards and recommended practices. A complete set of the standards goes to all the members of the Society, including the new and revised standards issued twice a year. The complete Handbook is available to nonmembers of the Society and a single copy of any standard may be obtained upon request to the Society. The standards are not copyrighted, but it is expected that the Society will be given due credit for any of its data reprinted in other publications.

The procedure of revising existing standards is the same as when the subjects are originally considered. As such revisions are not published in the S.A.E. Handbook until they have been approved by letter ballot of the voting members and this procedure usually takes several months, it is well for all interested to ascertain whether a standard is under revision in case production on a design involving the use of a S.A.E. Standard is contemplated. Complete information in reference to the current work of the Standards Committee is published regularly in The Journal.

The Production Man's Place in Our Industry

By A. B. C. HARDY[1] DETROIT PRODUCTION DINNER ADDRESS

THERE is more than passing significance in this two-day session, the first National Production Meeting of the Society of Automotive Engineers; that is, the first national production meeting of our great automotive industry. What is more natural than that in this meeting we should discuss the production man?

In getting at him and his place in our industry, I believe it may be not only permissible, but illuminating, to list in their natural order of operation the six main line divisions of automotive organization at the plant.

Given an automobile company, a plant and management, come then:

(1) Engineering: Designing, experimenting
(2) Purchasing: Buying, releasing and follow-up
(3) Production: Manufacturing, inspecting, assembling and testing
(4) Selling: Advertising, sales, service
(5) Traffic: A double service; inbound, to serve between purchase and production; outbound, to serve between the sales department and the company's distributor and dealer customers
(6) Accounting: A multiple service, serving all of the other five departments, as well as the stockholders and the management; estimates; costs, direct and overhead; accounts; credits, collections, disbursements and records of results

These six divisions of work, with a reasonable amount of general management sandwiched in between, make up the average central operating organization. One cannot help but note that five of these six main divisions of the work are almost self-defined by their titles.

Engineering, presenting the design charted in blueprints backed by data and records of tests and opinions of other engineers, paints and sells the picture to the "Old Man," the Committee or the Board, with enthusiastic assurance that it is so real a picture that the public can be depended upon to joyfully furnish the gold to frame it. And if the "Old Man," be he 25 or 50 years of age, buys the picture, we all clearly vision just what purchasing, selling, traffic and accounting proceed to do as their part of the work.

But production? Of course we can say, "Why, that is the manufacturing end of the business," and let it go at that. But it is a broad field and the outlines of its territory, its authority and its responsibilities are not so easily defined. We do note, however, that this production or manufacturing division of the work requires and occupies 80 to 90 per cent of the acres of land and 70 to 80 per cent of the acres of floor-space which the sales and advertising departments boast of, and that it requires and directs the work of 5, 10 or even 15 times the total number of people required by all the other divisions combined. It needs and uses more kinds of people, that is, people in more kinds of trades and training, than all the other five divisions combined.

To head, direct and lead the leaders of the many departments, trades and groups, appears The Production Man, *The Real Manufacturing Man,* whose place in our industry we are discussing. Before attempting to define his status in the industry, let us see what the production man must know, or know about, at least in a general way, to qualify for his job. He must, above all, be a good judge and a just judge of men and a fair and square man himself. He must be strong enough to play no favorites. Surely he must know the general labor conditions in the many subdivisions of his own industry, as well as in kindred or related industries, whether his factory be next door to the factory of its strongest competitor or hundreds of miles from it. He must at all times know the basic rates and changing labor conditions in his own city and the general sentiments of his city. He must know in a general way about day-work, piece-work and group-bonus systems, as he is the first man that we have to sell any new rating-system; he in turn must help sell it to his department-heads and they in turn to their employes. And he must know in a general way the results expected from each one of these systems.

He must have at least general knowledge about plant layout and plant construction. He must know generally much more about machinery, plant equipment, electric equipment, power transmission, patterns, tools, jigs, dies, fixtures and supplies; about ovens and methods of heating them by gas, electricity or oil burning. He must know generally about electricity for power and light, and about air-compressors and air transmission, steam power and heating, steam fitting, plumbing and sanitation.

MATERIALS

The production man must know something about the nature of castings—steel, gray iron and malleable; brass, bronze and aluminum alloys, from foundry into machined, finished and assembled parts; forgings and their machining; heat-treating, hardening, grinding and the like; sheet metals, through deep-drawing to stampings, and including nickeled, painted or enameled finish; woods of the various kinds and their working; fabrics, carpet and leather through their various automobile uses; paints and enamels and their processes and results; sub-assemblies, assemblies, finished units and complete-car assemblies and tests; and inspections, yes, endless inspections, from the receiving doors on through to the finished automobiles.

The production man must respond in his work to the service information from the field. In addition, he must be the type of man who can "steady" and lead in a crisis, when serious accident, a fire or a strike comes. Moreover, he must in some way keep in touch with and in sympathy with welfare work and the living conditions of the employes.

TRAINING OF PRODUCTION MEN

No small order, is it? And I have only touched the high spots. Is there any such man? Or, rather, are there any such men? There are, and more of them are coming up through the same long tedious but fascinating and always-changing courses of the automobile factory

[1] M.S.A.E.—President and general manager, Olds Motor Works, Lansing, Mich.

experience-schools. No college and no school can hope to deliver any such man as a finished product, ready to function. The colleges may be able to give to young men a start on such elements of the science of production work as are definitely recorded at the time the young men are coming through, and it is to be hoped that the schools will be frank enough at the close of their courses to apprise their young men of the truth, the fact that the theoretical or school course has only shown them how to begin to start over again by entry in the regular way past the employment window of the plant into the only real course, a stiff one and a long one, but the most inspiring and fascinating production course. It requires guts as well as brains. The place to look for the coming production man is back along the lines in your plants and in ours, for that is where the production man of to-day came from.

Is this production man, who must know broadly about so many things, a super-man? Not at all; and, fortunately for him and for the industry, he has never been touted as such. And does he do the work or even direct it alone? Not at all; no one accomplishes anything alone in our complicated and fast-moving industry. The fact is that his specially trained assistants and department-heads, from the plant engineer, who keeps the physical plant in sweet running-condition, on through the divisions, to and through those who conduct the final car-test and inspection, after which the product is turned over to the traffic department ready to ship, carry their share of the load and responsibility. And the production man, their leader, if he be a wise man, is always on the lookout in his own plant, and also takes a side-look into other plants, your plants, to seek out the production talent that is being developed, to help him with the work in hand; by going around his job so as to look in at it from the human or employe angle, from the service or customer angle, from the plant and machinery angles, from the material and finished car angles.

Production Man's Status

I wonder if we have not shown pretty clearly the status of the production man. And it is "some status" now. He not only uses most of the land, plant, machinery and equipment but constantly demands more, and that he be allowed to assist in their selection. He influences and controls most largely the hiring and firing, which means the pay rates or incomes and the working and living conditions of far the largest number of people on the job. He and his department-heads are the means of conveying to the workers the standards and character of the company. And the workers judge the company accordingly. He and his department-heads and inspectors affect, almost more than they realize, the future of the company and its success, by the manner in which they build, test and inspect every element in the car and the finished car itself.

As to his importance in the industry, he is not much concerned; for you and I and he know to which department the head of the house, the Committee or the Board naturally turns when there is a mess, a mix-up or a big job at hand. Perhaps the production man has been too backward in making himself known. But he has been too busy on the job. Besides, the spot-light was focused elsewhere, turned first on the engineer or designer with the great idea, then on the promotion engineer, next on the commercial promoter, and then on the star manager or sales campaigner, all of whom made good publicity during the rapid expansion of the industry.

But at length, while the production man was being ground and polished through the mill, the real bosses of all our jobs, the motor-using public, began to control the spot-light and to throw it no longer upon individuals nor upon any one or few departments, but upon what we all as organizations did to them; upon our product, our car; and upon the organization behind that car, as a whole. Our bosses, who have found out that there are few mysteries, and that personal names and reputations perform no miracles and make no cures, will hold this spot-light mercilessly upon us all, not as individuals, but as organizations, and upon our product.

If the question, Production versus Design, as to importance to our industry, were put up to us, I for one should have to vote for production, because I have seen some poor designing turned into fairly good cars by production work and refining, and because those companies and cars that have had the best production or manufacturing attention are strongest and safest to-day. I may be prejudiced, as I have lived so many years with production, buying, selling, and with the men on the job.

But why raise such a question? It is too late. The public, our friends and customers, is not interested in it. The public has come to realize that the individual and independent transportation units which we together as organizations design, produce, sell and service are the things most needed by them after food, clothing, fuel and housing; even more than the telephone. You cannot imagine any individual family or corporation that needs and uses the telephone, getting along without the automobile, but you know of many who must have and do use automobiles and get along some way without the telephone.

Cooperation in Production

Through the medium of the Society, the designing engineers have been able for years to cooperate extensively. This cooperation, augmented ten-fold by the necessities and pressure of the war, standardized many materials, parts and practices, thereby simplifying construction and making vast savings for the industry, its material suppliers and the car-owners. Production engineers, unfortunately, have been denied the opportunity of cooperating with each other to any extent, because only the largest companies felt justified in training and maintaining all-round men; most of the production engineers of to-day are trained only in very limited and specialized fields. In addition, the automotive industry had been so grievously disappointed by the work of the early so-called efficiency and economy experts and groups, brought in from other industries, that it has not looked with general favor upon production engineering until recently. However, dodging the word efficiency, which was in disrepute, practically every plant of any size now maintains a reasonable amount of production engineering as a Standards or Planning Department or Production Office. It is only to-day, when its commercial age is fully 21 years, that our own great automotive industry which, with the related industries producing tires, parts and accessories, constitutes the Individual Transportation Industry, the largest combined industry in the world, is holding with you its first national production meeting. So, our production engineers can take heart with renewed patience and, if the Society has not already done so, should not its logical expansion open a Section for Production Engineers and possibly for Plant Engineers?

With cooperation in production engineering as now developed we believe that our designing engineers are considering more carefully the present and standard plant-machinery equipment in designing new product.

They must know that any new product, while in the paper and experimental stages, must be sold to the production department, to reflex the experience of the production man and his department-heads as to the makableness of the product; and sold to the service department as to the accessibility and serviceability in the field. Some of us have drawn engineering and production into closer and constant cooperation by a combined engineering and manufacturing committee, consisting of general manager, assistant general manager, chief engineer and manufacturing manager; and by calling into their meetings from time to time the sales or service manager or field-service engineer or manufacturing department-heads we may hope to stop the "passing of the buck," and to make our organization so closely woven a production fabric that *we shall all become real production men on the one Job.*

We know that this way lies permanence and success.

THE CENTRIFUGAL PROCESS OF CASTING IRON

IN a digest[1] of a paper on Centrifugal Castings, presented before the West of Scotland Iron and Steel Institute by F. E. Hurst, new evidences of progress in this method are indicated and these are also commented upon editorially.[2] The process, as at present operated in the Scottish plant with which the author of the paper is connected, is being applied to cast iron for producing large-size castings for piston-ring sleeves; gas, oil and Diesel-engine-cylinder liners; chilled wheel and chilled-roll castings; and cylindrical castings of all descriptions. The machines at present in operation are capable of producing castings up to a maximum length of 36 in. and of varying diameters from 10 to 30 in., and other machines for producing both smaller and larger castings are under construction.

The principle of operation of the centrifugal casting process lies in the introduction of molten metal to the mold or die while it is rotating rapidly about a horizontal axis. No core is used for cylindrical articles and a perfectly cylindrical interior surface is produced directly because of the centrifugal force exerted while the molten metal is solidifying and due also to the rate of introduction of the metal to the mold.

The centrifugal casting machine consists essentially of a faceplate mounted on a shaft carried in bearings arranged to be rotated at the speeds required. The molds are attached to the faceplate by bolts and, when rotating, the molten metal is introduced by tilting a specially designed pourer that has been moved into position inside the rotating mold. The molds themselves are constructed in two parts: An outer holding casting that is arranged to bolt to the faceplate, and an inner liner that is arranged to fit loosely inside the holder casting. The liner is bored internally to the size and dimensions of the outer surface of the casting to be produced, and the outside of the liner is bored to fit the holder casting, which is designed to take a group of a series of liners, the dimensions of which are arranged to produce flanged cylindrical castings of a standard series of dimensions. By this means the cost of the renewals of dies for the production of a given size is reduced considerably and, as a rule, the series of holders will take most of the special liners that are required for the production of castings of special shapes.

[1] See *The Iron Age,* Aug. 31, 1922, p. 529.
[2] See *The Iron Age,* Sept. 14, 1922, p. 666.

The back end of the liner next to the faceplate of the machine is closed by a plate attached to the end of a threaded rod passing through the hollow shaft of the machine and arranged for the ejection of the castings. The front of the liner is closed by an annular plate having the internal diameter of the annular ring corresponding to the internal diameter of the casting to be produced. This plate is arranged so as to be removable after the completion of the casting operation and allow the casting to be extracted.

As indicated, a series of standard dimensions of flanged cylindrical castings has been compiled, and the installation of the necessary holders and liners to produce these makes it feasible to manufacture a wide variety of cylindrical castings within the limits of dimensions specified and with alterations that are comparatively trifling. For example, the only alteration to the die required to produce castings thicker than a given standard is an alteration to the diameter of the inner circle of the closing plate. Castings shorter than the standard range of size can be produced by an alteration in the position of the ejector plate. Castings having specially shaped flanges or external projections require a special liner, but in the majority of instances a special liner only is required and the expense is not necessarily prohibitive. Castings having internal projections, or closed cylinders, or dish-shaped castings, cannot be produced on a commercial basis as yet.

Observation of the mechanical properties of centrifugal castings should always be made in conjunction with the chemical composition, and this must be taken into consideration when making comparisons between centrifugal and sand castings. So far as the investigation of the properties of centrifugal cast-iron has gone, in all cases a distinct improvement in the mechanical properties has been found. An improvement would be expected on account of the use of a metal mold, although there is no doubt that the centrifugal method itself has considerable influence in modifying the properties of the cast-iron.

In the plant mentioned an improved melting-furnace has been installed to supply hot metal of uniform composition, which is of prime importance in producing high-grade castings from low total-carbon mixtures and steel mixtures.

In the editorial comment already mentioned[2] it is stated that it seems evident that there are great possibilities in the application of the centrifugal process to small castings of suitable section.

Activities of the Sections

Schedule of Sections Meetings

DECEMBER

6—MINNEAPOLIS SECTION—Design Considerations in Pattern and Foundry Practice—R. C. Hitchcock
8—WASHINGTON SECTION—Piston Rings—A. W. Morton.
14—INDIANA SECTION—What Engineering Owes to Pure Science—John H. Hunt
14—METROPOLITAN SECTION—Air-Cooling of Automotive Engines—C. P. Grimes
15—BUFFALO SECTION—Results of Testing of Materials and Their Effects upon Present-Day Engineering—Prof. G. B. Upton
15—CLEVELAND SECTION—Troubles Encountered in Car Operation—A. J. Killius
19—DAYTON SECTION—The Utilization of Low-Grade Fuels in Automotive Engineering—P. S. Tice
20—MID-WEST SECTION (At Milwaukee)—100-Ton-Miles per Gallon—H. L. Horning and J. B. Fisher
22—NEW ENGLAND SECTION—Mechanical Service—Knox Brown
22—DETROIT SECTION—Body Engineering Meeting
28—PENNSYLVANIA SECTION—Aeronautics—Com. R. D. Weyerbacher, U. S. N.

THE Sections of the Society were very active during November, a total of 10 meetings being held. Attendance figures reaching the New York office reflect an increased interest in Sections meetings. It is also noteworthy that membership in the Sections shows an increase over that at the same season last year. The character of papers presented to date and the valuable discussions that have followed them are evidence of an advance in Sections activity. Read the following reports carefully and join the Section nearest you so that you may enjoy its future meetings.

CLEVELAND SECTION

Close to a hundred members of the Cleveland Section spent a very enjoyable evening on Nov. 17 listening to Dr. Georg Madelung's presentation of the analytical study upon which the design of the famous Hannover glider was based. This glider or sail-plane remained in the air over 3 hr., attained an altitude of 1000 ft. above its starting point and won all prizes in the Rhoen competition in Germany. Dr. Madelung stated that this fine performance was due to the application of fundamental aeronautic theory to the design of the glider so as to attain a minimum rate of descent. This enabled it to fly the longest possible time without losing altitude. Each of the factors in the formula determining the rate of descent was analyzed and by proper proportioning of the machine all of the variables were reduced to a minimum. This study indicated the importance of utilizing a long wing-span in combination with a short wing-chord or width. This was the outstanding feature of the machine from an aerodynamic standpoint. It had a span exceeding 40 ft. and the chord was approximately 4 ft. The weight was only 430 lb., including the pilot, yet the plane possessed great structural strength and rigidity.

It was evident to those hearing Dr. Madelung that the glider or sail-plane competitions have a distinctly valuable place in the field of aeronautics. The Rhoen meeting enabled the German engineers to test and prove the merit of several radical departures from conventional practice at remarkably low costs. Notable among these innovations was a landing gear consisting of three football wheels mounted on rigid axles; a monoplane wing with an unusually long span in combination with a short chord; a special thick-wing construction with only one wing-beam; and a plywood leading-edge construction designed to carry all of the torsional load on the wing. Dr. Madelung's paper will appear in an early issue of THE JOURNAL. It will convince every thinking engineer that glider experiments will exert a strong influence on the aerodynamical features of the commercial airplane of the future.

The next meeting of the Cleveland Section will be devoted to a discussion of the troubles encountered by the average automobile owner in the operation of his car. The speaker, A. J. Killius, is service manager of the Cleveland Automobile Club. He has been making a special study of cars left for repairs at garages in Cleveland and plans to present a frank but constructive criticism of present-day automobiles from the service viewpoint. The meeting is scheduled for Dec. 15 at the Cleveland Engineering Society rooms, Hotel Winton, starting at 8 o'clock.

DETROIT SECTION

What Mechanical Men Should Know About Merchandising formed the subject of a very interesting talk given by E. S. Jordan before the Detroit Section, Nov. 24. Mr. Jordan, who is president of the Jordan Motor Car Co., made some very pertinent suggestions on the importance of engineers studying the demands of the automobile-buying public and creating features that will have a direct sales appeal. He urged a closer contact between the engineer and the salesman so that each might profit by the other's suggestions.

The next meeting of the Detroit Section is scheduled for the evening of Dec. 22 at the Detroit Board of Commerce.

MINNEAPOLIS SECTION

Piston-rings and pistons were the subject of two papers and a very thorough discussion at the meeting of the Minneapolis Section on Nov. 1. Allen W. Morton presented a well-prepared paper on piston-rings. He stated that multiple-piece rings are losing favor rapidly because of their greater liability to breakage, poor wearing quality in service and difficulty of installation. Individually cast rings are superior to pot-cast rings because of their uniformity, closer grained metallic structure, longer life and greater resiliency. Foundry practice of molding, melting and pouring is of greater importance than the chemical composition of the iron. Mr. Morton discussed methods of producing tension in rings and favored the practice that effects the result by artificial means such as peening, believing this assures better control of the final pressures on the cylinder-wall. He described a method of determining cylinder-wall pressures of rings by lapping a sample ring to produce rapid wear and then measuring the amount of ring wear at numerous points on the periphery. The importance of ring fit in the piston groove was stressed as a means of preventing oil-pumping. The discussion of the paper related to analysis of piston-ring iron, sticking of rings and ring thickness.

The second paper of the evening was read by George Bonthenon and set forth some practices followed by the author

in the manufacture of light-weight cast-iron pistons. Iron analysis, heat-treatment and machining operation data were given.

The next Minneapolis Section meeting will be held, Dec. 6, at the Builders' Exchange in that city. Motion pictures illustrating steel manufacture will be shown. Prof. Peter Christensen of the University of Minnesota, will give a descriptive talk on steel manufacture and R. C. Hitchcock will read a paper on Design Considerations in Pattern and Foundry Practice.

Metropolitan Section

About 50 members of the Metropolitan Section who attended the monthly meeting held on Nov. 14 at the Automobile Club of America, New York City, were entertained by a very comprehensive and instructive paper by Alden L. McMurtry, recently consulting engineer of the Connecticut State Motor Vehicle Commission, on the Regulations Governing the Use of Rural Highways.

The methods of investigating both the records of applicants for licenses and the causes of accidents were described in detail and were illustrated by numerous lantern slides. It was the speaker's contention that by penalizing offenders and by gradually weeding out the careless and irresponsible drivers good drivers would be protected and that speed limits, which are now the basis of legal restrictions, would be unnecessary. As the conditions under which a car is operated and the skill of the driver are usually the determining factors in accidents, a light passenger car traveling at a speed of 20 m.p.h. might be as dangerous as a heavier and better built vehicle at twice that speed. The importance of regulating the loading of trucks and of preventing overloading was emphasized.

David Beecroft in the discussion that followed said that highways were not built originally for present methods of transportation and were inadequate; that accidents for which the highways, rather than the drivers, were responsible would increase as the years go by; that lighting must be standardized throughout the Country, so that the same signal would not be indicated by two or three different colors in various localities; that at present there is too much light on passenger cars and not enough on trucks; that the highway lighting in vogue 40 years ago is still in use and is inefficient; and that the tendency of road hogs to hug the center of the road is due largely to the faulty illumination of the sides of the road. On account of the length of the paper very little time remained for discussion and members who wished to participate were requested to submit their comments in writing.

The report of the secretary showed that 51 new members had been admitted to the Section, making the present enrollment 254.

Air-Cooling of Automotive Engines will be treated in a paper by C. P. Grimes before the Metropolitan Section meeting, Dec. 14. Mr. Grimes is research engineer of the H. H. Franklin Mfg. Co. and consequently he speaks with authority on this subject. Metropolitan Section meetings are held at the Automobile Club of America, New York City, starting at 8 o'clock, and are preceded by an informal dinner at 6.30 p.m.

New England Section

The New England section held its November meeting in Springfield, Mass., on the 24th of the month. The members were the guests of the Westinghouse Electric & Mfg. Co. during the afternoon, visiting the Springfield plant of this company and viewing the methods of manufacturing automobile starting and lighting systems. The factory visit was followed by a meeting and dinner at the Hotel Kimball in the evening.

Automotive electrical service was the topic of a paper presented at the meeting by M. B. Speer, superintendent of the service department, Westinghouse Electric & Mfg. Co. He stated that many electrical service-stations are failing to impress each new customer with their ability and that automotive electrical service is a highly specialized branch of the automotive industry which cannot render service promiscuously. Most electrical service-stations make no attempt to analyze the average cost of a particular operation to determine whether that cost could be reduced by eliminating waste time and using special tools or equipment. A majority of stations accept a certain amount of idle time as a necessary evil and 40 to 50 per cent of the electrical repairman's time is unproductive for this reason. Mr. Speer cited his personal experience in managing and operating an electrical service-station to prove that idle time can be almost entirely eliminated by a simple method of time-keeping, a definite knowledge of the time required on each repair operation and by rendering service descriminately but impartially. Electrical service-stations have not paused often enough to ask themselves what constitutes a reliable service-station; cost and accounting systems have been neglected; shop and small-tool equipment has been inadequate; little has been done to increase trade. Immediate service is of no advantage if the service rendered is inefficient and results in a comeback.

The use of armatures rewound by specialty rewinding companies has proved unprofitable for many service-stations, according to Mr. Speer's experience. This is due to the unsatisfactory service such armatures have given. A recent investigation showed that 81 out of the 87 stations consulted, used rewound armatures in their service work. The average charge made to the customer is estimated at 75 per cent of that of a new armature, showing that the ultimate user pays dearly for a repaired piece of apparatus when a new armature would be a better and safer investment.

The New England Section meets in Boston, Dec. 22, at the Engineers Club when a paper on mechanical service will be read by Knox Brown of the Packard Boston service-station.

Washington Section

W. S. James, of the Bureau of Standards staff, has been appointed chairman of the committee that will be responsible for future meetings of the Washington Section. Mr. James is fortunate in having a number of automotive engineering authorities and scientists associated with him at the Bureau of Standards and this assures the presentation of many valuable papers before the Washington Section. The next meeting of the Section will be held at the Cosmos Club, Friday, Dec. 8.

Buffalo Section

Air-cooled automotive engines were the center of interest at the Buffalo Section meeting on Nov. 17 when C. P. Grimes, of the H. H. Franklin Mfg. Co., presented a very valuable paper. The meeting was most enthusiastic and the attendance exceeded 100 persons. Mr. Grimes outlined the major advantages of air-cooling as applied to passenger-car engines. He presented many data on performance, weight and efficiency, stating that air-cooling enabled the Franklin engineers to produce an engine weighing only 158 lb. with complete cooling equipment as compared with 213 lb. for a water-cooled engine of similar power characteristics. It was explained that Franklin economy of operation is dependent on the air-cooling principle to a large extent, not only because of the reduced weight but also because the engine operates at temperatures conducive to efficient fuel combustion. The fuel-consumption at 20 m.p.h. averaged 1.8 lb. per hp-hr. which compares with from 2.0 to 3.5 lb. per hp-hr. in other cars. At higher speeds the consumption drops to 0.7 lb. per. hp-hr. The author argued against providing powerplants in passenger cars that are capable of propelling the vehicle at speeds in excess of 50 m.p.h. when the average owner seldom exceeds 35 m.p.h. in his daily driving. Over-powering results in low fuel economy because of excessive weight throughout the entire chassis and because the engine is operated under a nearly closed throttle most of the time. Questions of blower design, piston fits, cylinder temperature, and blower noise were treated in the discussion following the paper.

The Buffalo Section has scheduled its next meeting on Dec. 15 at the Buffalo Engineers Club. Prof. G. B. Upton,

of Cornell University, will read what is expected to be a very instructive paper on Results of Testing Materials and Their Effects on Present-Day Engineering.

Dayton Section

The Dayton Section did not meet in November but has arranged what promises to be a very valuable session on Dec. 19 when P. S. Tice of the Stewart-Warner corporation will read a paper on the Utilization of Low-Grade Fuels in Automotive Engines. Dayton Section meetings are held at the Dayton Engineers Club and start at 8 o'clock. They are preceded by an informal get-together dinner at 6.30 p.m.

Pennsylvania Section

The design of passenger-car chassis springs received the attention of the Pennsylvania Section at its meeting, Nov. 24. H. B. Winchell, of William & Harvey Rowland, Inc., read the paper of the evening. He discussed spring action and flexibility, the effect of the Hotchkiss drive on riding-quality, natural periods of synchronism and methods of predicting probable spring action on completed vehicles. The paper includes passages on the effects of length, width, number and thickness of leaves, rebound leaves, rolled and diamond points and other details of spring construction. The various types of suspension were compared and their relative merits shown. Mr. Winchell strongly emphasized that it is better to give proper study to spring length, width and characteristics in the early stages of design than to find when the vehicle is completed that it is impossible to install proper type of springs in the space that has been provided for them by the designer.

The next meeting of the Pennsylvania Section will feature aeronautics and the principal speaker will be Com. R. D. Weyerbacher, U. S. N., who is stationed at Lakehurst, N. J. The meeting will be held, Dec. 28 at the Philadelphia Engineers Club, starting at 8 o'clock and will be preceded by the customary informal dinner at 6.30 p.m.

Indiana Section

The Indiana Section meeting on Nov. 9 was addressed by Fred E. Moskovics, vice-president of the Nordyke & Marmon Co., on the relation of the engineer to the sales and service branches of the industry. He divided motor-car engineering into two phases, theoretical or technical engineering and empirical or practical engineering. He felt it the duty of the service engineer to digest the troubles of the automobile operator so that they might be presented to the designing engineer as the basis of his empirical engineering work. The service department should function as the eyes and ears of the engineer in the operating field. Chief executives should organize channels for the assured consideration of the serviceman's suggestions, experience and complaints. The discussion of Mr. Moskovic's paper was very spirited and included remarks by Lon R. Smith, O. C. Berry, Fred Duesenberg, A. L. Nelson, David Landau and others.

J. H. Hunt, head of the electrical division of the General Motors Research Corporation, will address the next meeting of the Indiana Section on Dec. 14. His topic will be What Engineering Owes to Pure Science.

D. C. Teetor has been appointed treasurer of the Indiana Section to succeed Mark Smith who resigned because of his departure from Indianapolis.

Midwest Section

Dr. H. I. Schlesinger, professor of Chemistry at the University of Chicago, addressed the Midwest Section on the occasion of its meeting, Nov. 24. He explained the function of a catalyst in chemical reactions and stated the chemist's theory of catalytic action so that the automotive engineer could appreciate better the important part this scientific discovery has played in the refining and combustion of petroleum products.

The next meeting of the Midwest Section will be held in Milwaukee at the Milwaukee Athletic Club on Dec. 20. H. L. Horning will present the principal paper.

HIGH TEMPERATURE CAUSES EXCESSIVE VALVE-SEAT WEAR

THE Bureau of Standards was requested recently to investigate the cause of excessive valve-seat wear shown in a certain make of engine. These valve-seats were in some instances sunk into the cylinder block to a depth of nearly ¼ in. In every case the trouble was found in connection with the exhaust-valve seat, but it only occurred in a small number of engines of a given type, and even in the same engine all the exhaust-valve seats were not equally affected.

The cast iron of which the cylinder blocks were constructed was first examined, but this proved to be typical of the material usually employed in cylinder construction, and no difference was noted between the metal of the valve-seats showing excessive wear and those showing a normal amount. Next it was thought that difficulty might have been encountered in regrinding the valves, some of the valves not being affected by the grinding compound. This would result in the seat being ground away in a vain attempt to get satisfactory tightness. A few experiments were sufficient to eliminate this as a cause of the trouble.

Attention was next directed to conditions of operation that might account for the trouble. To determine the effect of cam outline and spring pressure, a cam that would give the valve about twice its normal lift and permit it to close very rapidly, thus striking the valve seat with considerable force was designed; but operation with this arrangement failed to show an abnormal rate of wear. However, with the valve and the seat heated to a dull red color, conditions were changed and the rate of wear was amply rapid to explain the trouble.

It should be mentioned in this connection that a reducing flame was used in these experiments to avoid any possibility of rapid wear through the oxidation of the metal. The temperatures reached in this experiment might result in actual operation from preignition or a very slow burning mixture, and it is probable that one of these conditions coupled with inadequate cooling of the valve-seats was responsible for the abnormal wear of the valve-seats in the engines under investigation.

Publications of Interest *to* S. A. E. Members

In this column are given brief items regarding technical books and publications on automotive subjects. As a general rule, no attempt is made to give an exhaustive review of the books, the purpose of this section of THE JOURNAL being rather to indicate from time to time what literature relating to the automotive industry has been published with a short statement of the contents.

SYMPOSIUM ON IMPACT TESTING OF MATERIALS. Collection of papers presented at the 25th annual meeting of the American Society for Testing Materials, 1315 Spruce Street, Philadelphia. 107 pp.; 27 illustrations.

This symposium, some of which may be of interest to automotive engineers, begins with a review of Impact Testing of Materials by H. L. Whittemore of the Bureau of Standards, in which the author gives a brief review of the subject, with a selected bibliography. This is followed by a series of three papers by D. J. McAdam, Jr., Thomas R. C. Wilson and Earl B. Smith, on machines and test specimens used in impact tests of metals, wood and road materials respectively.

Then follow two papers by C. L. Warwick and T. D. Lynch on American and British practice respectively, in Notched-Bar Impact-Tests of Metals. Finally, there are three interesting papers on Measurement of Impact by C. E. Margrum, Armin Elmendorf and H. F. Moore, and a paper on the Significance of Impact Tests by F. C. Langenberg and N. Richardson. The four last-mentioned papers in particular contain a critical discussion of some of the accepted methods of measuring impact pressures, and the significance of the results obtained with them, which should be of interest to engineers who are particularly concerned with the behavior of structural materials under conditions of impact, or under rapidly changing load.—H. C. D.

AIRCRAFT SPEED INSTRUMENTS. By Franklin L. Hunt and H. O. Stearns. National Advisory Committee for Aeronautics Report No. 127. Published by the National Advisory Committee for Aeronautics, City of Washington. 38 pp.; illustrated.

This report consists of three parts relating respectively to air-speed indicators, the testing of air-speed indicators and the principles of ground-speed measurement. Part 1 first discusses the different types of instruments that have been proposed for the measurement of air-speed and then gives detailed descriptions of most of the instruments that are used extensively and results of tests. A short discussion of the altitude effect on air-speed indicators is given with a practical table of corrections. Part 2 describes the methods of testing air-speed indicators used at the Bureau of Standards. Part 3 discusses the physical principles of ground-speed measurement and the methods that have been devised for their practical application to the determination of the ground speed of aircraft.

AERONAUTIC DIRECTION INSTRUMENTS, National Advisory Committee for Aeronautics Report No. 128. Published by the National Advisory Committee for Aeronautics, City of Washington.

This report, which is divided into four parts, covers the general field of direction instruments for aircraft. The adequacy of a consideration of the steady state of gyroscopic motion as a basis for a discussion of the displacements of a gyroscope mounted on an airplane is pointed out in the first part. The simple theory of the design of gyroscopic inclinometers and stabilizers is developed on this basis. The principal types of instruments are briefly described and their performance requirements stated. In the second part the testing and use of magnetic compasses for airplanes is dealt with briefly, while Part III contains a general treatment of the important features of the construction of aircraft compasses. Descriptions of the principal types used in America and in foreign countries are included. A brief history of the development of airplane turn indicators with detailed descriptions of all known types and makes is presented in the last section of the report. The results of laboratory and flight tests for the several available gyroscopic turn indicators are appended.

AUTOMOBILE CALCULATIONS. By James Watt. Paper read before the Institution of Automobile Engineers, 28 Victoria Street, Westminster, S. W., London, England. 72 pp.

The aim of the paper is primarily to aid draftsmen and designers, in bringing together in a convenient form calculation methods used in everyday design. The treatment is in the algebraic form and is applicable to light and heavy vehicle design. The calculations are grouped in four classes; the absolute, the comparative, the empirical and the approximate. The care used to distinguish between the types of formulas is very commendable for on the outset it helps the designer to gage the work according to the class of formulas that is being used.

The units of the vehicle treated in the paper are the engine, gearbox, clutch, universal-joints, rear axle, front axle and steering gear, springs, hand and foot brakes, frame and frame mountings, gear-ratio, road performances, special vehicles and accessories, weight of vehicle and the chassis as a unit and its component parts. The units are treated in full only so far as the author deems it to be advisable on the ground that many essential calculations are comparatively simple from a mathematical standpoint and such calculations have been omitted.

The paper is carefully prepared, clearly indicating its limitations as well as setting forth clearly its useful application. The author is to be complimented on his broadminded way in condensing so large a subject as automobile calculations into a paper of comparatively few pages.—A. L. N.

MATERIALS OF ENGINEERING. By Herbert F. Moore, includes a chapter on concrete by Harrison F. Gonnerman. Published by the McGraw-Hill Book Co., New York City. 305 pp.; 110 illustrations.

The author's preface states that the object of this text-book is to furnish a concise presentation of the physical properties of the common materials used in structures and machines, together with a brief description of their manufacture and fabrication. The book is intended primarily for use in technical schools, but may also prove useful to draftsmen, inspectors, machinists, and others, who, dealing with the materials of engineering in their daily work, wish to become familiar in an elementary way with the properties of those materials.

The author discusses the physical properties of materials and their ability to withstand deformation and to resist forces. He outlines the various methods used for testing materials to assure that service requirements will be met and describes concisely the methods of manufacture and fabrication into structures and machines of the different materials in common use.

While the text is elementary in character, for the reader who wishes to pursue his studies further there is given at the end of each chapter a list of selected references.

PNEUMATIC TIRES: AUTOMOBILE, TRUCK, AIRPLANE, MOTORCYCLE, BICYCLE. By Henry C. Pearson. Published by the India Rubber Publishing Co., New York City. 1323 pp.; numerous illustrations.

This book is described in the subtitle as "an encyclopedia of tire manufacture, history, processes, machinery, modern repair and rebuilding, patents, etc., etc., profusely illustrated." The author, a practical rubber-man, has made rubber tires and studied the subject in the leading tire factories of the United States and Europe. As publisher and editor

of the *India Rubber World* he is in a position to secure a large amount of material not accessible to others. The book may be accepted, therefore, as an authoritative treatment of the manufacture of the air-filled tire.

DURALUMIN: A DIGEST OF INFORMATION. By Horace C. Knerr. Paper presented at the Detroit Convention of the American Society for Steel Treating. 29 pp.; 18 illustrations.

Forged and heat-treated-aluminum alloys such as duralumin have been in use for a number of years, and it is not entirely clear why their use has been confined almost entirely to aircraft, in view of their apparent advantages in the automotive field. Aside from price, one of the factors that has limited their general use seems to have been a certain amount of distrust of these materials when produced in quantity for structural purposes.

H. C. Knerr, author of the paper reviewed here, is metallurgist of the Naval Aircraft Factory in Philadelphia. His paper should be of special interest to the automotive engineer, since it gives what appears to be an unbiased discussion of what duralumin is and what may be expected of it.

The history, composition and manufacture of this alloy are covered briefly. Its mechanical properties are discussed at some length and compared with steel for various purposes. There is a chapter on the fabrication of structural parts from duralumin stock and another on corrosion. The latter is treated at considerable length.

The last part of the paper is devoted to the constitution, metallography and theory of heat-treatment. This is of particular interest as it comprises a review of some of the work of several prominent metallurgists, accompanied by a discussion by the author. This is followed by a bibliography of the subject.

EXPERIMENTAL RESEARCH ON AIR PROPELLERS. By W. F. Durand and E. P. Lesley. National Advisory Committee for Aeronautics Report No. 141. 82 pp.; 51 illustrations; 8 plates. Published by the National Advisory Committee for Aeronautics, City of Washington.

This report will be of interest to those who are interested in air propellers. The previous reports of this series, Nos. 14, 30 and 64, have described in detail various findings of an important research project which has been in progress for a number of years at the Leland Stanford, Jr., University under Professor Durand. The present report is a review of the entire series of results of the preceding reports and includes a more complete analysis, graphical and otherwise, of these results. In making this review, any points that were considered doubtful from the results obtained in the first test have been checked by several repetitions of the tests.

The addition of a series of nomographic diagrams expresses the results of this important research in convenient form for use in determining the relative performance characteristics of a very wide variety of propeller designs selected so as to represent a continuous series, rather than distinct and independent types. We believe this report constitutes an important addition to the literature of this difficult subject.

OBITUARY

JOHN B. FOOTE, president and founder of Foote Bros. Gear & Machine Co., Chicago, died Oct. 12, 1922, aged 57 years. He was born at Chicago, April 6, 1865, received a public-school education and supplemented this with night lessons in drawing and in mathematics. He served an apprenticeship as a machinist for 4 years, followed this with practical work at die making and special machinery and, for a number of years, designed automatic machinery for can-manufacturing plants. For the last 29 years he was engaged in the production of automobile and tractor gears and tractor transmissions, having been a pioneer in the making of cut-steel, case-hardened, tough-cored gears for heavy-duty work. He was a charter member of the Society of Tractor Engineers. He was elected to Member grade in the Society of Automotive Engineers when the former organization was absorbed in 1917.

WILLIAM H. LITTLE, active in the automotive industry until his health failed in 1920, died at his home in Detroit, Oct. 26, 1922, aged 46 years. He was born at Westboro, Mass., Feb. 4, 1876. After his preliminary education, his early activities included 5 years of service as chief inspector and general foreman for the Locomobile Co., Bridgeport, Conn.; 1 year as superintendent for the Edison Storage Battery Co.; and 1½ years as manager of the New York City branch office of the Waltham Mfg. Co., Waltham, Mass.

In 1906 he became factory manager for the Buick Motor Co., Flint, Mich. Several years later he organized the Little Motor Car Co. that produced the "Little Car, made by Big Bill Little," as it was known to the trade. With the absorption of this enterprise by the Chevrolet Motor Co., Detroit, Mr. Little became its president. Later, he was general manager of the Sterling engine plant that built Chevrolet engines. Still later, he was made vice-president and managing director of the Scripps-Booth Corporation, Detroit.

Mr. Little was unmarried and is survived by his mother and one sister. He was elected to Member grade in the Society in 1908.

Applicants *for* Membership

The applications for membership received between Oct. 16 and Nov. 15, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ANDERSON, GEORGE POTTER, engineer, Graham Bros., *Detroit.*

AUSTIN, SAMUEL, service manager, Austin Auto Co., *Hartford, Conn.*

BAGGS, HOKE S., president, Norfleet-Baggs, Inc., *Salem, N. C.*

BEAL, S. H., assistant general manager, Jones & Lamson Machine Co., *Springfield, Vt.*

BLACKINTON, GEORGE WILSEY, works manager, Continental Motors Corporation, *Detroit.*

BLAUMAN, LOUIS W., sales engineer, Litho Etching Corporation, *Newark, N. J.*

BOWERS, GEORGE F., student, Purdue University, *Lafayette, Ind.*

BRADER, NORWOOD HAROLD, student, Cornell University, *Ithaca, N. Y.*

BROWN, JAY R., chief engineer, Western Automatic Machine Screw Co., *Elyria, Ohio.*

BROWNE, THEODORE C., vice-president, Brush Laboratories Co., *Cleveland.*

BULLARD, D. B., general mechanical engineer, Bullard Machine Tool Co., *Bridgeport, Conn.*

BULLARD, E. P., JR., president, Bullard Machine Tool Co., *Bridgeport, Conn.*

BULLARD, S. H., vice-president, Bullard Machine Tool Co., *Bridgeport, Conn.*

BURCH, LOUIS D., student, Purdue University, *Lafayette, Ind.*

BUSSARD, ROBERT MCCLELLAND, student, Purdue University, *Lafayette, Ind.*

CARMIN, KENNETH ARNOLD, student, Purdue University, *Lafayette, Ind.*

CORONADO, CHARLES A., student, Ohio State University, *Columbus, Ohio.*

COX, JOHN FITZHEW, student, Georgia School of Technology, *Atlanta, Ga.*

CROS, RENÉ L., student, Ohio State University, *Columbus, Ohio.*

DACOSTA, JOHN C., 3RD, mechanical engineeer, Baldwin Locomotive Works, *Philadelphia.*

DAVIS, LEWIS HENRY, student, Purdue University, *Lafayette, Ind.*

DORAN, JOSEPH F., production engineer, Crown Cork & Seal Co., *Baltimore.*

DYMENT, ALBERT ELLIOTT, sales engineer, Goulds Mfg. Co., *Seneca Falls, N. Y.*

ELLIS, J. H., service manager, Olds Motor Works, *Detroit.*

FEARNSIDE, RALPH STAFFORD, student, Purdue University, *Lafayette, Ind.*

FENN, GEORGE PRENTICE, student, University of Illinois, *Urbana, Ill.*

FREDERICK, CHARLES WALTER, student, University of Michigan, *Ann Arbor, Mich.*

GEBHARDT, C. W., vice-president and engineer, J. J. Schnerr Co., Inc., *San Francisco.*

GLOSSBRENNER, EDGAR L., student, Purdue University, *Lafayette, Ind.*

GODBOLD, E. J., engineer, Climax Engineering Co., *Clinton, Iowa.*

GOODFRIEND, IRVING FREDERICK, student, Columbia University, *New York City.*

GRAY, OTIS K., foreman, Remy Electric Co., *Anderson, Ind.*

GROSS, CLETUS, service manager, Welbon Dayton Motor Car Co., *Dayton, Ohio.*

GROSSER, NELSON O., service engineer, B. Robinson Supplies, Ltd., *Moose Jaw, Saskatchewan, Canada.*

GULF REFINING CO., *Pittsburgh.*

HANNUM, G. H., president and general manager, Oakland Motor Car Co., *Pontiac, Mich.*

HARWELL, ERNEST WILLIE, student, Georgia School of Technology, *Atlanta, Ga.*

HENRY, JAMES S., student, Georgia School of Technology, *Atlanta, Ga.*

HERBRAND CO., *Fremont, Ohio.*

HIGGINS, JAMES J., parachute engineer, Air Service, McCook Field, *Dayton, Ohio.*

HILGEDICK, RALPH V., technician, Stockland Road Machinery Co., *Minneapolis.*

HISCOX, DAVID C., student, Georgia School of Technology, *Atlanta, Ga.*

HOFFMAN, ROBERT G., engineer, Rochester Motors Corporation, *New York City.*

HOLWERDA, HARLEY C., student, Purdue University, *Lafayette, Ind.*

HORI, HISASHI, chief engineer, Yanese Automobile Co., *Tokyo, Japan.*

HUSING, HENRY A., draftsman, Holt Mfg. Co., *Stockton, Cal.*

INGALLS, H. D., assistant superintendent, Air Mail Field, *Cheyenne, Wyo.*

ISDAHL, EINAR, student, University of Wisconsin, *Madison, Wis.*

JOHNSON, PHILIP B., student, Ohio State University, *Columbus, Ohio.*

KELLOGG, H. DUDLEY, JR., student, Yale University, *New Haven, Conn.*

KINSEY, KENNETH HARRY, student, Purdue University, *Lafayette, Ind.*

KLOVER, PETER A., instructor airplane mechanics, United States Civil Service, *Rantoul, Ill.*

KNOWLTON, DALLAS, student, Purdue University, *Lafayette, Ind.*

LAKE, ENSIGN BURTON G., U. S. S. Wright, care Postmaster, *New York City.*

LANDES, JAMES U., engineer, Standard Oil Co., *Portland, Ore.*

LATIMORE, D. S., student, Alabama Polytechnic Institute, *Auburn, Ala.*

LEHMAN, MILTON S., student, Ohio State University, *Columbus, Ohio.*

LOCKWOOD, RALPH G., patent lawyer, Lockwood & Lockwood, *Indianapolis.*

MAEKAWA, EIICHI, electrical engineer, *Tokyo, Japan.*

MAGRAW, GEORGE F., draftsman, Packard Motor Car Co., *Detroit.*

MATSUMOTO, AKIYOSHI, engineer, Dodge Bros., *Detroit.*

MILLER, GEORGE LEE, vice-president and works manager, Gilliam Mfg. Co., *Canton, Ohio.*

MILLS, HARVEY FRETZ, student, Purdue University, *Lafayette, Ind.*

MOE, ORION G., student, Leland Stanford, Jr., University, *Stanford University, Cal.*

NAKASHIMA, CAPT. TOTARO, Motor Transport Corps, Imperial Japanese Army, *Setagaya, near Tokyo, Japan.*

NUTT, FORREST H., student, Purdue University, *Lafayette, Ind.*

PACKARD, JOSEPH A., student, University of Michigan, *Ann Arbor, Mich.*

PATTERSON, JACK WATKINS, student, Georgia School of Technology, *Atlanta, Ga.*

PECK, NELSON CHAFFEE, student, Yale University, *New Haven, Conn.*

PERKINS, ERNEST DELLA, student, University of Michigan, *Ann Arbor, Mich.*

PETERS, JENNINGS D., student, University of Washington, *Seattle, Wash.*

PHILLIPS, WILLIAM RUSSELL, JR., student, Georgia School of Technology, *Atlanta, Ga.*

PLACE, REUBEN MEREDITH, branch manager, Ahlberg Bearing Co., *Toledo.*

REYBURN, JOHN R., engineer, American Chain Co., *Bridgeport, Conn.*

RHAME, P. W., instructor, University of Minnesota, *Minneapolis.*

RIGGS, HAROLD T., student, Purdue University, *Lafayette, Ind.*

ROSS, WALTER DAVID, student, Purdue University, *Lafayette, Ind.*

ROSSIN, MAURICE S., student, Purdue University, *Lafayette, Ind.*

ROTH, GEORGE F., factory manager, Anchor Top & Body Co., *Cincinnati.*

RUFF, EDWARD ALBERT, student, Ohio State University, *Columbus, Ohio.*

SCHADT, E. K., electric engineer, Cadillac Motor Car Co'., *Detroit.*

SCHNERR, J. J., manufacturer of gears and automobile parts, J. J. Schnerr Co., Inc., *San Francisco.*

SCHWIZER, PAUL EUGENE, student, Polytechnic Institute of Brooklyn, *Brooklyn, N. Y.*

SENDELBACH, EDWARD C., manager of wheel department, Hopkins Mfg. Co., *Hanover, Pa.*

SHELLER, GEORGE A., service manager, William Parkinson Motor Sales Co., *New York City.*

SHICK, WILLIAM KENNETH, student, Purdue University, *Lafayette, Ind.*

SIMPSON, E. L., president and engineer, E. L. Simpson, Ltd., *Montreal, Que., Canada.*

SLOCOMBE, W. VERNON, assistant engineer in charge of production, Turbulator Corporation, *Chicago.*

SOLT, VIVIAN M., builder of racing cars and student, Solt Motors, *Manhattan, Kan.*

SPRING, FRANK S., chief engineer, Courier Motors Co., *Sandusky, Ohio.*

STENBERG, THORNTON R., sales representative, Walden-Worcester, Inc., *Worcester, Mass.*

STEVENSON, ROSS J., student, University of Illinois, *Urbana, Ill.*

STRAIN, CLIFFORD, student, Stevens Institute of Technology, *Hoboken, N. J.*

SWANSON, RAYMOND E., student, Purdue University, *Lafayette, Ind.*

TAYLOR, J. H., student, Purdue University, *Lafayette, Ind.*

TAYLOR, ROBERT HUGH, assistant chief engineer, Universal Boiler Co., *Denver, Col.*

TERMAN, MARK J., student, Purdue University, *Lafayette, Ind.*

TEXTILEATHER CO., *New York City.*

THOMAS, THEODORE P., student, Purdue University, *Lafayette, Ind.*

THOMPSON, GORDON B., student, Purdue University, *Lafayette, Ind.*

TODD, FILLMORE W., president, Accessories Mfg. Co., *Chicago.*

VIDALIE, RENÉ M., designer-engineer, Doble Steam Motors, *San Francisco.*

WALLACE, BERNARD W., student, Purdue University, *Lafayette, Ind.*

WIEBERG, T. C., student, Purdue University, *Lafayette, Ind.*

WILCOX, FRED A., engineer, Advance-Rumely Co., *La Porte, Ind.*

WILKINSON, JAMES MCCLELLAN, student, Georgia School of Technology, *Atlanta, Ga.*

WILLIAMS, EMERSON MARION, student, University of Michigan, *Ann Arbor, Mich.*

WOLEVER, WALTER BRIAN, student, Purdue University, *Lafayette, Ind.*

WORMLEY, HAROLD W., assistant carbureter engineer, Cadillac Motor Car Co., *Detroit.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Oct. 10 and Nov. 10, 1922. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member; (E S) Enrolled Student.

ANTHONY, JOHN EDWARD (M) designing engineer, tractor works, International Harvester Co., *Chicago,* (mail) 4112 West Jackson Boulevard.

BEAN, WILLIAM LLOYD (M) mechanical assistant to president, New York, New Haven & Hartford Railroad, *New Haven, Conn.*

BREWSTER, WILLIAM (M) president, Brewster & Co., Bridge Plaza, *Long Island City, N. Y.*

BROWN, JOHN W. (A) general manager, John W. Brown Mfg. Co., *Columbus, Ohio.*

BUTCHER, HAROLD E. (A) sales manager, Champion Spark Plug Co., *Toledo.*

BUTTRICK, W. B. (A) chief mechanic and superintendent, Miller North Broad Storage Co., *Philadelphia,* (mail) 1521 Venango Street.

COLEY, GLENN (A) metallurgist, Timken-Detroit Axle Co., 136 Clark Avenue, *Detroit.*

HOLZ, FRED C. (A) service inspector, Gomery-Schwartz Motor Car Co., *Philadelphia,* (mail) 157 North 20th Street.

JACKSON, E. F. (A) manager of manufacturers sales, Goodyear Tire & Rubber Co., Inc., *Detroit,* (mail) 2817 East Grand Boulevard.

LUTZ, EARL C. (M) assistant superintendent, International Harvester Co., Chicago, (mail) 1913 South 49th Avenue, *Cicero, Ill.*

NATIONAL MALLEABLE CASTINGS CO. (Aff) 10,600 Quincy Avenue, *Cleveland.*
Representatives:
Bellman, W. B., sales agent, Toledo.
Hiatt, H. I., sales agent, Chicago.
Slater, James A., assistant manager of sales.
Wasson, S. C., sales agent, Indianapolis.

NEAL, JAMES BENSON (M) manager, Norton Laboratories, Inc., *Lockport, N. Y.*

NORWOOD, H. E. (M) general manager and engineer, Perfect Window Regulator Co., 20 Exchange Place, *New York City.*

SMITH, STANFORD ALLEN (M) chief inspector, Lexington Motor Co., *Connersville, Ind.*

STREETER, EDWARD L., JR. (A) general manager, J. N. LaPointe Co., *New London, Conn.,* (mail) 85 Squire Street.

TAKAO, SHIGEZO (F M) engineer, Takao Iron Works Co., Kobe, Japan, (mail) *Oishi near Kobe, Japan.*

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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VOL·XI NO·6

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

S·A·E

DECEMBER 1922

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
29 WEST 39TH STREET NEW YORK

MOTOR CAR SPRINGS, because they are capable of being compressed, keep the *bumps* from kicking the car body and passengers, and Stabilators then keep the compressed *springs* from kicking the car body and passengers.

Stabilators control this spring kick or recoil in proportion to the *force* of the recoil. As this force may be anything from 1 pound to 1000 or 2000 pounds, depending upon how far the spring has been compressed by the bump, the necessity for *proportional* control is quite evident.

Stabilators are the only devices constructed today which give proportional control.

John Warren Watson Company
Twenty-Fourth and Locust Streets
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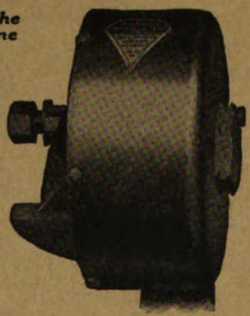
Look for the Silver Name Plate



Exactly opposite to snubbing

In checking spring recoil, Stabilators work exactly opposite to the snubbing principle. Instead of checking with a jerk at the tail end ot the recoil movement, Stabilators get on the job at the very beginning of the movement and smoothly ease you back to normal. Results produced by the ne method give no conception of those produced by the other. They are different to the point of absolute oppositeness.

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